INTERNAL COMBUSTION ENGINE FUNDAMENTALS

SECOND EDITION



Mc Graw Hill Education

JOHN B. HEYWOOD

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Internal Combustion Engine Fundamentals

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This second edition of my text is dedicated to my family: my wife Peggy and our sons Jamie, Stephen (who died from ALS in 2006), and Ben. They, and their families, have been wonderfully supportive of my efforts in this challenging endeavor. For this, I am truly grateful.

A documentary film, So Much So Fast, was made about ALS , Stephen, Jamie, and our life together. The Los Angeles film critic's review said of our family: "And what a family; close-knit, loving, and fiercely loyal." I treasure those words.

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Preface

There are about two billion internal combustion engines in use in the world today. These engines enable key areas of our daily lives, propelling our many vehicles, generating electricity, and providing mechanical power in a wide range of applications. Their origin dates back to 1876 when Nicolaus Otto first developed the spark-ignition engine, and 1892 when Rudolf Diesel invented the compression-ignition engine. Since then, the utility of spark-ignition and diesel engines has steadily improved as our understanding of engine processes has increased, new technologies have become available, and market and regulatory requirements have become more demanding. A sense of urgency, driven largely by our need to combat global climate change, now requires ever-faster development of better engines and fuels, and exploration of alternative approaches. Increasing engine power density and efficiency and reducing engine emissions are really important objectives. The availability and effective use of our expanding knowledge base on engines and fuels are therefore critical.

The 1988 edition of this book has served as an educational text and professional reference in response to that need for some thirty years. Since 1988, we have steadily improved engine performance, reduced engine fuel consumption, developed air pollutant control technologies that have reduced engine emissions, improved the quality of our mainstream petroleum-based fuels, and made a start on reducing transportation's greenhouse gas emissions. Obviously, over these past thirty years, much new engineering knowledge relevant to engines and fuels has been developed. The purpose of this second edition of my book is to incorporate this new material, update the existing knowledge base, and make available a modern, broader, and thus more useful engine text and reference.

There is a massive amount of material, both analytical and experimentally based, available on internal combustion engines, and no text can include it

all. The emphasis here is on the key physical and chemical processes that govern engine operation and design. These include the thermodynamics of energy conversion in engines, the physics and chemistry that govern engine combustion, the engine's fuel requirements, the important fluid flow, heat transfer, friction, and lubrication processes in engines, and the engine's dynamic behavior. These all influence engine performance, efficiency, and emissions.

There are two main types of internal combustion engine: spark-ignition and diesel. The primary organizing approach for the material in this text is how the mixture of fuel and air formed inside the engine cylinder is ignited. From *method of ignition*—spark-ignited or compression-ignited—follows each type of engine's operating cycle, fuel requirements, mixture preparation approach, combustion process, combustion chamber configuration, method used to control load, air pollutant formation mechanisms and control approaches, performance and efficiency characteristics, and greenhouse gas emissions. While many engine processes are similar in both types of engines, the method of ignition is fundamentally different. The consequences of that difference underlie the overall organization of this book.

The book is arranged in four major sections. The first (Chaps. 1 to 5) provides an introduction to, and overview of, the operating characteristics of spark-ignition and compression-ignition (diesel) engines, defines the parameters used to characterize engine operation, and develops the thermodynamics and combustion theory required for a quantitative analysis of engine behavior. It concludes with an integrated treatment of the methods used for analyzing idealized models of internal combustion engine operating cycles.

The second section (Chaps. 6 to 8) focuses on engine flow phenomena. The details of the gas exchange process—intake and exhaust processes in four-stroke and scavenging in two-stroke cycle engines—and the various methods of boosting engines—turbocharging and supercharging—are discussed. The several fuel metering approaches used in spark-ignition engines are reviewed next. Then, the key features of the flows setup inside the engine cylinder are described. These flow processes (along with engine boost levels) control the amount of air the engine will induct, and thus its power, and largely govern the rate at which the fuel-air mixture in the cylinder will burn.

The third section of the book examines engine combustion phenomena. These chapters (9 to 11) are especially important to smooth and robust engine behavior. The combustion process releases the fuel's chemical energy within the engine's cylinders for eventual conversion to work. The fraction of the fuel's energy that is converted depends strongly on how that combustion process occurs. The spark-ignition and compression-ignition combustion processes (Chaps. 9 and 10, respectively) thus influence essentially all aspects of engine behavior. Air pollutants are undesirable byproducts of combustion. Our extensive knowledge of how air pollutants form inside the engine and how such emissions can be controlled is reviewed in Chap. 11 .

The last section of the book focuses on the operating characteristics of these engines. First, the basics of engine heat transfer and friction, both of which degrade engine performance, are developed in Chaps. 12 and 13. Then, Chap. 14 describes the methodologies available for predicting details of key engine processes and overall engine behavior based on realistic models of engine flow and combustion phenomena. The various thermodynamic- and fluid-mechanic-based computer codes that have been developed over the past several decades are widely used in engine research, development, and design, so knowledge of their basic structure and capabilities is important. Chapter 15 then summarizes how the operating characteristics—power, efficiency, and emissions—of spark-ignition and diesel engines depend on the major engine design and operating variables. These final two chapters effectively integrate our analysis-based understanding and practical knowledge of individual engine processes together to describe and explain overall spark-ignition engine, and compression-ignition engine, behavior.

While this book contains much advanced material on engine design and operation intended for the practitioner, each major topic is developed from its beginnings and the more sophisticated chapters have introductory sections to facilitate their use in undergraduate courses. The chapters are extensively cross-referenced and indexed. Each chapter is fully illustrated and referenced, and includes problems for both undergraduate and graduate student courses.

John B. Heywood

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I am indebted also to many companies in the automotive and energy industries that have supported the research that my colleagues and I have carried out on engine and fuels topics. The fellow professionals in these companies who have helped us in our work (much of it relevant to topics I cover in this book) have provided important practical knowledge and insights that significantly enhanced the value of what we have been able to do. Many engineering advances are a result of team efforts, and as a researcher I really appreciate the contributions of engine practitioners. I also thank those in the engine business who helped me by providing drawings of recent engines that help anchor engine theory in the real world. Their companies are acknowledged in the relevant figure captions.

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Commonly Used Symbols, Subscripts, and Abbreviations [a]

SYMBOLS

a	Acceleration
	Crank radius
	Sound speed
	Specific availability
A	Area
A_C	Valve curtain area
A_{ch}	Cylinder head area
A_e	Exhaust port area
A_E	Effective area of flow restriction
A_i	Inlet port area
A_p	Piston crown area
A_v	Valve open area
B	Cylinder bore
	Steady-flow availability
c	Distance of piston from TC position
	Specific heat
c_p	Specific heat at constant pressure
c_s	Soot concentration (mass/volume)
c_v	Specific heat at constant volume

C	Absolute gas velocity
C_D	Discharge coefficient
	Vehicle drag coefficient
C_s	Swirl coefficient
D	Diameter
	Diffusion coefficient
D_d	Droplet diameter
D_{SM}	Sauter mean droplet diameter
D_v	Valve diameter
e	Radiative emissive power
	Specific energy
E_A	Activation energy
f	Coefficient of friction
	Fuel mass fraction
F	Force
g	Gravitational acceleration
	Specific Gibbs free energy
G	Gibbs free energy
h	Clearance height
	Oil film thickness
	Specific enthalpy
h_c	Heat-transfer coefficient
h_p	Port open height
h_s	Sensible specific enthalpy
H	Enthalpy
I	Moment of inertia
J	Flux
k	Thermal conductivity
	Turbulent kinetic energy
k_i^+, k_i^-	Forward, backward rate constants for ith reaction

K	Constant
K_c	Equilibrium constant expressed in concentrations
K_p	Equilibrium constant expressed in partial pressures
l	Characteristic length scale
	Connecting rod length
l_T	Characteristic length scale of turbulent flame
L	Piston stroke
L_I	Sound intensity level
$\tilde{L}_{\mathrm{LO}}$	Normalized spray lift-off length
L_n	Fuel-injection-nozzle orifice length
L_v	Valve lift
m	Mass
$\dot{m}$	Mass flow rate
m_r	Mass of residual gas
M	Mach number
	Molecular weight
n	Number of moles
	Polytropic exponent
n_{c}	Number of cylinders
n_R	Number of crank revolutions per power stroke
N	Crankshaft rotational speed
	Soot particle number density
	Turbocharger shaft speed
p	Cylinder pressure
	Pressure
P	Power
$\dot{q}$	Heat-transfer rate per unit area
	Heat-transfer rate per unit mass of fluid
Q	Heat transfer
$\dot{Q}$	Heat-transfer rate

Q_{ch}	Fuel chemical energy release or gross heat release
Q_{HV}	Fuel heating value
Q_n	Net heat release
r	Radius
r_c	Compression ratio
R	Connecting rod length/crank radius
	Gas constant
	Radius
R^+, R^-	One-way reaction rates
R_s	Swirl ratio
s	Crank axis to piston pin distance
	Specific entropy
S	Entropy
	Spray penetration
S_b	Turbulent burning speed
S_L	Laminar flame speed
S_p	Piston speed
t	Time
T	Temperature
	Torque
u	Specific internal energy
	Velocity
u′	Turbulence intensity
u_s	Sensible specific internal energy
u_T	Characteristic turbulent velocity
U	Compressor/turbine impellor tangential velocity
	Fluid velocity
	Internal energy
v, u	Specific volume
	Velocity

v_{ps}	Valve pseudo-flow velocity
v_{sq}	Squish velocity
V	Cylinder volume
	Volume
V_c	Clearance volume
V_d	Displaced cylinder volume
w	Relative gas velocity
	Soot surface oxidation rate
W	Work transfer
W_c	Work per cycle
W_p	Pumping work
x, y, z	Spatial coordinates
x	Mass fraction
$\tilde{x}$	Mole fraction
x_b	Burned mass fraction
x_r	Residual mass fraction
y	H/C ratio of fuel
	Volume fraction
Y_α	Concentration of species α per unit mass
z	Distance, piston crown to cylinder head
Z	Inlet Mach index
a	Angle
	Thermal diffusivity $k/(\rho c)$
β	Angle
γ	Specific heat ratio c_p/c_v
Γ	Angular momentum of charge
δ	Boundary-layer thickness
δ_L	Laminar flame thickness
$\Delta h^\circ_{f,i}$	Molal enthalpy of formation of species i
$\Delta\theta_b$	Rapid burning angle

$\Delta\theta_d$	Flame development angle
ε	$4/(4 + y)$: y = H/C ratio of fuel
	Turbulent kinetic energy dissipation rate
ζ	Percentage of stoichiometric air entrained into fuel spray
η_a	Availability conversion efficiency
η_c	Combustion efficiency
η_{ch}	Charging efficiency
η_f	Fuel conversion efficiency
η_T	Turbine isentropic efficiency
η_{tr}	Trapping efficiency
η_v	Volumetric efficiency
θ	Crank angle
λ	Relative air/fuel ratio
Λ	Delivery ratio
μ	Dynamic viscosity
μ_i	Chemical potential of species i
ν	Kinematic viscosity μ/ρ
ν_i	Stoichiometric coefficient of species i
ξ	Flow friction coefficient
ρ	Density
$\rho_{a,0}, \rho_{a,i}$	Air density at standard, inlet conditions
σ	Normal stress
	Standard deviation
	Stefan-Boltzmann constant
	Surface tension
τ	Characteristic time
	Induction time
	Shear stress
τ_{id}	Ignition delay time
ϕ	Fuel/air equivalence ratio

Φ	Flow compressibility function [Eq. (C.11)]
	Isentropic compression function [Eqs. (4.15 *b*) , (4.25*b*)]
ψ	Molar N/O ratio
Ψ	Isentropic compression function [Eqs. (4.15 *a*) , (4.25*a*)]
ω	Angular velocity
	Frequency

SUBSCRIPTS

a	Air
b	Burned gas
c	Coolant
	Cylinder
C	Compression stroke
	Compressor
cr	Crevice
e	Equilibrium
	Exhaust
E	Expansion stroke
f	Flame
	Friction
	Fuel
g	Gas
i	Indicated
	Intake
	Species *i*
ig	Gross indicated
in	Net indicated
l	Liquid
L	Laminar
p	Piston

	Port
P	Prechamber
r , θ, z	r, θ, z components
R	Reference value
s	Isentropic
T	Nozzle or orifice throat
	Turbine
	Turbulent
u	Unburned
v	Valve
w	Wall
x , y, z	x, y, z components
0	Reference value
	Stagnation value

NOTATION

Δ	Difference
-	Average or mean value
~	Value per mole
[ ]	Concentration, moles/vol
{ }	Mass fraction
.	Rate of change with time
u	Bold type, vector (e.g., velocity)

ABBREVIATIONS

(A/F)	Air/fuel ratio
BC, ABC, BBC	Bottom-center crank position, after BC, before BC
bmep	Brake mean effective pressure

CN	Fuel cetane number
Da	DamkÖhler number τ_T/τ_L
EGR	Exhaust gas recycle
EI	Emission index
EPC, EPO	Exhaust port closing, opening
EVC, EVO	Exhaust valve closing, opening
(*F/A*)	Fuel/air ratio
(*G/F*)	Gas/fuel ratio
gimep	Gross indicated mean effective pressure
IPC, IPO	Inlet port closing, opening
IVC, IVO	Inlet valve closing, opening
mep	Mean effective pressure
nimep	Net indicated mean effective pressure
Nu	Nusselt number $h_c l/k$
ON	Fuel octane number
Re	Reynolds number $\rho u l/\mu$
sfc	Specific fuel consumption
TC, ATC, BTC	Top-center crank position, after TC, before TC
We	Weber number $\rho_l u^2 D/\sigma$

[a] Nomenclature specific to a section or chapter is defined in that section or chapter.

CHAPTER 1

Engine Types and Their Operation

1.1 INTRODUCTION AND HISTORICAL PERSPECTIVE

The purpose of internal combustion engines is to produce mechanical power from the chemical energy contained in the fuel. In *internal* combustion engines, as distinct from *external* combustion engines, this energy is released by burning or oxidizing the fuel *inside* the engine. The fuel-air mixture before combustion and the burned products after combustion are the actual working fluids. The work transfers that provide the desired power output occur directly between these working fluids and the mechanical components of the engine. The internal combustion engines that are the subject of this book are spark-ignition (SI) engines (sometimes called Otto engines, or gasoline or petrol engines, though other fuels can be used) and compression-ignition (CI) or diesel engines. [a] Because of their simplicity, ruggedness, high power to weight ratio, efficiency, and low cost, these two types of engine have found wide application in transportation (land, sea, and air) and power generation. It is the fact that combustion takes place inside the work-producing part of these engines that makes their design and operating characteristics fundamentally different from those of other types of engine.

Power-producing engines have served human beings for over two and a half centuries. For the first 150 years, water, converted to steam, was interposed between the combustion gases produced by burning the fuel and the work-producing piston-in-cylinder expander. It was not until the 1860s that the internal combustion engine became a practical reality. [1, 2] The early

engines developed for commercial use burned coal-gas air mixtures at atmospheric pressure—there was no compression before combustion. J. J. E. Lenoir (1822–1900) developed the first marketable engine of this type. Gas and air were drawn into the cylinder during the first half of the piston stroke. The charge was then ignited with a spark, the pressure increased, and the burned gases then delivered power to the piston for the second half of the stroke. The cycle was completed with an exhaust stroke. Some 5000 of these engines were built between 1860 and 1865 in sizes up to six horsepower. Efficiency was at best about 5%.

A more successful development—an atmospheric engine introduced in 1867 by Nicolaus A. Otto (1832–1891) and Eugen Langen (1833–1895)—used the pressure rise resulting from combustion of the fuel-air charge early in the outward stroke to accelerate a free piston and rack assembly so its momentum would generate a vacuum in the cylinder. Atmospheric pressure then pushed the piston inward, with the rack engaged through a roller clutch to the output shaft. Production engines, of which about 5000 were built, obtained thermal efficiencies of up to 11%. A slide valve controlled intake, ignition by a gas flame, and exhaust.

To overcome this engine's shortcomings of low thermal efficiency and excessive size and weight, Otto proposed an engine cycle with four piston strokes: an intake stroke, then a compression stroke before ignition, an expansion or power stroke where work was delivered to the crankshaft, and finally an exhaust stroke. He also proposed incorporating a stratified-charge induction system, though this was not achieved in practice. His prototype four-stroke engine first ran in 1876. A comparison between the Otto engine and its atmospheric-type predecessor indicates the reason for its success (Table 1.1): the enormous reduction in engine weight and volume. This was the breakthrough that effectively founded the internal combustion engine industry. By 1890, almost 50,000 of these engines had been sold in Europe and the United States.

TABLE 1.1 Comparison of Otto's early four-stroke cycle and Otto-Langen's engines [2]

	Otto and Langen	Otto's four-stroke
Brake horsepower	2	2
Weight, lb, approx.	4000	1250
Piston displacement, in^3	4900	310
Power strokes per minute	28	80
Shaft speed, rev/min	90	160
Mechanical efficiency, %	68	84
Overall efficiency, %	11	14
Expansion ratio	10	2.5

In 1884, an unpublished French patent issued in 1862 to Alphonse Beau de Rochas (1815–1893) was found that described the principles of the four-stroke cycle. This chance discovery cast doubt on the validity of Otto's own patent for this concept, and in Germany, it was declared invalid. Beau de Rochas also outlined the conditions under which maximum performance and efficiency in an internal combustion engine could be achieved. These were:

1. The largest possible cylinder volume with the minimum boundary surface
2. The greatest possible working speed
3. The greatest possible expansion ratio
4. The greatest possible pressure at the beginning of expansion

The first condition holds heat losses from the charge to a minimum. The second condition increases the power output from a given size engine. The third condition recognizes that the greater the expansion of the postcombustion gases, the greater the amount of work extracted. The fourth condition recognizes that higher initial pressures make greater expansion possible and give higher pressures throughout the process, both resulting in greater work transfer. Although Beau de Rochas' unpublished writings predate Otto's developments, he never reduced these ideas to practice. Thus Otto, in the broader sense, was the inventor of the modern internal combustion engine as we know it today.

Further developments followed fast once the full impact of what Otto had achieved became apparent. By the 1880s, several engineers (e.g., Dugald Clerk, 1854–1913, James Robson, 1833–1913, in England, and Karl Benz, 1844–1929, in Germany) had successfully developed two-stroke cycle

internal combustion engines where the exhaust and intake processes occur during the end of the power stroke and the beginning of the compression stroke. James Atkinson (1846–1914) in England made an engine with a longer expansion than compression stroke, which had a high efficiency for the times but mechanical weaknesses. It was recognized that efficiency was a direct function of expansion ratio, yet compression ratios were limited to less than four if serious knock problems were to be avoided with the available fuels. Substantial carburetor and ignition system developments were required, and occurred, before high-speed gasoline engines suitable for automobiles became available in the late 1880s. Stationary engine progress also continued. By the late 1890s, large single-cylinder engines of 1.3-m bore fueled by low-energy blast furnace gas produced 600 bhp at 90 rev/min. In Britain, legal restrictions on volatile fuels turned their engine builders toward kerosene. Low compression ratio "oil" engines with heated external fuel vaporizers and electric ignition were developed with efficiencies comparable with those of gas engines (14 to 18%). The Hornsby-Ackroyd engine became the most popular oil engine in Britain, and was also built in large numbers in the United States. [2]

In 1892, the German engineer Rudolf Diesel (1858–1913) outlined in his patent a new form of internal combustion engine. His concept of initiating combustion by injecting a liquid fuel into the high-temperature air in the cylinder produced by compression permitted a doubling of efficiency over the other internal combustion engines then available. Much greater compression and expansion ratios, without detonation or knock, were now possible. However, even with the efforts of Diesel and the resources of M.A.N. in Ausburg combined, it took 5 years to develop a practical engine.

Engine developments, perhaps less fundamental but nonetheless important to the steadily widening internal combustion engine markets, have continued ever since. [2-4] There has always been an interest in engine geometries different from the standard reciprocating piston-in-cylinder, connecting rod, and crankshaft arrangement. Especially, there has been an interest in rotary internal combustion engines. Although a wide variety of experimental rotary engines have been proposed over the years, [5] the first practical rotary internal combustion engine, the Wankel, was not successfully tested until 1957. That engine, which evolved through many years of research and development, was based on the designs of the German inventor Felix Wankel.

[6, 7] While the Wankel engine has been used in niche markets, its advantages of compactness and smoother operation have not been sufficient to overcome its high manufacturing cost.

Fuels have also had a major impact on engine development. The earliest engines used for generating mechanical power burned gaseous fuels. Gasoline, and lighter fractions of crude oil, became available in the late 1800s, and various types of carburetors were developed to vaporize the fuel and mix it with air. Before about 1905, there were few issues with gasoline; though compression ratios had to be low (4 or less) to avoid knock, the highly volatile fuel made starting easy and gave good cold weather performance. However, a serious crude oil shortage developed, and to meet the fivefold increase in gasoline demand between 1907 and 1915, the yield from crude had to be raised. Through the work of William Burton (1865–1954) and his associates of Standard Oil of Indiana, a thermal cracking process was developed whereby heavier oils were heated under pressure and decomposed into less complex, more volatile compounds. These thermally cracked gasolines satisfied demand, but their higher boiling point range created cold weather starting problems. Fortunately, electrically driven starters, introduced in 1912, came along just in time.

On the farm, kerosene was the logical fuel for internal combustion engines since it was used for heat and light. Many early farm engines had heated carburetors or vaporizers to enable them to operate with such a fuel.

The period following World War I saw a tremendous advance in our understanding of how fuels affect combustion, and especially the problem of knock. The antiknock effect of tetraethyl lead was discovered at General Motors, [4] and it became commercially available as a gasoline additive in the United States in 1923. In the late 1930s, Eugene Houdry found that vaporized oils passed over an activated catalyst at 450 to 480°C were converted to high-quality gasoline in much higher yields than was possible with thermal cracking. These advances, and others, permitted fuels with ever better antiknock properties to be produced in large quantities; thus engine compression ratios steadily increased, improving power and efficiency.

During the past several decades, new factors for change have become important and now significantly affect engine design and operation. These factors are, first, the need to control the automotive contribution to urban air

pollution and, second, the need to achieve significant improvements in automotive fuel consumption.

The automotive air-pollution problem became apparent in the 1940s in the Los Angeles basin. In 1952, it was demonstrated by Prof. A. J. Haagen-Smit that the smog problem there resulted from reactions between oxides of nitrogen and hydrocarbon compounds in the presence of sunlight. [8] In due course it became clear that the automobile was a major contributor to hydrocarbon and oxides of nitrogen emissions, as well as the prime cause of high carbon monoxide levels in urban areas. Diesel engines are a significant source of small soot or smoke particles, as well as hydrocarbons and oxides of nitrogen. Table 1.2 outlines the dimensions of the problem. As a result of these developments, emission standards for automobiles were introduced first in California, then nationwide in the United States, starting in the 1960s. Emission standards in Japan and Europe, and for other engine applications, have followed. Substantial reductions in emissions from spark-ignition and diesel engines have been achieved. Both the use of catalysts in SI engine exhaust systems for emissions control and concern over the toxicity of lead antiknock additives have resulted in the reappearance of unleaded gasoline as the dominant part of the automotive fuels market. These emission-control requirements and fuel developments have produced significant changes in the way internal combustion engines are now designed and operated.

TABLE 1.2 The automotive urban air-pollution problem: typical vehicle emissions *

Pollutant	Impact	Mobil source emissions as % of total†	Automobile emissions, SI engines: Precontrol vehicles, g/km‡	Automobile emissions, SI engines: Current vehicles, g/km	Truck emissions, diesel engines: Precontrol engines, g/kWh	Truck emissions, diesel engines: Current engines, g/kWh
Oxides of nitrogen (NO and NO_2)	Reactant in photochemical smog; NO_2 is toxic	50–60	2.0	0.03	21	0.25
Carbon monoxide (CO)	Toxic	60	60	2	~ 20	low
Unburned hydrocarbons (HC, many hydrocarbon compounds)	Reactant in photochemical smog	25	10	0.05	~ 1	low
Particulates (soot, hydrocarbons, sulfates)	Some of HC compounds mutagenic; reduces visibility	5–10	0.5§	0.007§	1	0.02

*Varies from country to country. The United States, Canada, Western Europe, and Japan have standards with different degrees of severity. The United States, Europe, and Japan have different test procedures. Standards are strictest in the United States and Japan.
†Depends on type of urban area and source mix. Approximate percentages.
‡Average values for pre-1968 automobiles that had no emission controls, determined by U.S. test procedure that simulates typical urban and highway driving. Exhaust emissions, except for HC where 55% are exhaust emissions, 20% are evaporative emissions from fuel tank and carburetor, and 25% are crankcase blowby gases.
§Diesel engine automobiles only. Particulate emissions from spark-ignition engines are relatively low.

Internal combustion engines are also an important source of noise. There are several sources of engine noise: the exhaust system, the intake system, the fan used for cooling, and the engine block surface. The noise may be generated by aerodynamic effects, may be due to forces that result from the combustion process, or may result from mechanical excitation by rotating or reciprocating engine components. Vehicle noise legislation to reduce this impact on the ambient environment (and thus on people) was first introduced in the early 1970s.

During the 1970s, the price of crude petroleum rose rapidly to several times its cost (in real terms) in 1970. In the 1980s, the price of crude oil fell, and then fluctuated at relatively low levels until the early 2000s when it rose to close to its late 1970s values. The price then fell rapidly, and then rose again. Currently, the growth in oil demand in the developing world, the uncertainty in future extraction from established fields and discovery of new sources of oil, and the nonuniform concentration of petroleum reserves in a few nations, suggest that the balance between global oil production and transportation fuel demand will be tight over the next few decades. This uncertainty regarding the longer-term availability of adequate supplies of petroleum-based fuels is creating substantial pressures for significant improvements in internal combustion engine efficiency (in all the engine's many applications). Much work is being done to develop the supply and use of alternative fuels to gasoline and diesel. Of the nonpetroleum-based fuels,

natural gas, methanol (methyl alcohol), and biomass-derived fuels such as ethanol (ethyl alcohol) and biodiesel are receiving significant attention. Synthetic gasoline and diesel are being made from tar (oil) sands, and could be produced from shale oil or coal. Hydrogen is being considered as a longer-term zero carbon containing possibility.

The growing consumption of fossil fuels has raised the concern that the greenhouse gas (GHG) emissions from our energy supply and use are causing global warming that could lead to changes in our climate. Emissions of carbon dioxide, along with other GHGs—methane, nitrous oxide, three groups of fluorinated gases (sulfur hexafluoride, hydrofluorocarbons, and perfluorocarbons), ozone—will need to be significantly reduced over the next several decades. Thus, internal combustion engines will need to become more efficient, and low GHG emitting sources of energy will need to be developed so that consumption of petroleum-based fuels—gasoline and diesel—can be significantly reduced. Transportation is estimated to be the source of about one-quarter of the world's GHG emissions.

Table 1.3 lists the CO_2 emissions of various fuels and other sources of energy that might be used in transportation. Emissions from the various fossil fuels listed vary by about a factor of two. Emissions from biofuel production are generally lower (and could be significantly lower), depending on the biomass feedstock, the choice of fuel produced, and the process used to produce that fuel. [b]

TABLE 1.3 CO_2 emissions per unit chemical energy from various fuels or energy sources [9]

	gCO_2/MJ
Gasoline	93
Diesel (fuel oil)	99
Natural gas	74
Liquid petroleum gas	86
Ethanol*	34–73
Biodiesel	45–73
Hydrogen†	100–200
Electricity‡	90–160

*Varies with biomass feedstock and process used.
†From steam reforming of natural gas (low end) or from electrolysis (high end).
‡Depends on electricity generating system source mix (especially the fraction from coal).

The lower value given for hydrogen (which contains no carbon) is based on the current industrial hydrogen production process—steam reforming of natural gas. The electricity carbon dioxide-emissions intensity value depends on the mix of coal, natural gas, nuclear, hydro, wind (and solar) used to generate the electricity. While this electricity generating mix varies country to country, the major roles of coal and natural gas are common to most regions.

What would such fuel changes mean for internal combustion engines? With appropriate changes in engine design and operation, natural gas and the liquid fuels listed in Table 1.3 can be effectively utilized; indeed engines using these fuels are in use today. While the potential for hydrogen as a major transportation energy source (actually an *energy storage* medium) is partly based on large-scale use of highly efficient fuel cell technology, it can be used effectively in suitably designed SI engines. Vehicle propulsion system electrification is already occurring through the use of hybrid electric vehicle (HEV) technology—a combination of a battery, electric motor, internal combustion engine, and generator. The next step in vehicle electrification is to expand the battery's energy storage capacity and recharge (in part) from the electricity supply grid: deployment of this plug-in hybrid (PHEV) technology is occurring. HEV and PHEV propulsion systems require an internal combustion engine, albeit with specific characteristics that improve its efficiency (see Sec. 1.7.2). Some view the pure battery electric vehicle is

the final step in this electrification process. Whether, and how far into the future complete electrification might occur is currently unclear.

This brings us back to internal combustion engines. It might be thought that after over a century of development, the internal combustion engine has reached its peak and little potential for further improvement remains. Such is not the case. As spark-ignition and diesel engine technology evolves, these engines continue to show substantial improvements in efficiency, power density, degree of emission control, and operational capacity. Changes in engine operation and design are steadily improving engine performance in its broadest sense. New materials becoming available and more knowledge-based design offer the potential for continuing to reduce engine weight, size, and cost, for a given power output, and for different and more efficient internal combustion engine concepts. Emissions control technologies, in both the engine and the exhaust system, are becoming more effective and robust. Variable valve control is replacing fixed valve control approaches, with performance and efficiency benefits. Direct-injection gasoline engines, which offer improved dynamic engine control relative to port fuel injection, are now in large-scale production. These technologies are enabling increasing deployment of more highly boosted turbocharged gasoline and diesel engines. [10] Looking ahead, the engine development opportunities of the future are many and substantial. While they present a formidable challenge to automotive engineers, they will be made possible in large part by the enormous expansion of our knowledge of engine processes that the last several decades have witnessed.

1.2 ENGINE CLASSIFICATIONS

There are many different types of internal combustion engines. They can be classified by: [11]

1. *Application.* Automobile, truck, bus, locomotive, light aircraft, marine, portable power system, power generation
2. *Basic engine configuration.* Reciprocating engines (in turn subdivided by arrangement of cylinders: e.g., in-line, V, radial, opposed), rotary engines (Wankel and other geometries)

3. *Working cycle.* Four-stroke cycle: naturally-aspirated (admitting atmospheric air), supercharged (admitting precompressed air), and turbocharged (admitting air compressed in a compressor driven by an exhaust turbine). Two-stroke cycle: crankcase scavenged, supercharged, and turbocharged
4. *Valve or port design and location.* Four-stroke cycle: Overhead (or I-head) valves, underhead (or L-head) valves, with two, three, or four valves per cylinder, and fixed or variable valve control (timing, opening and closing points, and lift), rotary valves. Two-stroke cycle: cross-scavenged porting (inlet and exhaust ports on opposite sides of cylinder at one end), loop-scavenged porting (inlet and exhaust ports on same side of cylinder at one end), through- or uniflow-scavenged (inlet and exhaust ports or valves at different ends of cylinder)
5. *Fuel.* Gasoline (or petrol), fuel oil (or diesel fuel), natural gas, liquid petroleum gas (LPG), alcohols (methanol, ethanol), hydrogen, dual fuel
6. *Method of mixture preparation.* Carburetion or single-point fuel injection upstream of the throttle, fuel injection into the intake ports, fuel injection directly into the engine cylinder
7. *Method of ignition.* Spark ignition in engines where the in-cylinder fuel-air mixture is uniform and in stratified-charge engines where the mixture is nonuniform; compression ignition locally of the evolving in-cylinder fuel-air mixture in diesel engines, as well as ignition in natural gas engines by pilot injection of fuel oil)
8. *Combustion chamber design.* Open chamber (many designs: e.g., disc, wedge, hemisphere, pent-roof, bowl-in-piston), divided chamber (small and large auxiliary chambers; many designs: e.g., swirl chambers, prechambers)
9. *Method of load control.* Varying fuel and air flow together so mixture composition is essentially unchanged, control of fuel flow alone, a combination of these
10. *Method of cooling.* Water cooled, air cooled, uncooled (other than by natural convection and radiation)

All these distinctions are important and they illustrate the breadth of engine designs available. Because this book approaches the operating and emissions characteristics of internal combustion engines from a fundamental point of view, method of ignition has been selected as the primary classifying feature. From the method of ignition—SI or CI [c]—follow the important characteristics of the fuel used, method of mixture preparation, method of load control, combustion chamber design, details of the combustion process, engine emissions, and operating characteristics. Some of the other classifications are used as subcategories within this basic classification. The engine operating cycle—four-stroke or two-stroke is next in importance; the principles of these two cycles are described in the following section.

1.3 ENGINE OPERATING CYCLES

Most of this book is about *reciprocating engines*, where each piston moves back and forth in a cylinder and transmits power from the high-pressure and temperature burned gases inside the cylinder through the piston and the connecting rod and crank mechanism to the drive shaft as shown in Fig. 1.1. The rotation of the crank produces a cyclical piston motion. The piston comes to rest at the top-center (TC) crank position and bottom-center (BC) crank position when the cylinder volume is a minimum or maximum, respectively. [d] The minimum cylinder volume is called the clearance volume V_c. The volume swept out by the piston, the difference between the maximum or total volume V_t and the clearance volume, is called the displaced or swept volume V_d. The ratio of maximum volume to minimum volume is the compression ratio r_c. Values of r_c are 8 to 12 for SI engines and typically in the ranges of 14 to 22 for CI engines.

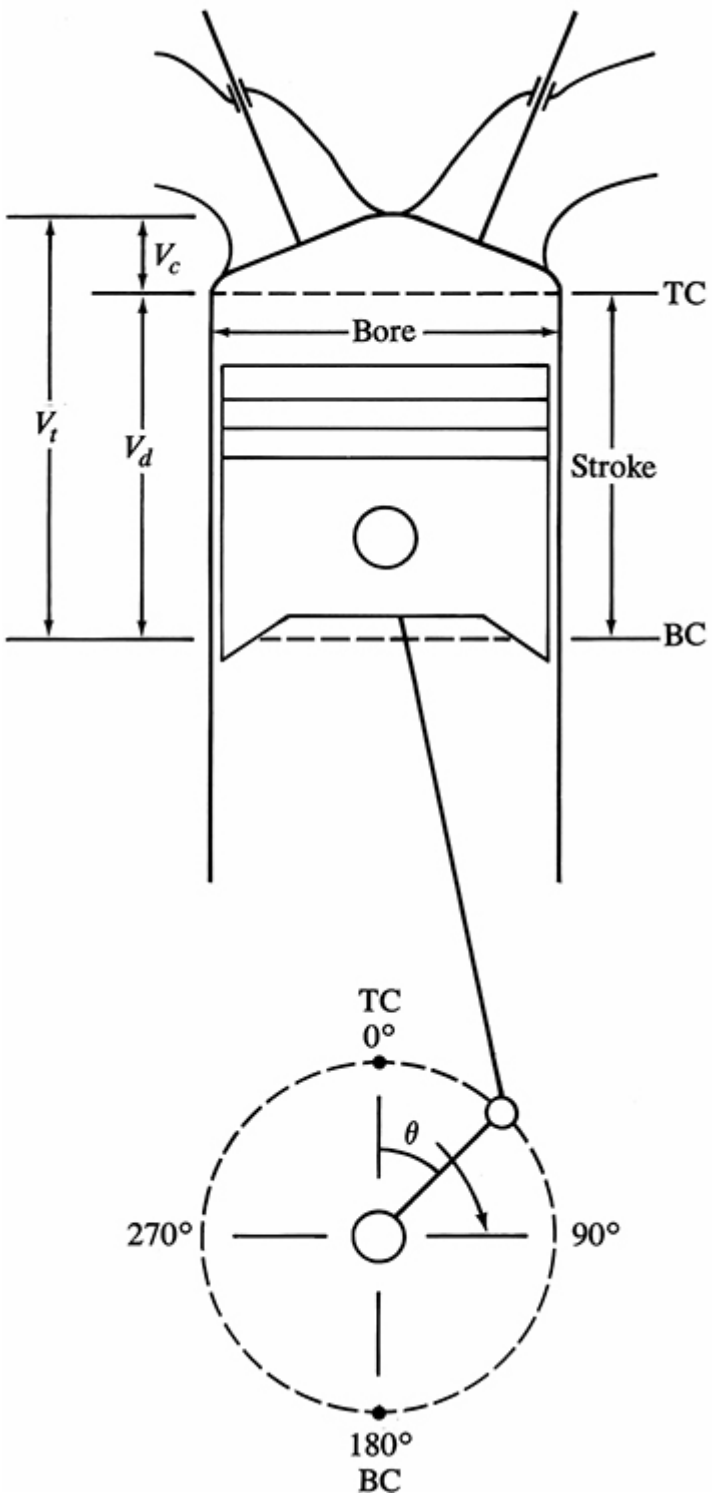


Figure 1.1 Basic geometry of the reciprocating internal combustion engine. V_c, V_d, and V_t indicate clearance, displaced, and total cylinder volumes.

The majority of reciprocating engines operate on what is known as the *four-stroke cycle*. Each cylinder requires four strokes of its piston—two revolutions of the crankshaft—to complete the sequence of events that produces one power stroke. Both SI and CI engines use this cycle that comprises (Fig. 1.2):

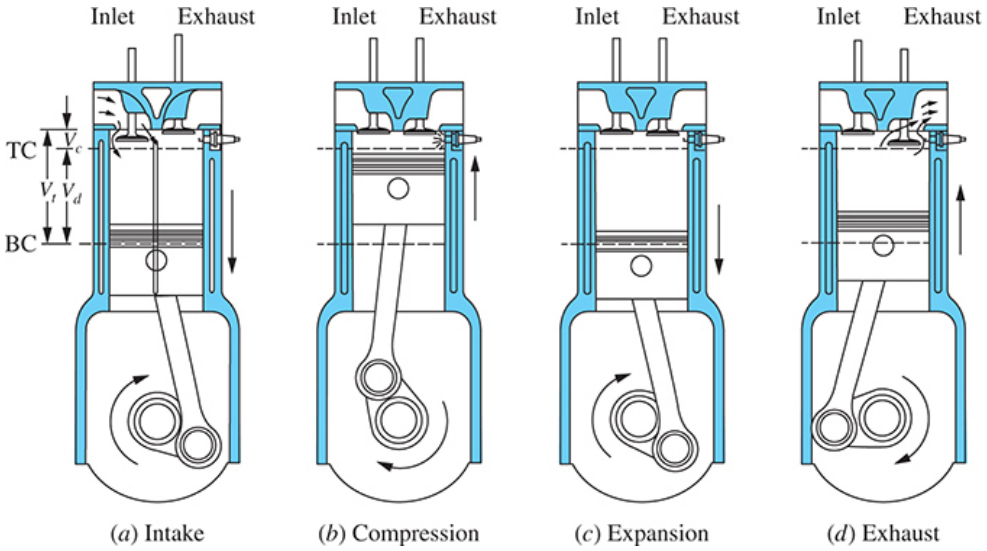


Figure 1.2 The four-stroke operating cycle.[12]

1. *An intake stroke*, which starts with the piston at TC and ends with the piston at BC, which draws fresh air or fuel-air mixture into the cylinder. To increase the mass inducted, the inlet valve opens shortly before the stroke starts and closes after it ends.
2. *A compression stroke*, which starts with the piston at BC and ends at TC, when the mixture inside the cylinder is compressed to a small fraction of its initial volume. Toward the end of the compression stroke, combustion is initiated and the cylinder pressure rises more rapidly.
3. *A power stroke*, or expansion stroke, which starts with the piston at TC and ends at BC as the high-temperature, high-pressure gases push the piston down and force the crank to rotate. About five times as much work is done on the piston during the power stroke as the piston had to do during compression. As the piston approaches BC, the exhaust valve opens to initiate the exhaust process and drop the cylinder pressure to close to the exhaust system pressure.

4. *An exhaust stroke*, where, as the piston moves from BC to TC, the remaining burned gases exit the cylinder: first, because the cylinder pressure may be significantly higher than the exhaust pressure; then as these gases are swept out by the piston as it moves toward TC. As the piston approaches TC the inlet valve opens and just after TC the exhaust valve closes. The cycle then starts again.

Though often called the Otto cycle after its inventor, Nicolaus Otto, who built the first engine operating on these principles in 1876, the more descriptive four-stroke nomenclature is preferred.

The four-stroke cycle requires, for each engine cylinder, two crankshaft revolutions for each power stroke. To obtain a higher power output from a given engine size, and a simpler valve design, the *two-stroke* cycle was developed. The two-stroke cycle is applicable to both SI and CI engines.

Figure 1.3 shows one of the simplest types of two-stroke engine designs. Ports in the cylinder liner, opened and closed by the piston motion, control the exhaust flow out of the cylinder and the fresh charge flow into the cylinder, while the piston is close to BC. The two strokes are:

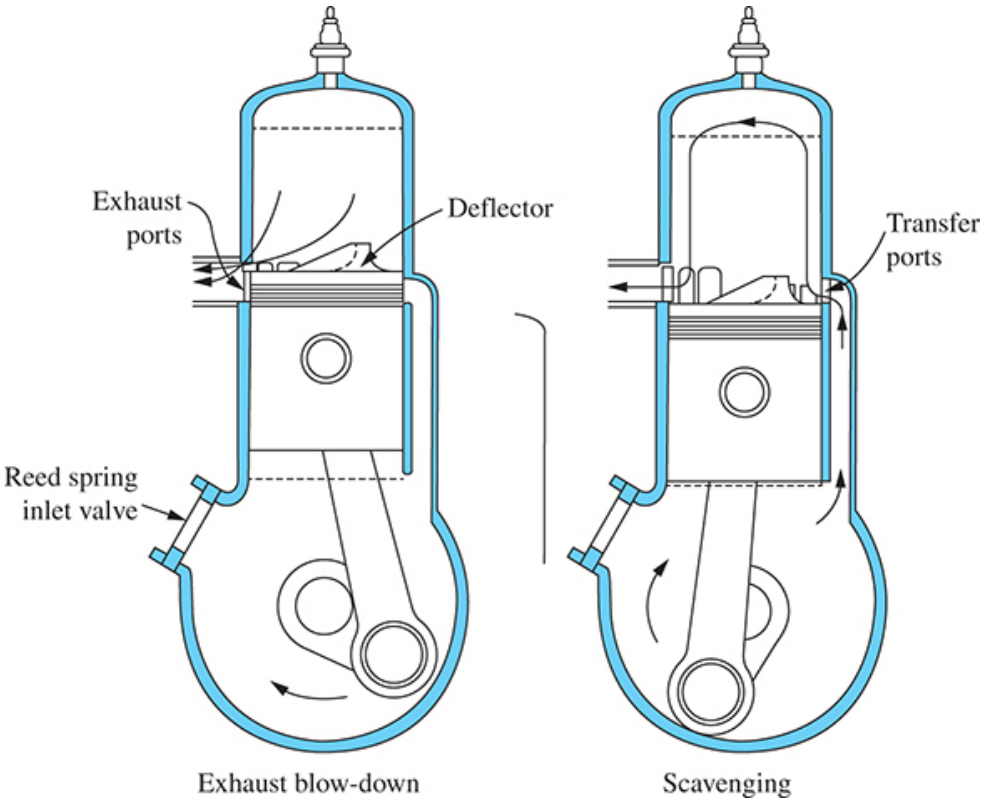


Figure 1.3 The two-stroke operating cycle. A crankcase-scavenged engine is shown. [12]

1. *A compression stroke*, which starts with the closing of the fresh charge transfer ports and then the exhaust ports, and compresses the cylinder contents as the piston moves up the cylinder, and also draws fresh charge into the crankcase through the inlet Reed valve. As the piston approaches TC, combustion is initiated.
2. *A power or expansion stroke*, similar to that in the four-stroke cycle until the piston approaches BC, when first the exhaust ports and then the transfer ports are uncovered (Fig. 1.3). Most of the burnt gases exit the cylinder in an exhaust blowdown process. When the transfer ports are uncovered, the fresh charge that has been compressed in the crankcase flows into the cylinder. The piston and the ports are generally shaped to deflect the incoming charge from flowing directly into the exhaust ports, and to achieve effective scavenging of the residual in-cylinder burned gases by this fresh charge.

Each engine cycle with one power stroke is completed in one crankshaft revolution. However, it is difficult to fill completely the displaced volume with fresh charge, and some of the fresh mixture flows directly out of the cylinder during the scavenging process. [e] The example shown is a *cross-scavenged* design; other approaches use *loop-scavenging* or *uniflow* gas exchange processes (see Sec. 6.6).

1.4 ENGINE COMPONENTS

Cutaway drawings of a four-stroke spark-ignition (SI) engine and a diesel (CI) engine are shown in Figs. 1.4 and 1.5, respectively. The SI engine is a four-cylinder in-line automobile engine. The major components are labeled. The diesel is a six-cylinder in-line heavy-duty truck engine. The function of the major components of these engines and their construction materials will now be reviewed.

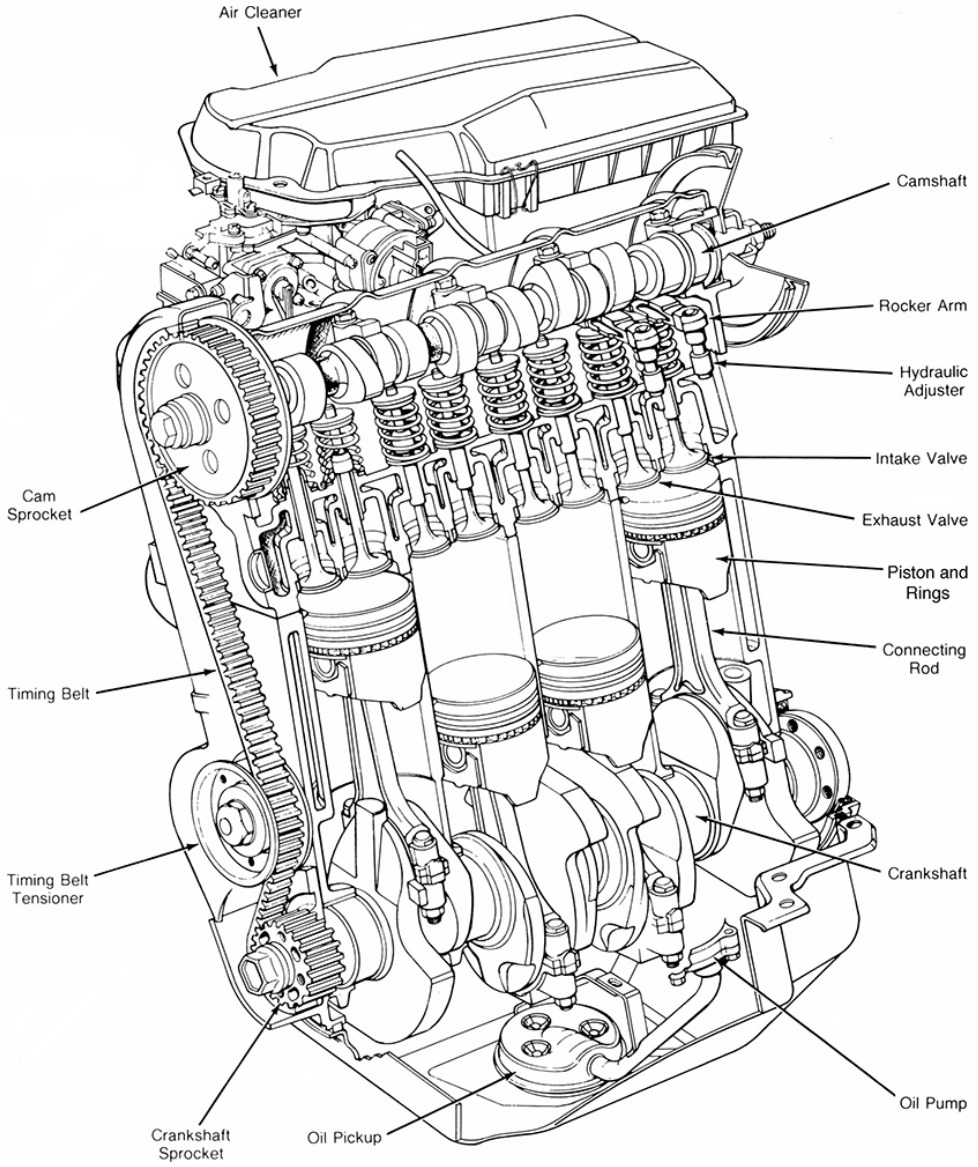


Figure 1.4 Cutaway drawing of 2.2-liter displacement four-cylinder spark-ignition engine. Bore 87.5 mm, stroke 92 mm, compression ratio 8.9.

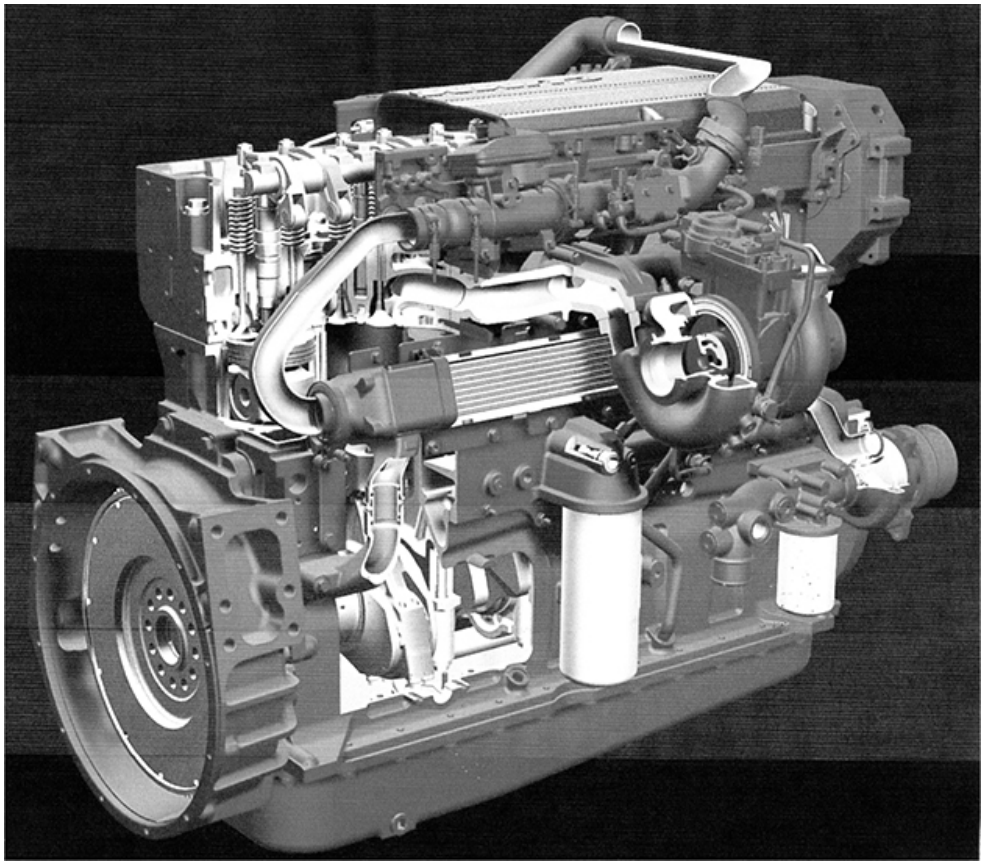


Figure 1.5 Cross-section drawing of a four-stroke cycle 6.7-liter in-line six-cylinder turbocharged diesel engine. Bore 107 mm, stroke 124 mm, compression ratio 17.3, maximum torque 1200 N · m at 1600 rev/min, maximum power 285 kW at 2800 rev/min. (*Courtesy Cummins Engines.*)

The engine cylinders are contained in the engine block. The block has traditionally been made of gray cast iron because of its good wear resistance and low cost, but is often now made of aluminum. Passages for the cooling water are cast into the block. Heavy-duty and truck engines often use removable cylinder sleeves pressed into the block that can be replaced when worn. These are called *wet liners* or *dry liners* depending on whether the sleeve is in direct contact with the cooling water. Aluminum is used in automotive SI engine blocks to reduce engine weight. Iron cylinder liners may be inserted at the casting stage, or later on in the machining and assembly process. The crankcase is often integral with the cylinder block.

The crankshaft has traditionally been a steel forging; nodular cast iron crankshafts are also accepted practice in automotive engines. The crankshaft

is supported in main bearings. The number of crankshaft bearings depends largely on the engine's loading and maximum speed. The maximum number of main bearings is one more than the number of cylinders; there may be less. The crank has eccentric portions (crank throws); the connecting rod big-end bearings attach to the crank pin on each throw. Both main and connecting rod bearings use steel-backed precision inserts with bronze, babbit, or aluminum as the bearing materials. The crankcase is sealed at the bottom with a pressed-steel or cast aluminum oil pan, which acts as an oil reservoir for the lubricating system.

Pistons are made of aluminum in smaller engines or cast iron in larger slower-speed engines. The piston both seals the cylinder and transmits the combustion-generated gas pressure to the crank pin via the connecting rod. The connecting rod, usually a steel or alloy forging (though sometimes aluminum), is fastened to the piston by means of a steel piston pin through the rod upper end. The piston pin is usually hollow to reduce its weight.

The oscillating motion of the connecting rod exerts an oscillating force on the cylinder walls via the piston skirt (the region below the piston rings). The piston skirt is usually shaped to provide appropriate thrust surfaces. The piston is fitted with rings that ride in grooves cut in the piston head to seal against gas leakage and control oil flow. The upper ring is the compression ring that is forced outward against the cylinder wall and downward onto the groove face. The lower rings scrape the surplus oil from the cylinder wall to reduce exposure to the hot burned gases, and return it to the crankcase. The crankcase must be ventilated to remove gases that blow by the piston rings, to prevent pressure buildup. The crankcase gases are recycled to the engine intake.

The cylinder head (or heads in V engines) seals off the cylinders and is made of aluminum or cast iron. It must be strong and rigid to distribute the gas forces acting on the head as uniformly as possible through the engine block. The cylinder head contains the spark plug (for an SI engine) or fuel injector (for a CI or direct-injection engine), and, in overhead valve engines, parts of the valve mechanism.

The valves shown in Fig. 1.4 are poppet valves, the valve type normally used in four-stroke engines. The engine shown has one intake and one exhaust valve: most modern engines have four valves per cylinder (two intake and two exhaust valves), or three valves (two intake and one exhaust). Valves are made from forged alloy steel; the cooling of the exhaust valve, which

operates at up to about 700°C, may be enhanced by using a hollow stem partially filled with sodium, which through evaporation and condensation carries heat from the hot valve head to the cooler stem. Most modern SI engines have overhead valve locations (sometimes called valve-in-head or I-head configurations) as shown in Fig. 1.4. This geometry leads to a compact combustion chamber with minimum heat losses and flame travel time, and improves the breathing capacity. Older geometries such as the L head where valves are to one side of the cylinder are now only used in small low-cost engines.

The valve stem moves in a valve guide, which can be an integral part of the cylinder head (or engine block for L-head engines), or may be a separate unit pressed into the head (or block). The valve seats may be cut in the head or block metal (if cast iron) or hard steel inserts may be pressed into the head or block. A valve spring, attached to the valve stem with a spring washer and split keeper, holds the valve closed. A valve rotator turns the valves a few degrees on opening to wipe the valve seat, avoid local hot spots, and prevent deposits building up in the valve guide.

A camshaft made of cast iron or forged steel with one cam per valve (or pair of valves in four valves per cylinder engines) is used to open and close the valves. The cam surfaces are hardened to obtain adequate life. In four-stroke cycle engines, camshafts turn at one-half the crankshaft speed. Mechanical or hydraulic lifters or tappets slide in the block and ride or roll on the cam. Depending on valve and camshaft location, additional members are required to transmit the tappet motion to the valve stem; for example, in in-head valve engines with the camshaft at the side, a push rod and rocker arm are used. A trend in high-speed automotive engines is to mount the camshaft over the head with the cams acting either directly or through a pivoted follower on the valve. Also, variable control of valve opening and closing as a function of engine operating conditions, in its simplest form using a camshaft phasing device, is replacing fixed valve timing engine designs. Camshafts are gear, belt, or chain driven from the crankshaft.

An intake manifold (aluminum, cast iron, or plastic) and an exhaust manifold (generally of cast iron) complete the SI engine assembly. Other engine components specific to SI engines—fuel injectors, ignition systems—are described in more detail in the remaining sections in this chapter.

The four-stroke cycle diesel engine shown in Fig. 1.5 is an in-line six-cylinder design commonly used for large trucks. The engine is turbocharged

to increase the amount of air that enters the cylinder each cycle. The turbocharger consists of a centrifugal compressor (which compresses the air prior to entry into the cylinder on the same shaft as the exhaust-gas-driven turbine that powers the compressor). In diesel engines, the fuel injectors are mounted in the cylinder head. Diesel fuel-injection systems are discussed in more detail in Sec. 1.8 .

1.5 MULTICYLINDER ENGINES

Small engines are used in many applications: for example, lawn mowers, chain saws, in portable power generation, as outboard motorboat engines, and in motorcycles. These are often single-cylinder engines. In the above applications, simplicity and low cost in relation to the power generated are the most important characteristics; fuel consumption, engine vibration, high power to weight or volume ratio, and engine durability are usually less important. A single-cylinder engine gives only one power stroke per revolution (two-stroke cycle) or two revolutions (four-stroke cycle). Hence, the individual cycle torque pulses are widely spaced, and engine vibration and smoothness are significant issues.

Multicylinder configurations are invariably used in practice in all but the smallest engines. As rated power increases, the advantages of smaller cylinders in regard to bulk size, weight, improved engine performance, and engine balance and smoothness all point toward increasing the number of cylinders so the engine's total displaced volume is spread out amongst several smaller cylinders. The increased frequency of power strokes with smaller and increasing number of cylinders produces more frequent and smaller torque pulses, and thus smoother output. The forces in each component are smaller, so structural design requirements are reduced. Multicylinder engines can also achieve a much better state of balance than single-cylinder engines. A force must be applied to each piston to accelerate it during the first half of its travel from BC or TC. The piston then exerts a force on the crankshaft as it decelerates during the second part of the stroke. It is desirable to cancel these inertia forces through the choice of number and arrangement of cylinders to achieve a *primary* balance. Note, however, that the motion of the piston is more rapid during the upper half of its stroke than during the lower half (a consequence of the connecting rod and crank

mechanism evident from Fig. 1.1; see also Sec. 2.2). The resulting inequality in acceleration and deceleration of pairs of pistons (one moving up and one moving down) produces corresponding differences in inertia forces generated. Certain combinations of cylinder number and arrangement balance out these *secondary* inertia force effects.

For a given engine displaced volume, the larger the number of cylinders, the higher the engine's maximum power. The reciprocating speed of an engine's pistons is limited by the airflow into each cylinder. Once the flow through the intake valve becomes sonic—reaches the speed of sound—higher piston speeds do not increase airflow. For a given engine displacement, increasing the number of cylinders, and thus reducing their size, raises the crankshaft rotational speed at which this sonic airflow limit is reached. Since engine power is proportional to the engine's rotational (crankshaft) speed, maximum performance is improved.

Other operational issues are affected by cylinder size. The relative importance of heat losses from the in-cylinder gases depends on the relative importance of the combustion chamber surface area to its volume. The SI engine compression-ratio limiting phenomenon called knock is adversely affected by the flame travel distance (spark plug gap to farthest combustion chamber wall).

Common four-stroke multicylinder configurations are shown in Fig. 1.6. [13] These multicylinder configurations normally use equal crankshaft rotation firing intervals between cylinders. In *in-line engines*, the cylinders are arranged in a single plane. Three-, four-, five-, and six-cylinder in-line configurations are used. Four-cylinder in-line engines are the most common arrangement for automobile engines from 1.2 to about 2.5-liter displacement. An example of this in-line arrangement is shown in Fig. 1.4. It is compact—an important consideration for small passenger cars. It provides two torque pulses per revolution of the crankshaft, and primary inertia forces (though not secondary forces) are balanced. Six-cylinder in-line diesel engines are commonly used in the truck market with up to 12-liter displacement.

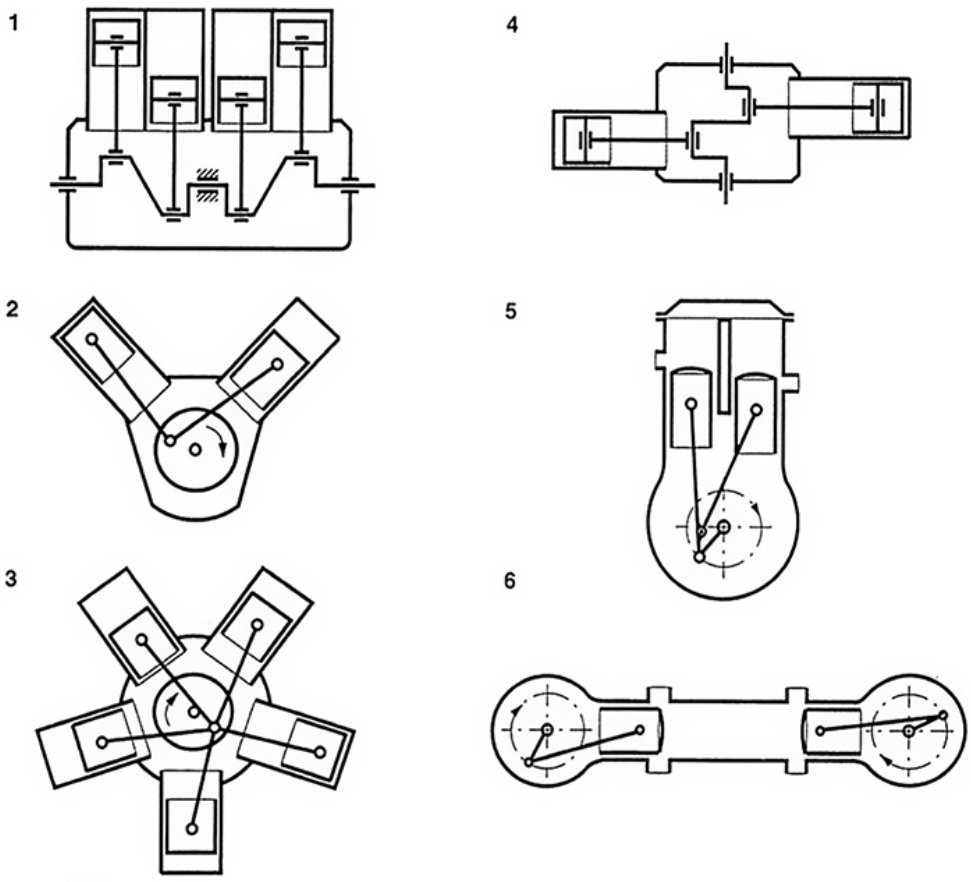


Figure 1.6 Multicylinder engine configurations: (1) In-line engine; (2) V-engine; (3) Radial engine; (4) Opposed-cylinder engine; (5) U-engine; (6) Opposed-piston engine.[13] (*Courtesy Robert Bosch GmbH and SAE.*)

The vee (V) arrangement, with two banks of cylinders set at an angle to each other, provides a compact engine block and is used extensively for larger displacement automotive engines. Vee six, eight, ten, and twelve configurations are used. In a V-6 engine, the six cylinders are arranged in two banks of three, usually with a 60° angle between their axis. Six cylinders are normally used in gasoline SI engines in the 2.4- to 3.6-liter displacement range. Six-cylinder engines provide smoother operation with three torque pulses per revolution. The in-line arrangement is fully balanced. However, it gives rise to crankshaft torsional vibration, and also makes even distribution of air to each cylinder more difficult. The V-6 arrangement is more compact than an in-line 6, and provides primary balance of the reciprocating components. With the V-engine, however, a rocking moment is imposed on the crankshaft due to the secondary inertia forces, which results in the engine

being less well balanced than the in-line version. The V-8 arrangement, in sizes between 3.2 and 6 or more liters, is commonly used to provide compact, smooth, low-vibration, larger-displacement, SI engines, as are V-10 and V-12 designs.

The radial engine configuration, with cylinders arranged in one or more radial planes, as shown, was common in larger piston-driven aircraft engines. Opposed cylinder engine designs are occasionally used. As Fig. 1.6 indicates, the motion of pairs of pistons with this design is fully balanced. The U-cylinder configuration, where the pistons move in the same direction, and the opposed-piston configuration have been used in special purpose two-stroke engine concepts.

1.6 SPARK-IGNITION ENGINE OPERATION

In SI engines, the fuel must be vaporized and well mixed with the air inducted into the cylinder, prior to combustion. Historically, the fuel flow was metered with a carburetor or single-point fuel-injection system (Fig. 1.7*a*) upstream of the throttle, which controls the airflow. This approach has been superceded by intake-port fuel injection (Fig. 1.7 *b*) where a pulsed liquid fuel spray is directed toward the intake valve. Injection of gasoline directly into each cylinder (Fig. 1.7 *c*) is now in large-scale production. Moving the point of fuel injection closer to the cylinder enables better dynamic response during engine transients.

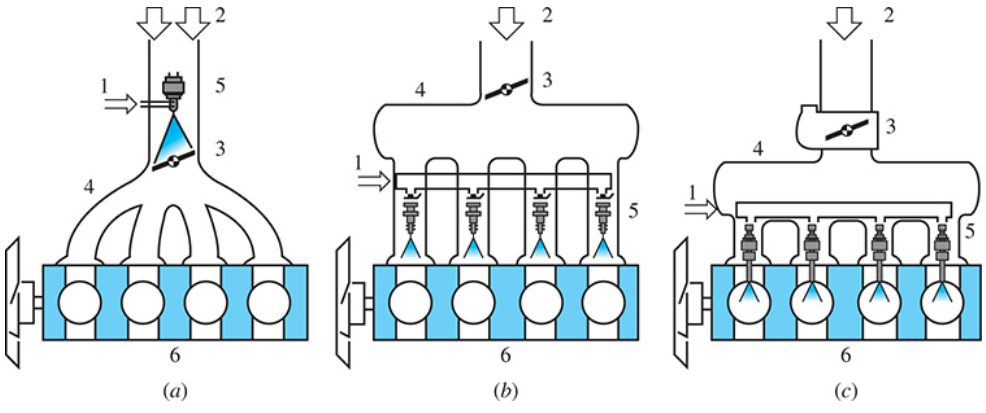


Figure 1.7 Different fuel-injection approaches for gasoline spark-ignition engines. (*a*) Single-point injection; (*b*) Multipoint port injection; (*c*) Direct

in-cylinder injection. (1) Fuel supply; (2) Air supply; (3) Throttle valve; (4) Intake manifold; (5) Injectors; (6) Engine. [13] (*Courtesy Robert Bosch GmbH and SAE*.)

Figure 1.8 shows the layout of a modern SI engine management system. The airflow, fuel flow, exhaust gas characteristics, and engine operating state are all monitored and controlled as shown to provide the desired engine performance with good combustion characteristics, high efficiency, and low exhaust air pollutant emissions. The ratio of mass flow of air to mass flow of fuel must be held approximately constant at about 15 to ensure reliable combustion and facilitate exhaust emissions control. The appropriate fuel flow is determined for the engine airflow in the following manner. The airflow into the intake system is measured with an air mass-flow meter. A throttle valve or plate, which can be opened or closed, controls the airflow. The appropriate amount of fuel required per cylinder per cycle to generate the desired engine output is then determined by the engine control unit. In naturally-aspirated engines, the intake airflow is reduced by throttling to below atmospheric pressure by reducing the flow area when the power required (at any engine speed) is below the maximum, which is obtained when the throttle is wide open.

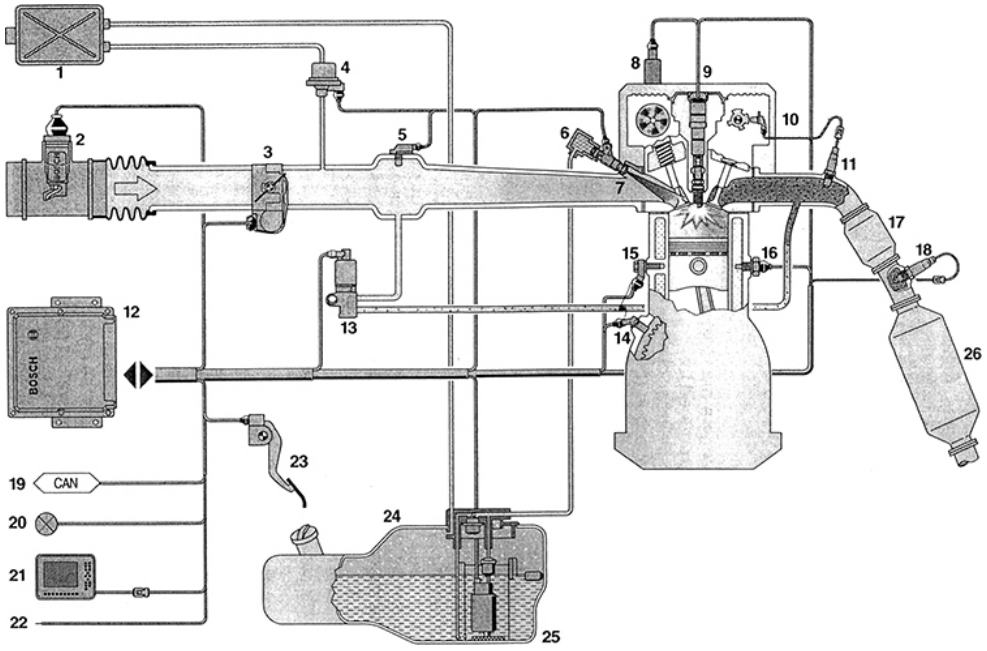


Figure 1.8 Schematic of modern port-injected engine management system (Bosch ME-Motronic system). (1) Carbon fuel-vapor absorbing canister; (2) Hot-film air-mass meter with integrated temperature sensor; (3) Throttle device; (4) Canister-purge valve; (5) Intake-manifold pressure sensor; (6) Fuel-distribution pipe; (7) Injector; (8) Actuators and sensors for variable valve timing; (9) Ignition coil with attached spark plug; (10) Camshaft phase sensor; (11) Lambda oxygen sensor upstream of primary catalytic converter; (12) Engine control unit; (13) Exhaust-gas recirculation valve; (14) Speed sensor; (15) Knock sensor; (16) Engine-temperature sensor; (17) Primary three-way catalytic converter; (18) Lambda oxygen sensor downstream of primary catalytic converter: (19) CAN interface; (20) Fault lamp; (21) Diagnosis interface; (22) Interface to immobilizer control unit; (23) Accelerator-pedal module with pedal-travel sensor; (24) Fuel tank; (25) In-tank unit with electric fuel pump, fuel filter, and fuel-pressure regulator; (26) Main three-way catalytic converter. [13] (*Courtesy Robert Bosch GmbH and SAE*.)

The sequence of events that take place inside the engine cylinder is illustrated in Fig. 1.9. Several variables are plotted against crank angle through the entire four-stroke SI engine cycle. Crank angle is a useful independent variable because the various engine processes occupy almost constant crank angle intervals over a wide range of engine-operating conditions. The figure shows the valve opening and closing angles, and volume relationship, for a typical fixed valve-timing automotive SI engine. To maintain high mixture flows at high engine speeds (and thus high power outputs) the inlet valve, which opens before TC, closes substantially after BC. During intake, the inducted fuel and air mix in the cylinder with the *residual* burned gases remaining from the previous cycle. After the intake valve closes, the cylinder contents are compressed to above atmospheric pressure and temperature as the cylinder volume is reduced. Some heat transfer between the in-cylinder gases and the piston, cylinder head, and cylinder walls occur—first a heating of the gases, then a cooling, but the effect on unburned gas properties is modest.

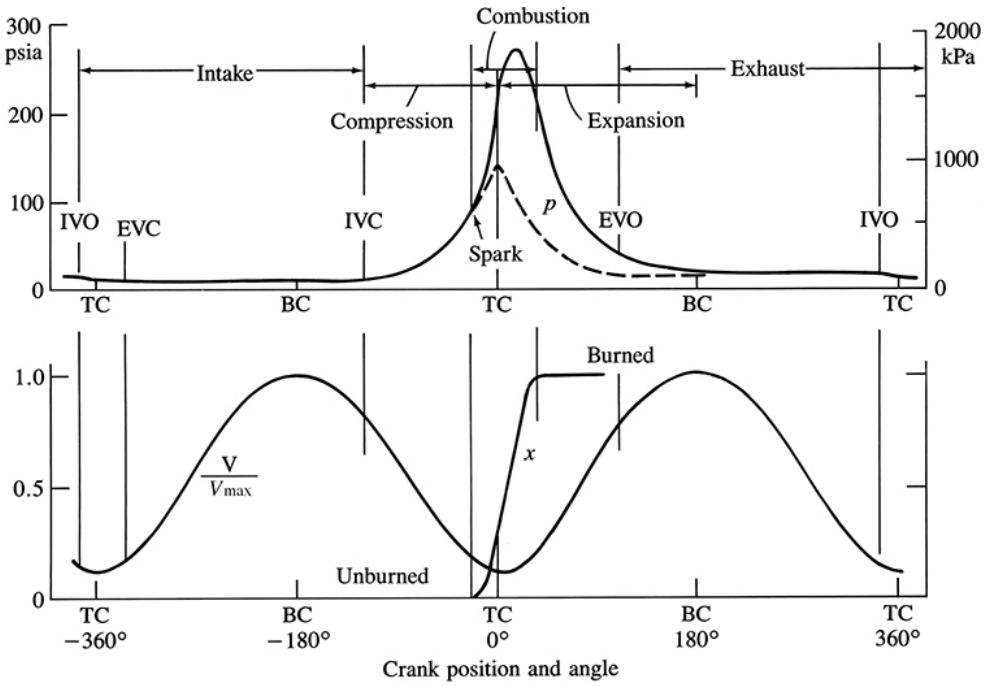


Figure 1.9 Sequence of events in four-stroke spark-ignition engine-operating cycle. Cylinder pressure p (solid line, firing cycle; dashed line, motored cycle), cylinder volume V/V_{max}, and mass fraction burned x_b are plotted against crank angle.

Between about 10 and 40 crank angle degrees before TC, an electrical discharge across the spark plug starts the combustion process. Before the desired ignition point, the ignition driver switches a current to the primary circuit of the ignition coil. At the ignition point, the primary winding is interrupted, generating in the secondary ignition coil winding that is connected to the spark plug, a high voltage across the plug electrodes as the magnetic field collapses. This switching is done electronically. A flame develops from the spark discharge, propagates through the mixture of air, fuel, and residual gas in the cylinder, and extinguishes at the combustion chamber walls. The duration of this burning process varies with engine design and operation, but is typically 40 to 60 crank angle degrees, as shown in Fig. 1.9. As fuel-air mixture burns in the flame, the cylinder pressure (solid line in Fig. 1.9) rises above the level due to compression alone (dashed line). This latter curve—called the *motored* cylinder pressure—is the pressure trace obtained from a motored or nonfiring engine.[f] Note that due to differences in the flow pattern and mixture composition between

cylinders and within each cylinder, cycle-by-cycle, the development of each combustion process differs somewhat. As a result, the shape of the pressure versus crank angle curve in each cylinder, and cycle-by-cycle, is not exactly the same.

There is an optimum spark timing which, for a given mass of fuel, air, and residual inside the cylinder, gives maximum torque. More advanced (earlier) timing or retarded (later) timing than this optimum gives lower output. Called *maximum brake-torque* (MBT) timing, [g] this optimum timing is an empirical compromise between starting combustion too early in the compression stroke (when the work transfer is *to* the cylinder gases) and completing combustion too late in the expansion stroke (and so lowering peak expansion stroke pressures).

About two-thirds of the way through the expansion stroke, the exhaust valve starts to open. The cylinder pressure is significantly higher than the exhaust manifold pressure and a *blowdown* process occurs. The burned gases flow through the valve into the exhaust port and manifold until the cylinder pressure and exhaust pressure equilibrate. The duration of this process depends on the pressure level in the cylinder. The piston then *displaces* most of the remaining burned gases from the cylinder into the manifold during the exhaust stroke. The exhaust valve opens before the end of the expansion stroke to ensure that the blowdown process does not last too far into the exhaust stroke when the piston travels upwards. The actual timing is a compromise that balances reduced work transfer *to the piston* before BC against reduced work transfer *to the cylinder contents* after BC.

The exhaust valve remains open until just after TC; the intake opens just before TC. The valves are opened and closed slowly to avoid noise and excessive cam wear. To ensure the valves are fully open when piston velocities are at their highest, the valve open periods usually overlap somewhat. If the intake flow is throttled to below exhaust manifold pressure, then backflow of burned gases from the cylinder into the intake manifold occurs when the intake valve is first opened. During the valve overlap period, backflow of burned gas from the exhaust port into the cylinder occurs.

With variable valve control, the trade-offs that fixed valve timing requires can be relaxed. The simplest approach varies intake and exhaust valve timing by rotating the camshafts to change their phasing relative to the crankshaft. More complex systems vary valve lift as well as varying the valve opening

and closing angles. Variable valve control is attractive because it improves maximum engine power (at high speed) and maximum torque at lower speeds, and can improve part-load engine efficiency. It can also be used to control the mass of burned residual gas, and fresh air, trapped in the engine cylinder.

1.7 DIFFERENT TYPES OF FOUR-STROKE SI ENGINES

A variety of SI engines are used in practice, depending on the application. Small SI engines are used in many applications: in the home (e.g., lawn mowers, chain saws), in portable power generation, as outboard motorboat engines, and in motorcycles. These are often single-cylinder engines producing a few kW of power. In the above applications, light weight, small bulk, and low cost in relation to the power generated are the most important characteristics; fuel consumption, engine vibration, and engine durability are less important. A single-cylinder engine gives only one power stroke per crank revolution (two-stroke cycle) or two revolutions (four-stroke cycle). Hence, the torque pulses are widely spaced, and engine vibration and smoothness are significant problems.

Multicylinder engines are invariably used in automotive practice. As rated power increases, the advantages of smaller cylinders in regard to size, weight, power density, improved engine balance, and smoothness point toward increasing the number of cylinders per engine: see Sec. 1.5 . Multicylinder SI engines (in the power range 25 to 400 kW) are used in cars, light trucks, vans, light-duty commercial vehicles, and in stationary applications to produce mechanical and electrical power. Many of these markets are shared with diesel (CI) engines. Low engine emissions and high operating efficiency are important, especially in these transportation applications. Precise control of fuel and airflow is critical to achieving these objectives. What used to be the dominant fuel metering device—the carburetor—has been superceded by electrically controlled fuel injection into each intake port. Now, injection of the gasoline directly into each engine cylinder is coming into production (Fig. 1.7). Each of these technology steps improves control of the amount of fuel entering each cylinder per cycle, and

thus the dynamic response of the engine to changes in load (engine output) and speed.

The work transfer per cycle to each piston depends on the amount of fuel burned per cylinder per cycle, which depends on the amount of fresh air inducted each cycle. Variable valve control over the engine's speed range can be used to increase the mass of air inducted into each cylinder in four-stroke SI engines (especially at low and high speed), and thus increase the wide-open-throttle torque and power. Engine output from a given displacement engine can be increased by boosting—increasing the density of the air supplied to the engine intake by compressing atmospheric air. Thus, compressing the air prior to entry into the cylinder with a supercharger or a turbocharger increases the output from a given displacement engine.

Examples of various types of SI engines in practical use follow to provide the context for reviewing critical engine processes, a primary objective of this text.

1.7.1 Spark-Ignition Engines with Port Fuel Injection

Spark-ignition (SI) engines have traditionally been operated with a premixed fuel vapor/air mixture inside the cylinder, prepared by feeding liquid gasoline into the engine intake. Carburetors were used to meter the fuel flow in proportion to the airflow. This technology has largely been replaced by port fuel injection (see Fig. 1.8) where an injector in each cylinder's intake port or manifold injects a pulsed fuel spray toward the intake valve, once per cycle. The hot valve surface and warm intake port (once the engine has warmed-up) promote rapid evaporation of the liquid fuel, and the airflow through the port(s) past the intake valve(s) and into the cylinder, coupled with the in-cylinder flow and mixing with the hot residual gas, produces a nearly homogeneous mixture by the time combustion starts. Here, we show some examples of SI engines with this method of mixture preparation.

Figure 1.10 shows a small single-cylinder air-cooled SI engine with a displaced volume of 149 cm^3 and power output of 2.8 kW (3.9 hp). The objective of such simple construction SI engines is to produce modest power levels at low cost. A primary benefit of air-cooled, as compared to the water-cooled, engines is lower engine weight. The fins on the cylinder block

and head are necessary to increase the external heat-transfer surface area to achieve the required heat rejection. In small engines, such as in Fig. 1.10, natural convection promotes adequate airflow around the outside of the engine. In larger engines, an air blower provides forced air convection over the block. The blower is driven off the driveshaft.

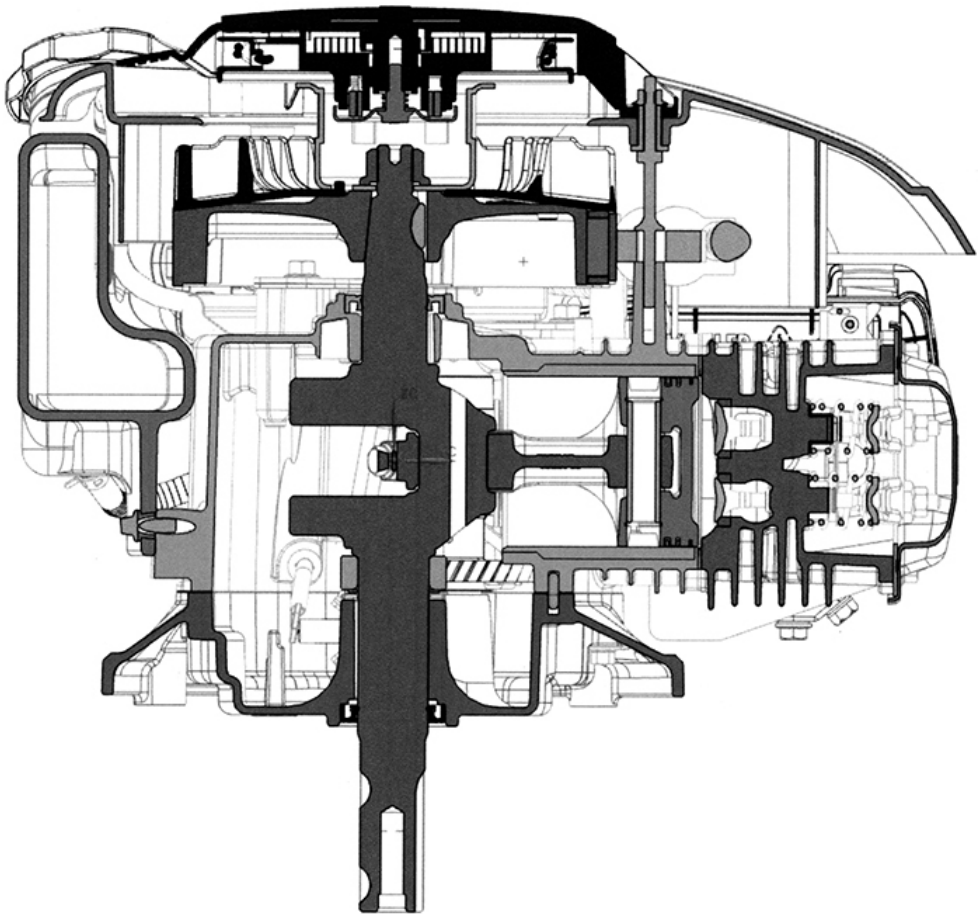


Figure 1.10 Cutaway drawing of single-cylinder air-cooled spark-ignition engine. Displacement 149 cm^3, bore 65 mm, stroke 45 mm, compression ratio 9.2, maximum power 2.8 kW at 3000 rev/min. (*Courtesy Kohler Co.*)

Figure 1.11 shows a turbocharged automobile engine that incorporates many of the features now used to improve engine performance and efficiency. The in-line arrangement with four cylinders provides a compact block, and when turbocharged, increases the power per unit engine displaced volume significantly. This engine features all-aluminum construction, four valves per cylinder, dual overhead camshafts, friction reducing roller finger followers

in the valve train, variable phasing on each cam to control the relative phasing of intake and exhaust valves, and piston-cooling oil jets to control piston temperatures in this high-performance engine. Other performance enhancing features now being designed into such automobile engines are cylinder cut out (or displacement on demand) where, for example, in a V-8 engine, at the lighter loads, the valves in half the cylinders are deactivated so only four cylinders provide torque. This reduces the pumping work over the exhaust and intake strokes and thereby improves engine fuel consumption.

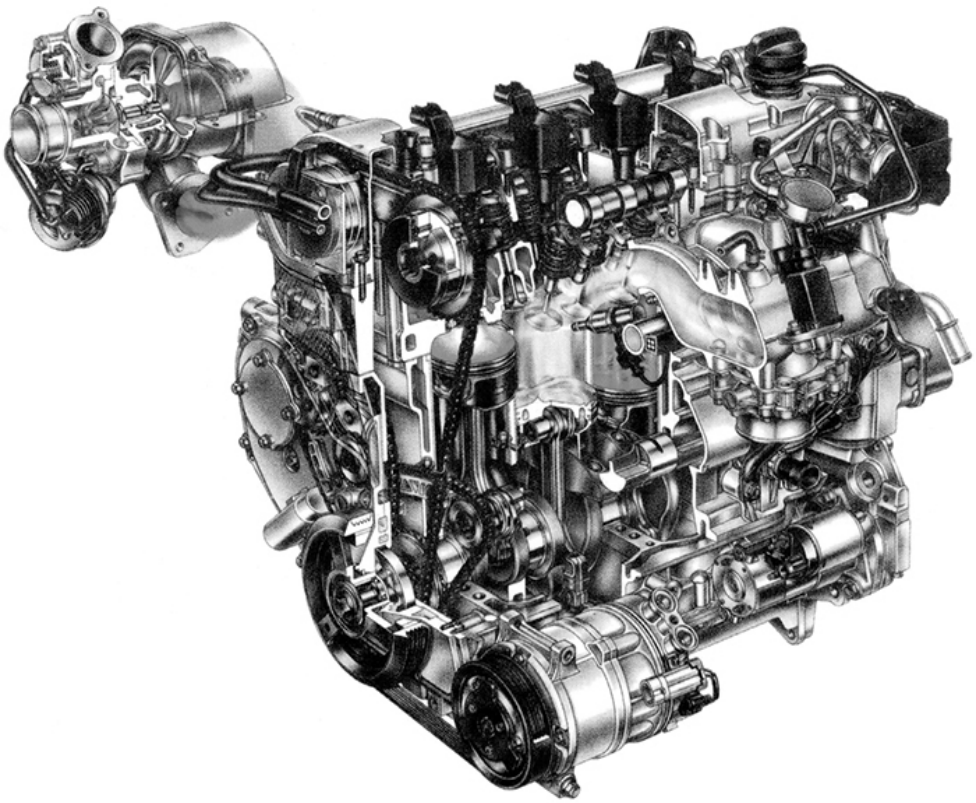


Figure 1.11 Cutaway drawing of General Motors four-cylinder turbocharged DI Ecotec gasoline spark-ignition engine. Displacement 2.0 liters, bore 86 mm, stroke 86 mm, compression ratio 9.2, maximum power 187 kW (250 hp) at 5300 rev/min, maximum torque 353 N · m (260 lb · ft) at 2000 rev/min. [14] (*Courtesy General Motors Corporation*.)

Variable valve control improves engine performance, efficiency, and emissions (see Sec. 6.3.3). The simpler systems used vary the relative phasing of the intake and exhaust valve opening and closing by rotating the

camshafts relative to the crankshaft. Valve lift profiles and open duration remain fixed. This is the approach used in the engine shown in Fig. 1.11. (The cam-phasing system is apparent upper center of the engine drawing.) More sophisticated approaches vary valve timing, lift profile, and open duration (e.g., BMW's Valvetronic system [15]). This technology can eliminate the need for throttle valves by accurately controlling the cylinder charging process by intake valve control.

1.7.2 SI Engines for Hybrid Electric Vehicles

The use of internal combustion engines in automotive hybrid propulsion systems is prompting additional SI engine developments. In such a hybrid system, an internal combustion engine, a generator, battery, and electric motor are combined. Figure 1.12 shows three categories of hybrid systems: a *parallel* hybrid, a *series* hybrid, and a *power split* hybrid. In the parallel approach, the engine can drive the wheels directly, the battery can drive via the electric motor, or both can be combined to drive the wheels to obtain a high overall propulsion system efficiency at all loads and speeds.

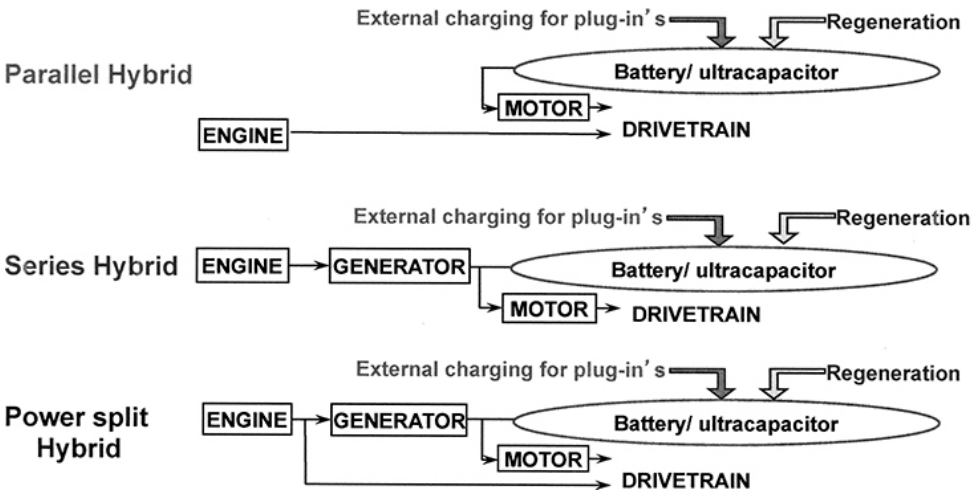


Figure 1.12 Diagrams of parallel, series, and power split hybrid electric vehicle propulsion systems.

In the series approach, the electric motor drives the vehicle's wheels. The engine can drive through the generator and motor, or recharge the battery via the generator. Since the vehicle is propelled solely by electrical energy, the

engine is not coupled to the wheels. Thus its operating conditions are not dependent on the vehicle's operation so it can be operated in its higher efficiency modes. In the power-split system, a planetary gear set is used to transmit power from the engine. This arrangement allows both parallel and series-type operation to be combined. Power from the engine can flow directly to the wheels via the ring of the planetary gear system. Engine power can also flow through the generator, producing electrical power that can drive the wheels through the electric motor.

These hybrid propulsion systems provide increased vehicle drive efficiency relative to direct internal combustion engine drive for three basic reasons. First, *regenerative braking*—applying a braking torque by connecting the generator to the vehicle's wheels is then used to recharge the battery—converts a substantial fraction of the vehicle's kinetic energy as the vehicle slows down to store electrical energy. Second, when the engine is being used, it can operate much of the time at a higher efficiency than would be the case with a stand-alone engine vehicle propulsion system. Third, the battery electric drive mode allows the engine to be shut down when the vehicle is decelerating or idling.

Figure 1.13 shows an SI engine designed specifically for this application. The engine employs a modified version of the four-stroke cycle called the *Atkinson cycle*, where the volume ratio used for expansion is higher than the volume ratio for compression. The engine shown has a displaced volume of 1.5 liters, a geometric (TC to BC) compression/expansion ratio of 13:1, and uses variable valve timing with late intake valve closing during compression and late exhaust valve opening during expansion to achieve a higher effective expansion than compression. This increases engine efficiency. Maximum engine speed is held to 5000 rev/min to minimize the pumping penalties of this Atkinson cycle approach.

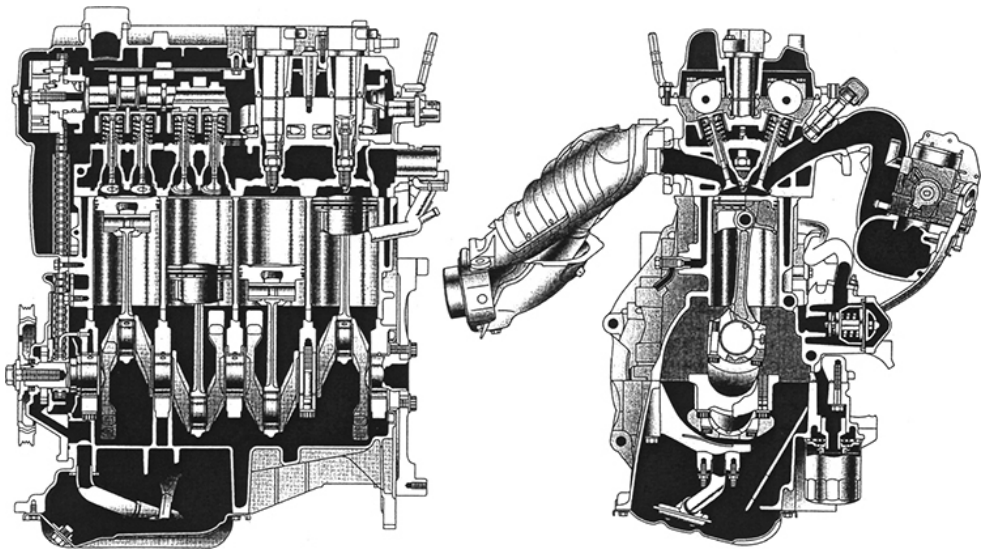


Figure 1.13 Four-cylinder Toyota spark-ignition engine designed for a hybrid electric automobile propulsion system.[16] This 1.5-liter (bore = 75 mm, stroke = 84.7 mm), four valves per cylinder, variable valve timing engine uses the Atkinson cycle with a geometric compression/expansion ratio of 13:1. Maximum power is 57 kW (76 hp) at 5000 rev/min. Valve timings are: intake opening 18 to –15° BTC, closing 72 to 105° ABC; exhaust opening 34° BBC, closing 2° ATC.

An alternative to this hybrid electric vehicle (HEV) system, which overall is powered solely by a fuel such as gasoline, is the plug-in hybrid (PHEV) system. Here a larger battery, with some 10 to 30 mile (15 to 50 km) all electric driving range rather than the electric range of a few miles of the HEV system, is used that can be recharged from the electrical grid. Thus the PHEV can be driven with electricity or with a hydrocarbon fuel similarly to an HEV. There is an important but different role for SI engines (and potentially diesels) to play as a key component of these more efficient hybrid systems: the engine preserves the driving flexibility that vehicles require, as the electrification of propulsion systems continues to evolve.

1.7.3 Boosted SI Engines

The work transfer to each piston per cycle that can be obtained from a given displacement engine determines the amount of torque the engine can deliver. This work transfer depends on the amount of fuel that can be burned in each

cylinder each cycle. This depends on the amount of fresh air that is inducted into each cylinder each cycle. Increasing the air density prior to its entry into each cylinder thus increases the maximum torque that an engine of a given displacement can deliver. This can be done with a supercharger, a compressor mechanically driven by the engine. More often it is done with a turbocharger, a compressor-turbine combination, which uses the energy available in the engine exhaust stream to provide via the turbine the power required to compress the intake air.

Figure 1.14 shows a cutaway drawing of a turbocharged automobile SI engine, which illustrates how the turbocharger connects with the engine's cylinders. The airflow passes through an air filter (1) into a centrifugal compressor (2) where the radially outward flowing air is compressed by the rotating varies. Next the air flows through an intercooler (3) to reduce the compressed air temperature (further increasing its density), through the intake manifold (4) into the intake port where the fuel is injected, past the intake valve (5), and into the cylinder (6). When the exhaust valve (7) opens, the hot and higher-than-atmospheric pressure exhaust gas flows through the valve and exhaust manifold (8) into the turbine (9). The exhaust gas is directed radially inward and circumferentially at high velocity by vanes (nozzles) onto the turbine wheel's blades where some of the exhaust gas energy is extracted as work or power. The turbine drives the compressor. A wastegate (valve) just upstream of the turbine bypasses some of the exhaust gas flow when necessary to prevent the boost pressure becoming too high. The wastegate linkage (11) is controlled by a boost pressure regulator (12). Figure 1.15 shows a cutaway drawing of a small automotive turbocharger. The arrangement of the compressor and turbine rotors connected via the central shaft and of the turbine and compressor flow passages are evident.

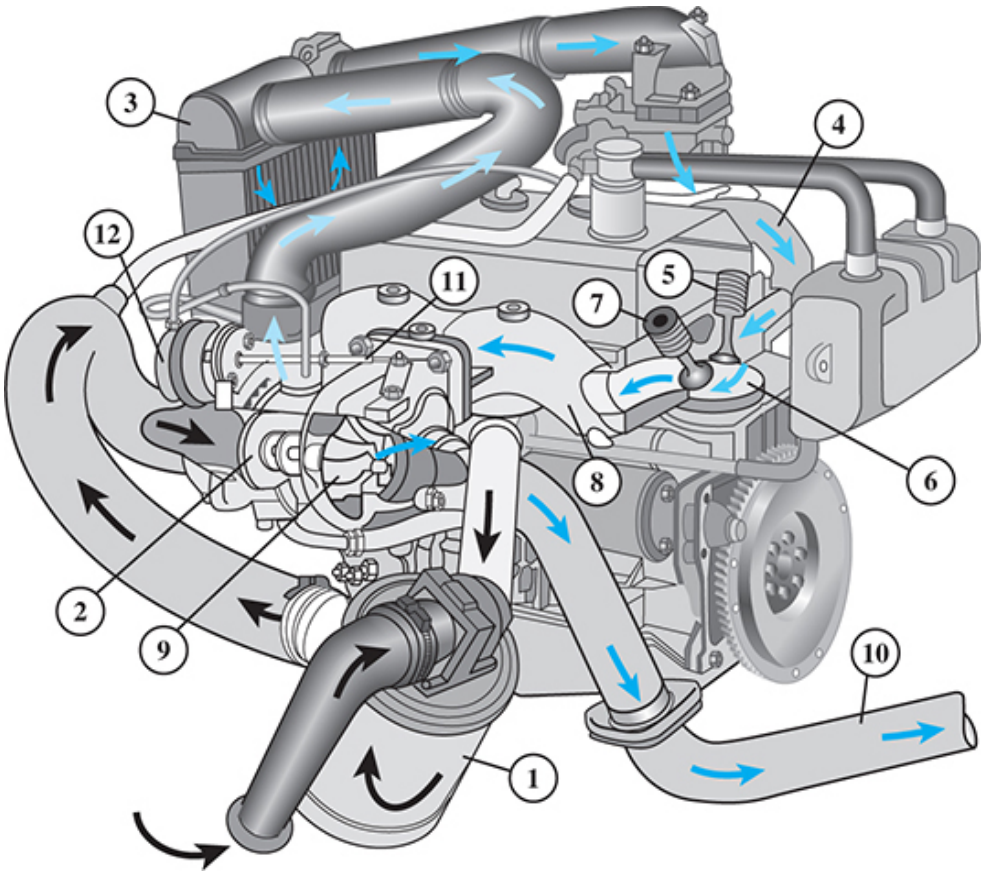


Figure 1.14 Drawing of turbocharger system connected to four-cylinder automobile spark-ignition engine. See text for details. (*Courtesy Regie Nationale des Usines*.)

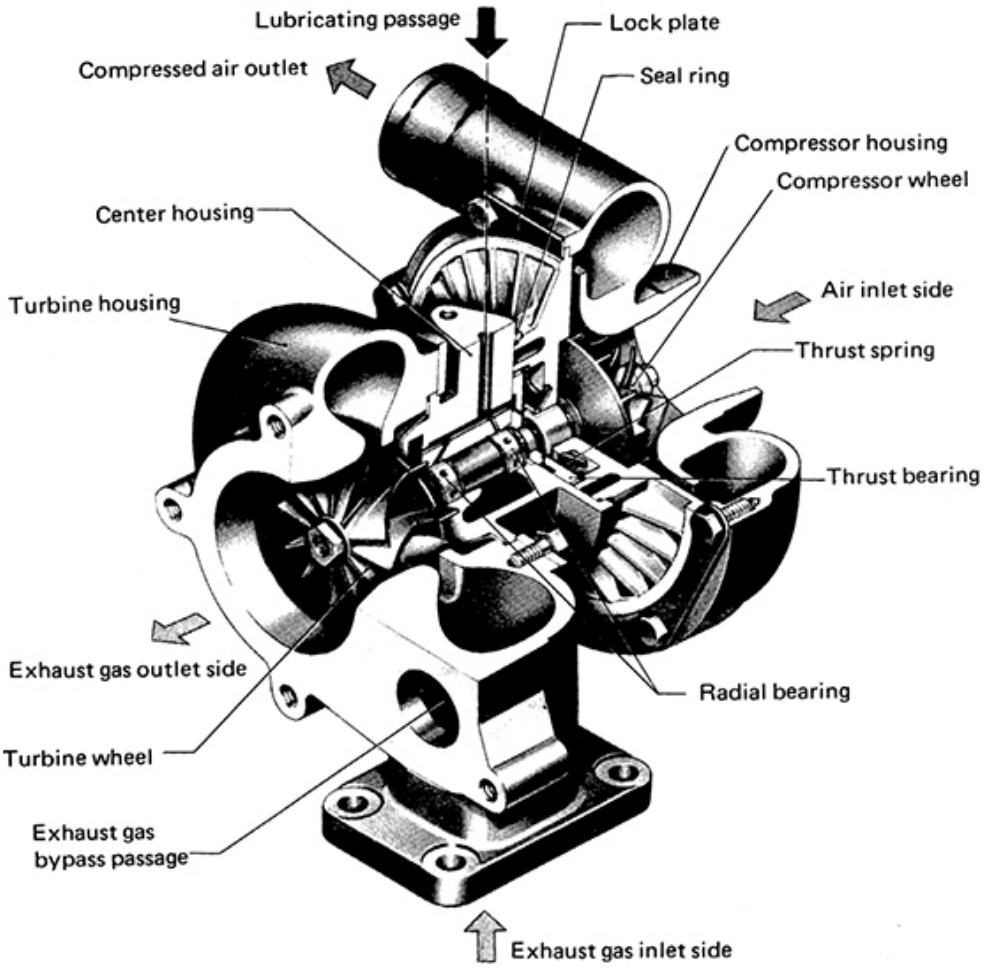


Figure 1.15 Cutaway view of small automotive SI engine turbocharger. (*Courtesy Nissan Motor Co. Ltd.*)

Increasing the intake air density, through boosting, increases the mass of air trapped within the cylinder, the mass of fuel burned, and thus the torque a given size engine can protrude. Torque increases of more than a factor of two can be realized. Turbocharging of SI engines is made difficult by the SI engine's knock constraint. The onset of knock (the rapid spontaneous ignition of a fraction of the in-cylinder fuel-air mixture) during the latter part of combustion) depends on the maximum mixture temperature and pressure reached inside the engine cylinder, and boosting raises both these variables. Special measures such as reducing the compression ratio, higher octane—better knock resisting—fuels have to be used to control knock. Direct fuel injection into the cylinder (see following section), with its charge-cooling effect, eases this problem.

A different type of boosted SI engine is large natural-gas fueled engines. These are used in electric power generation, propulsion, and marine applications. An example is shown in Fig. 1.16. It uses an encapsulated spark plug with orifices to improve ignition.

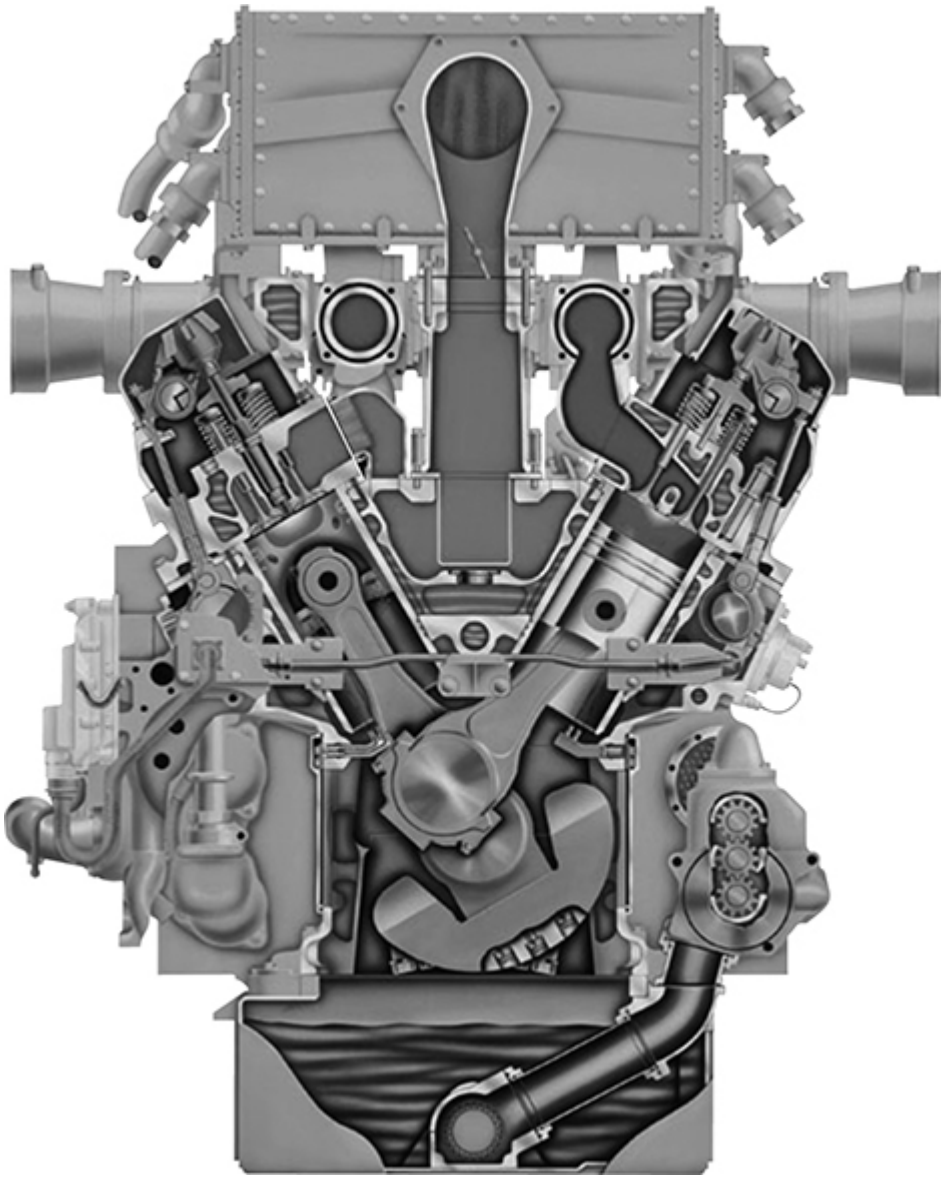


Figure 1.16 Large natural-gas-fueled boosted SI engine used in electric power generation. Bore 170 mm, stroke 190 mm, displaced volume per cylinder 4.3 liters, compression ratio 12:1, power (eight cylinders) 965 kW at 1500 rev/min. (*Courtesy Caterpillar, Inc.*)

1.7.4 Direct-Injection SI Engines

Since the 1920s, attempts have been made to develop internal combustion engines that combine the best features of the SI engine and the diesel. By injecting the gasoline fuel directly into each cylinder of the engine, better control of the fuel's behavior can be achieved, improving the engine's dynamic performance, permitting use of higher compression ratios, and reducing the losses resulting from throttling the airflow in the standard port-injected SI engine. Diesels are more efficient because they operate close to the optimum compression ratio (14 to 18), operate fuel lean (with excess air), and control engine output by varying the fuel flow rate while leaving the airflow unthrottled. Historically, direct-injection SI engines have often been called *stratified-charge engines* since to realize all these benefits, the mixing process between the evaporating fuel jet and the air in the cylinder must produce a "stratified" or nonuniform fuel-air mixture, with an easily ignitable composition at the spark plug at the time of ignition, and with excess air surrounding the fuel-containing spray.

Over the years, many different types of stratified-charge engine have been proposed; some are now being used in practice. [17] The operating principles of three of these early designs are shown in Fig. 1.17. The combustion chambers are bowl-in-piston designs, and a high degree of air swirl (rotation about the cylinder axis) is created during intake and enhanced in the piston bowl during compression to achieve rapid fuel-air mixing. With the Texaco [18] and MAN [19] systems (Figs. 1.17*a* and *b*), fuel is injected into the cylinder in tangentially into the bowl during the latter stages of compression. A long-duration spark discharge ignites the fuel-air jet as it passes the spark plug; the flame spreads downstream, and consumes the fuel-air mixture. Figure 1.17 *c* shows the Ford PROCO system[20] with its centrally located injector and hollow cone spray injected earlier in the compression stroke to get more complete fuel vapor/air mixing, so that high air utilization could be achieved to obtain high outputs.

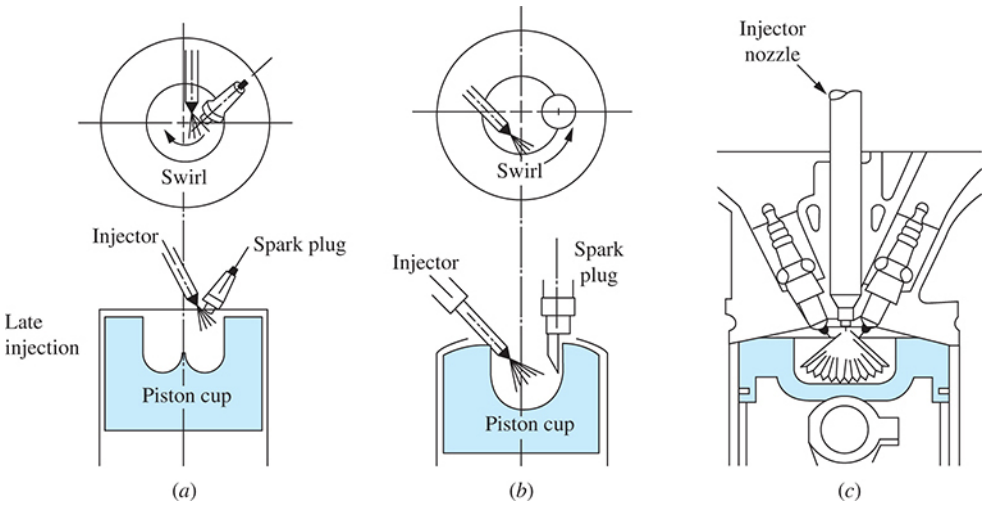


Figure 1.17 Three historical stratified-charge engines that were developed for production: (*a*) Texaco Controlled Combustion System (TCCS);[18] (*b*) M.A.N.-FM Combustion System;[19] (*c*) Ford PROCO Combustion System.[20]

Modern direct-injection SI engines are often divided in so-called spray-guided, wall-guided, and air-guided categories: see Fig. 1.18. This classification is based on the primary mechanism used to control the development of the fuel spray. In practice, mixture stratification is achieved through a combination of these mechanisms. The Texaco TCCS system in Fig. 1.17 *a* and the PROCO system in Fig. 1.17 *c* are examples of the former. The MAN system, Fig. 1.17 *b*, is primarily wall guided (with air swirl also playing an important role). The Texaco system, Fig. 1.17 *a*, is also air guided, with high air swirl generated during intake and augmented by the bowl-in-piston combustion chamber during compression. Many systems with significantly different geometric details are now being developed and employed in production: [17] see Sec. 7.7.2 . Generally, spray-guided approaches require a closer spacing between the injector and spark plug electrode location, as shown in Fig. 1.18, to limit the dispersion of the fuel spray and provide substantial mixture stratification. Wider spacing allows more time for fuel-air mixing, produces a more uniform composition spray, but then requires a specific combination of charge motion and wall guiding to achieve the desired spray behavior, and combustion, and emissions characteristics.

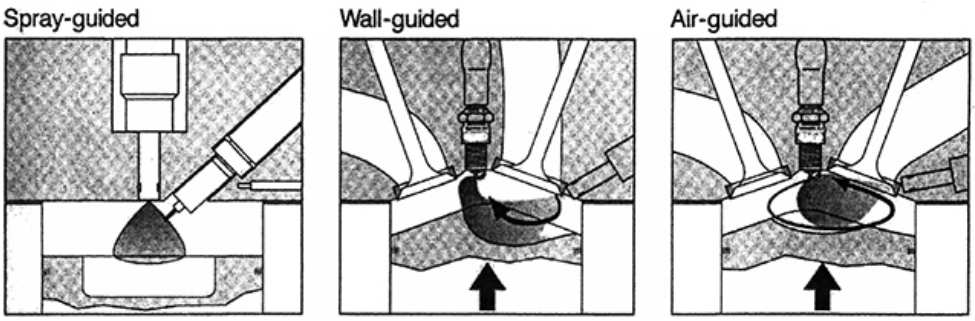


Figure 1.18 Illustrations of spray-guided, wall-guided, and air-guided direct-injection SI combustion systems.[17]

Figure 1.19 shows a production example of a direct-injected (DI) wall-guided system. This Mitsubishi gasoline DI engine used a spherically shaped cavity in the piston crown and an upright intake port to generate a reverse tumbling airflow in the cylinder during intake, to "guide" the developing spray toward the spark plug in the center of the cylinder head. Figure 1.20 shows a Toyota direct-injection engine design that uses a fan-shaped fuel spray directed into a shell-shaped bowl in the piston crown to provide rapid air-fuel mixing and fuel vaporization, and by the in-cylinder flow set-up by the straight intake port, to guide the spray so it reaches the spark plug location at the appropriate point in the cycle. To achieve high engine outputs, both these concepts transition from *late injection* (i.e., injection during the latter half of the compression stroke) when stratified operation is desired, to *early injection* (injection during the intake stroke) when essentially complete mixing of the injected fuel with *all* the air in the cylinder is required. This latter mode is called *homogeneous-charge* operation, as distinct from stratified operation.

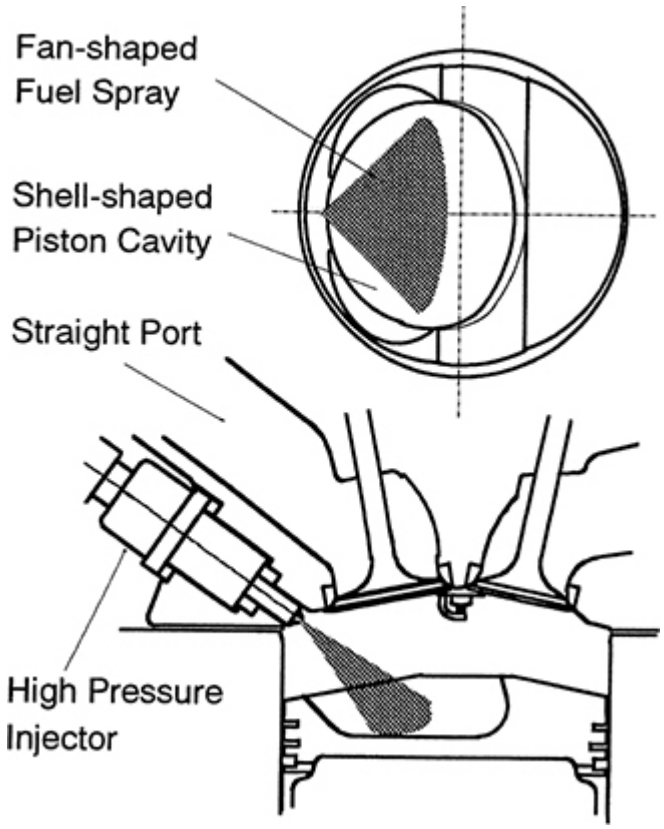


Figure 1.20 Toyota gasoline direct-injection SI engine design uses a wide-angle fan-shaped fine-atomization fuel spray injected into a shell-shaped piston cavity to achieve a stratified mixture with late injection, and homogeneous mixture with early (during intake stroke) injection.[22] This concept is also used in Toyota's homogeneous-charge direct-injection engines.

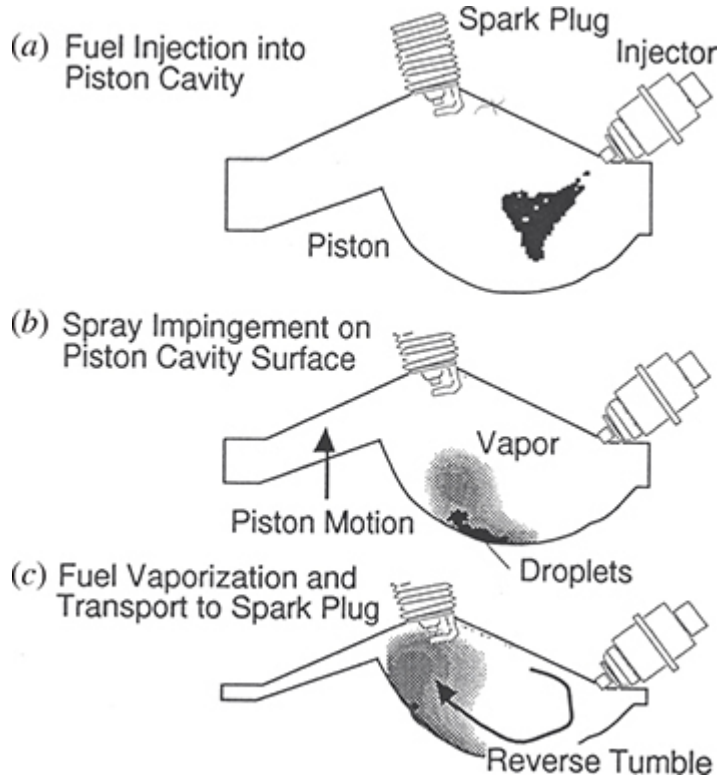


Figure 1.19 Mitsubishi gasoline direct-injection SI engine design. It uses a wide spacing between injector and spark plug; the spray is guided by the hemispherical piston cavity, and the reverse tumble produced by the upright intake port. In this 1.83-liter, four-cylinder, 12:1 compression ratio engine, the bore is 81 mm and the stroke is 89 mm. The fuel system uses an electromagnetic-controlled high pressure (5 MPa) swirl injector. [21]

Homogeneous-charge operation at all engine loads and speeds is a viable direct-injection SI engine approach, and is used in production engines. While the efficiency benefit of stratified operation with excess air (which the diesel enjoys) is lost, the in-cylinder charge cooling due to liquid fuel vaporization that increases the amount of air inducted and reduces the propensity of the engine to knock, and the more accurate control of fuel flow during engine transients, are retained. Homogeneous direct-injection engine concepts thus can increase compression ratio and efficiency, increase maximum power, and benefit from the effective emission-control technology that has been developed for port-injected SI engines.

Most production designs of direct-injection engines have used the four-stroke cycle. Direct-injection is, however, especially helpful in controlling

fuel carry through in two-stroke cycle engines (see Sec. 1.10). Direct fuel is especially attractive with turbocharged engines to increase their power density. The charge cooling, which evaporation of the in-cylinder fuel spray produces, reduces the engine's propensity to knock.

1.7.5 Prechamber SI Engines

An alternative to these open-chamber SI engines described above is a prechamber engine concept, which has been mass produced. It uses a small *prechamber* fed during intake with an auxiliary fuel system to obtain an easily ignitable mixture around the spark plug. This concept, first proposed by Ricardo in the 1920s and extensively developed in the Soviet Union and Japan, is often called a *jet-ignition* or *torch-ignition* stratified-charge engine. Its operating principles are illustrated in Fig. 1.21, which shows a three-valve carbureted version of the concept. [23] A separate carburetor and intake manifold feed a fuel-rich mixture (which contains fuel beyond the amount that can be burned with the available air) through a separate small intake valve into the prechamber that contains the spark plug. At the same time, a lean mixture (which contains excess air beyond that required to burn the fuel completely) is fed to the main combustion chamber through the normal intake manifold. During intake, the rich prechamber flow fully purges the prechamber volume. After intake valve closing, lean mixture from the main chamber is compressed into the prechamber bringing the mixture at the spark plug to an easily ignitable, slightly rich composition. After combustion starts in the prechamber, rich burning mixture issues as a jet (or series of jets) through one or more orifices into the main chamber, entraining and igniting the lean main chamber charge. This engine is really a jet-ignition concept whose primary function is to extend the operating limit of conventionally ignited SI engines to mixtures leaner than could normally be burned. This approach has been used in large natural gas engines to provide rapid initiation of the combustion process.

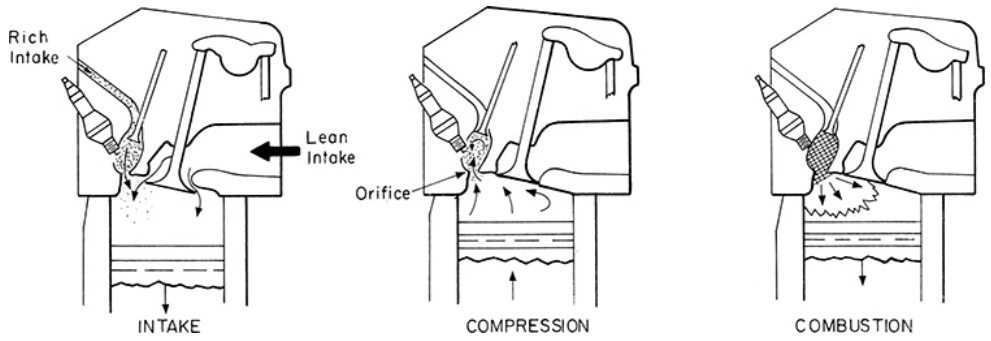


Figure 1.21 Schematic of three-valve torch-ignition stratified-charge spark-ignition engine.[23]

1.7.6 Rotary Engines

The reciprocating engine geometry discussed so far dominates the practical world of internal combustion engines. However, motivated by the fact that engine power is delivered through a rotating drive shaft, over the years many rotary engine designs have been proposed. [5] One of these, the Wankel rotary engine [6, 7] has and continues to be used in limited production. Its attractive features are its compactness and higher engine speed (which result in high power/weight and power/volume ratios), and its inherent balance and smoothness. These benefits are, however, offset by its higher heat transfer, and the engine's gas sealing and leakage problems.

Figure 1.22 shows the major mechanical parts of a simple single-rotor Wankel engine and illustrates its geometry and operation. There are two rotating parts: the triangular-shaped rotor and the output shaft with its integral eccentric. The rotor revolves directly on the eccentric. The rotor has an internal timing gear, which meshes with the fixed timing gear on one side housing to maintain the correct phase relationship between the rotor and eccentric shaft rotations. Thus the rotor rotates and orbits around the shaft axis. Breathing is through ports in the center housing (and sometimes the side housings). The combustion chamber lies between the center housing and rotor surface and is sealed by seals at each apex of the rotor and around the perimeters of the rotor sides. Figure 1.22 also shows how the Wankel rotary geometry operates with the four-stroke cycle. The figure shows the induction, compression, power, and exhaust processes of the four-stroke cycle for the chamber defined by rotor surface AB. The remaining two chambers defined

by the other rotor surfaces undergo exactly the same sequence. As the rotor makes one complete rotation, during which the eccentric shaft rotates through three revolutions, each chamber produces one power "stroke." Three power pulses occur, therefore, for each rotor revolution; thus for each eccentric (output) shaft revolution, there is one torque pulse. Figure 1.23 shows a cutaway drawing of an intake-port injected two-rotor automobile Wankel engine. The two rotors are out of phase to provide a greater number of torque pulses per shaft revolution. Note the combustion chamber cut out in each rotor face, and the rotor apex and side seals. Two spark plugs per firing chamber are often used to obtain a faster combustion process. Large area side intake and exhaust ports are used to increase airflow, and improve burned gas outflow.

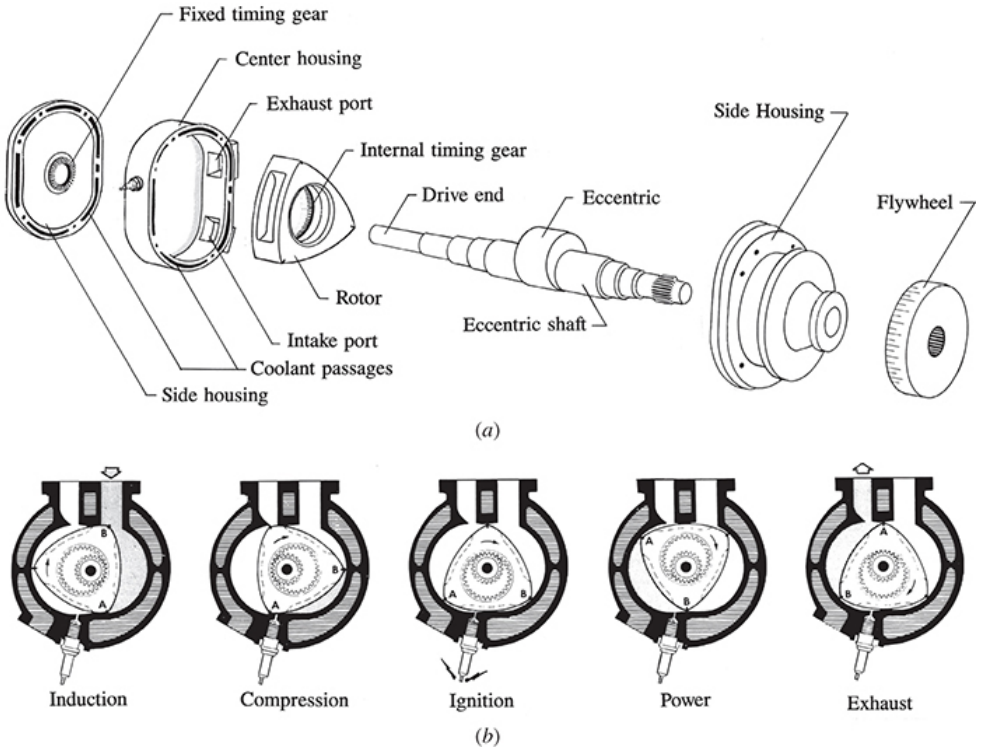


Figure 1.22 (*a*) Major components of the Wankel rotary engine. (*b*) Induction, compression, power, and exhaust processes of the four-stroke cycle for the chamber defined by rotor surface AB. (*From Mobil Technical Bulletin*, Rotary Engines, © *Mobil Oil Corporation, 1971*)

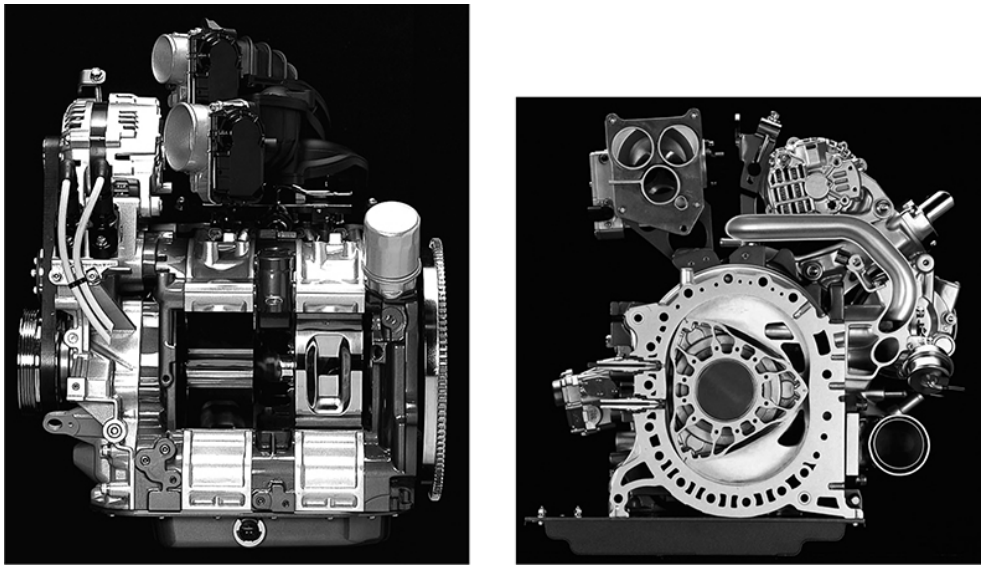


Figure 1.23 Mazda 1.3-liter RENESIS two-rotor Wankel rotary engine. Compression ratio 10:1, 15-mm eccentricity (offset between eccentric shaft axis and rotor centerlines), 105-mm generating radius (distance between rotor centerline and apex), trochaic (rotor) chamber width is 80 mm, giving 654 cm^3 displaced volume for each rotor chamber (1308 cm^3 total). Produces 184 kW at 8200 rev/min and 216 N · m of torque at 5500 rev/min.[24] (*Courtesy Mazda Motor Co.*)

1.8 COMPRESSION-IGNITION ENGINE OPERATION

In compression-ignition or diesel engines, air alone is drawn into the cylinder during intake. The fuel (in most applications a light fuel oil, though heated residual fuel is used in large marine and power-generating diesels) is injected directly into each cylinder just before the combustion process is required to start. Load control is achieved by varying the amount of fuel injected each cycle; the airflow at a given engine speed is not directly controlled, and in naturally-aspirated engines is essentially unchanged. There are a great variety of CI engine designs in use in a wide range of applications —automobile, truck, locomotive, marine, power generation. Both naturally-aspirated engines where atmospheric air is inducted, and boosted engines where the inlet air is compressed by a turbocharger—an exhaust-driven

turbine-compressor combination—are common. Turbocharging increases engine output by increasing the air mass flow per unit displaced volume, thereby allowing an increase in fuel flow. These devices are used to reduce engine size and weight for a given power output. In large engine sizes, the two-stroke cycle is competitive with the four-stroke cycle because in these large low-speed engines, the cylinder can be scavenged more effectively and, with the diesel's direct fuel injection, only air is lost in the scavenging process.

The operation of a typical four-stroke naturally-aspirated CI engine is illustrated in Fig. 1.24. The compression ratio of diesels is much higher than typical SI engine values, and is in the range 14 to 22, depending on the type of diesel engine and whether the engine is naturally aspirated or turbocharged. The valve timings used are similar to those of SI engines. In a naturally-aspirated engine, air at close-to-atmospheric pressure is inducted during the intake stroke and then compressed to a pressure of about 5 MPa (50 atm) and temperature of about 900 K (600°C) during the compression stroke. At about 20 crank angle degrees before TC, fuel injection into the engine cylinder commences; a typical rate of injection profile is shown in Fig. 1.24 *b*. Usually there are four to eight or more liquid fuel jets; each jet exiting the injector nozzle atomizes into drops and entrains air to form a set of sprays that penetrate into the bowl-in-piston combustion chamber, as shown in Fig. 1.25. In each spray, the liquid fuel drops evaporate, and the fuel vapor then mixes with the entrained air. The fuel-vapor air mixture temperature and pressure are above the ignition point where fuel oxidation chemistry can occur. Thus, after a short *delay period*, spontaneous ignition (autoignition) of parts of the nonuniform fuel-air mixture within these sprays initiates the combustion process, and the cylinder pressure (solid line in Fig. 1.24 *c*) rises above the nonfiring engine level as fuel chemical energy is released. A diffusion flame then spreads rapidly to surround each fuel spray, with partly reacted fuel-air mixture in the spray on the inside of the flame, and the additional air in the cylinder on the outside. As the expansion process proceeds, mixing between fuel vapor, air, and burning gases continues, accompanied by further combustion (Fig. 1.24 *d*). At full load, the mass of fuel injected is about 5% of the mass of air in the cylinder. At higher fueling levels, increasing amounts of black smoke in the exhaust limit the quantity of fuel that can be burned efficiently. The exhaust process is similar to that of the four-stroke SI engine: a rapid outflow or blowdown of burned gases as

soon as the exhaust valves start opening, followed by displacement of the remaining burned gases from the cylinder during the exhaust stroke. At the end of the exhaust stroke, the cycle starts again.

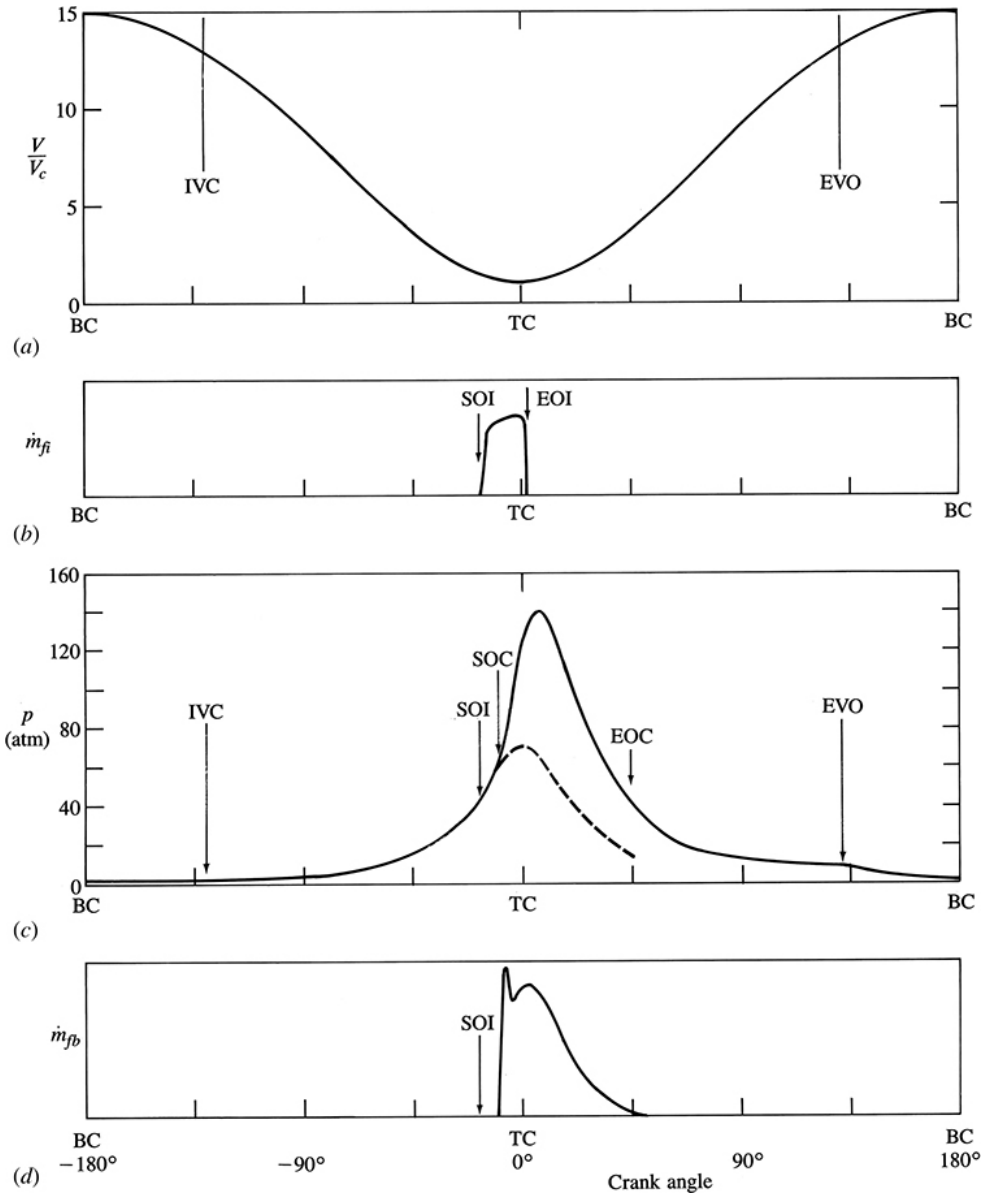


Figure 1.24 Sequence of events during compression, combustion, and expansion processes of a naturally-aspirated compression-ignition engine operating cycle. (*a*) Cylinder volume/clearance volume V/V_c, (*b*) rate of fuel injection $\dot{m}_{fi}$, (*c*) cylinder pressure p (solid line, firing cycle; dashed line, motored cycle), and (*d*) rate of fuel burning (or fuel chemical energy release rate) $\dot{m}_{fb}$ are plotted against crank angle.

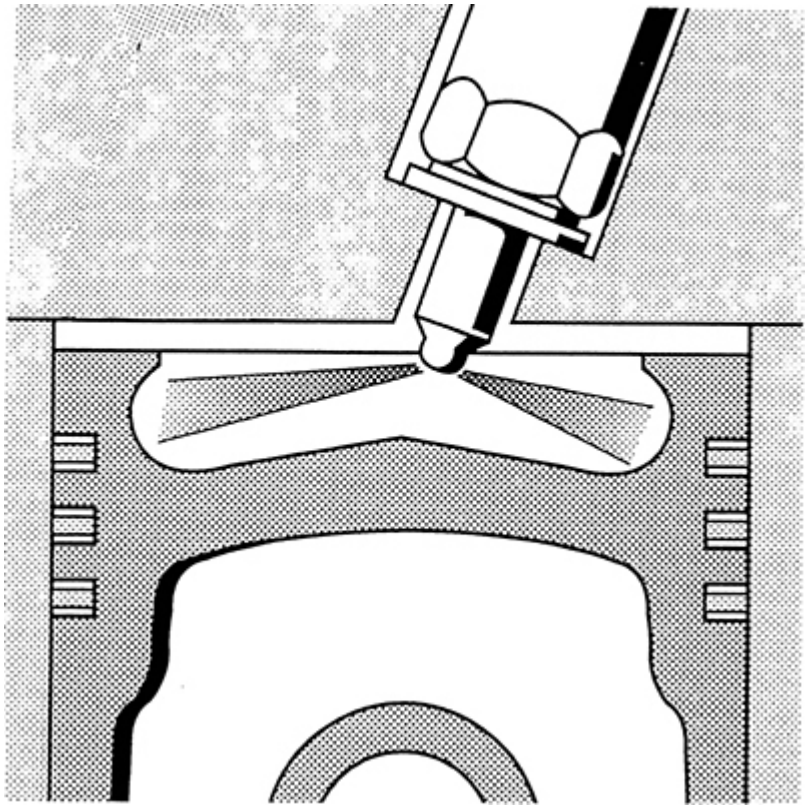


Figure 1.25 Schematic of diesel engine fuel sprays, formed from liquid fuel jets injected at high pressure through individual injection nozzle holes, penetrating the diesel bowl-in-piston combustion chamber.

In this diesel combustion process, the fuel is injected directly into the engine cylinder at a pressure of between several hundred and more than 2000 bar. [25] The diesel fuel-injection system consists of a low-pressure pump, and a high-pressure injection pump, delivery pipes, and fuel injector nozzles. Several different types of injection pumps and nozzles are used. In one common system shown in Fig. 1.26, an in-line pump containing a set of cam-driven plungers (one for each cylinder) operate in closely fitting barrels. Early in the stroke of the plunger, the inlet port is closed and the fuel trapped above the plunger is forced through a check valve into the injection line. The injection nozzle (Fig. 1.27) has one or more holes through which the fuel sprays into the cylinder. A spring-loaded valve closes these holes until the pressure in the injection line, acting on part of the valve surface, overcomes the spring force and opens the valve. Injection starts shortly after the line pressure begins to rise. Thus, the phasing of the pump camshaft relative to the

engine crankshaft controls the start of injection. Injection is stopped when the inlet port of the pump is uncovered by a helical groove in the pump plunger, because the high pressure above the plunger is then released (Fig. 1.27, bottom). The amount of fuel injected (which controls the load) is determined by the injection pump cam design and the position of the helical groove. Thus for a given cam design, rotating the plunger and its helical groove varies the load.

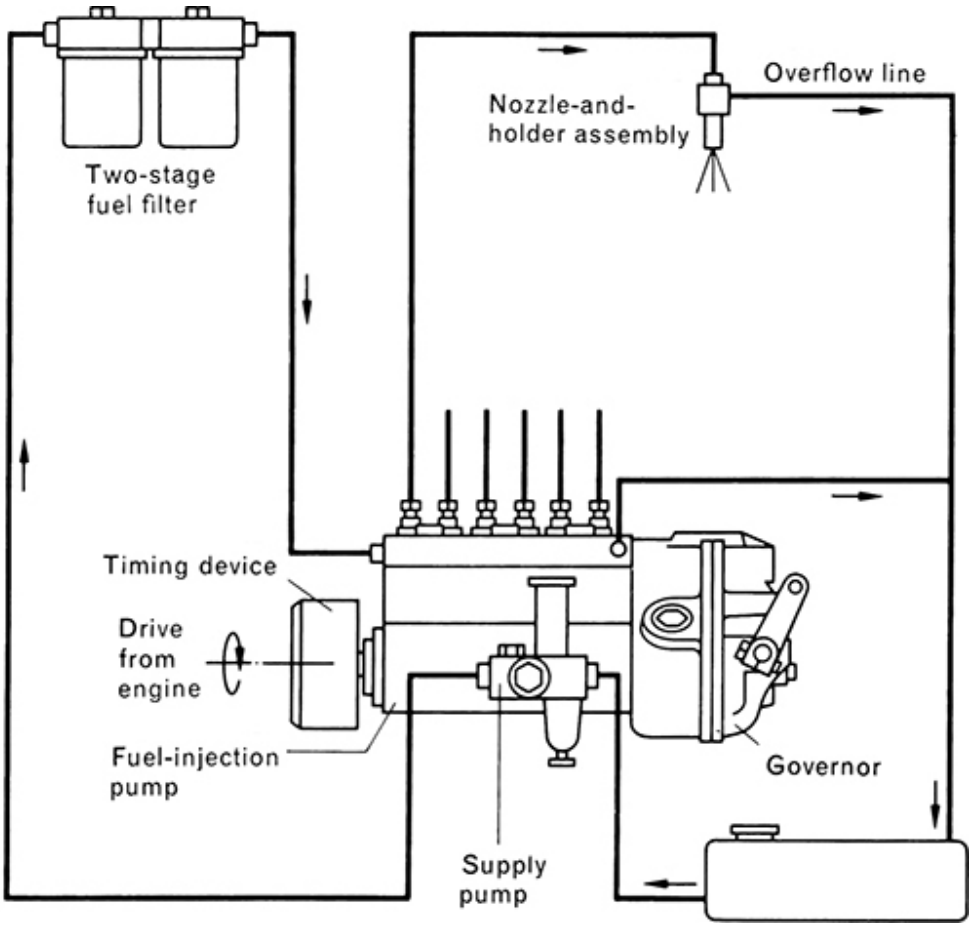


Figure 1.26 Diesel fuel system with in-line fuel-injection pump.[25] (*Courtesy Robert Bosch GmbH and SAE.*)

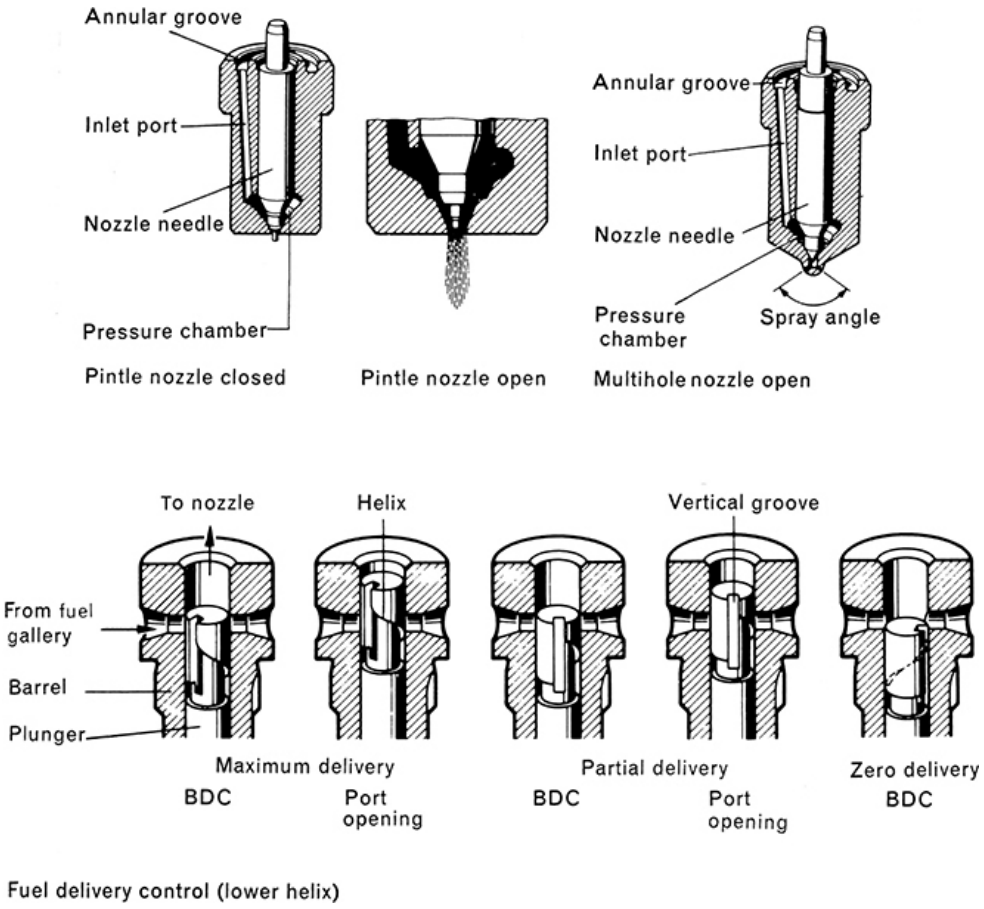


Figure 1.27 Details of fuel-injection nozzles and fuel-delivery control.[26] (*Courtesy Robert Bosch GmbH and SAE.*)

In smaller diesel engines, distributor-type fuel pumps are often used. These have one pump plunger and barrel, which meters and distributes the fuel to all the injection nozzles. The unit contains a high-pressure injection pump, an overspeed governor, and an injection timer. High pressure is generated by the plunger, which is made to describe a combined rotary and stroke movement. This rotary motion distributes the fuel to the individual injection nozzles. Distributor pumps can operate at higher speed and take up less space than in-line pumps. They are normally used on smaller three- to six-cylinder diesel engines. In-line pumps are used in larger, midsize engines.

An alternative fuel-injection approach uses individual single-barrel injection pumps, close mounted to each cylinder with an external drive. These unit injector systems (UIS) combine the pump and injector into a single

unit. Figure 1.28 illustrates how this system operates. The unit is driven by the engine camshaft, and the start of injection and injected fuel quantity are controlled by a solenoid valve in the injector. With this type of system, very high injection pressures (some 2000 bar) are achieved along with precise control of the amount and timing of injection.

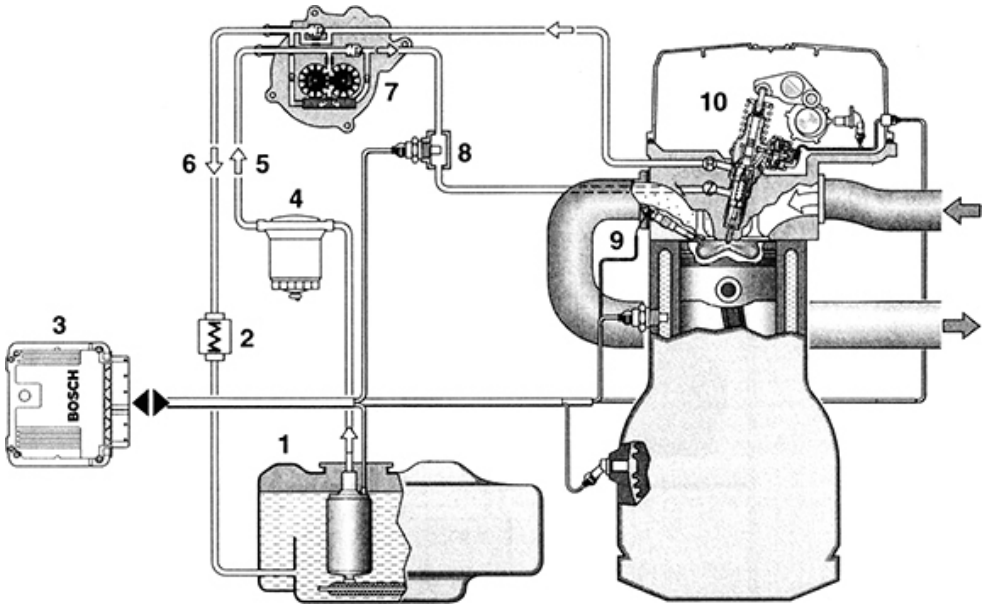


Figure 1.28 Diesel fuel-injection system with unit injector (for passenger cars). (1) Fuel tank with fuel supply pump; (2) Fuel cooler; (3) Electronic control unit; (4) Fuel filter; (5) Fuel feed line; (6) Fuel return line; (7) Tandem pump; (8) Fuel-temperature sensors; (9) Glow plug; (10) Injector. [26] (*Courtesy Robert Bosch GmbH and SAE.*)

Common rail (or fuel accumulation) injection systems allow greater freedom to control the fuel-injection process, and thus combustion. The functions of fuel pressure generation and fuel injection are separated by an accumulator or common rail. Figure 1.29 shows the system layout. The high-pressure pump feeds the common rail, which feeds each of the injectors. Control of this system can readily be integrated with other engine parameters as indicated. Injection pressures of 1400 to 1800 bar can be achieved. By repeated activation of the fast-acting solenoid valve within the injector, multiple injection pulses in each injection cycle can be realized. It is often advantageous to use a short *pilot injection* before the main injection, to initiate combustion with a small amount of fuel, and reduce engine noise.

Multiple pulse main injections can help control emissions. Both solenoid-controlled injectors and piezo-actuated injectors are used. Figure 1.30 shows a piezoelectric injector that provides more precise control of injection pulses and improves fuel atomization within the cylinder.

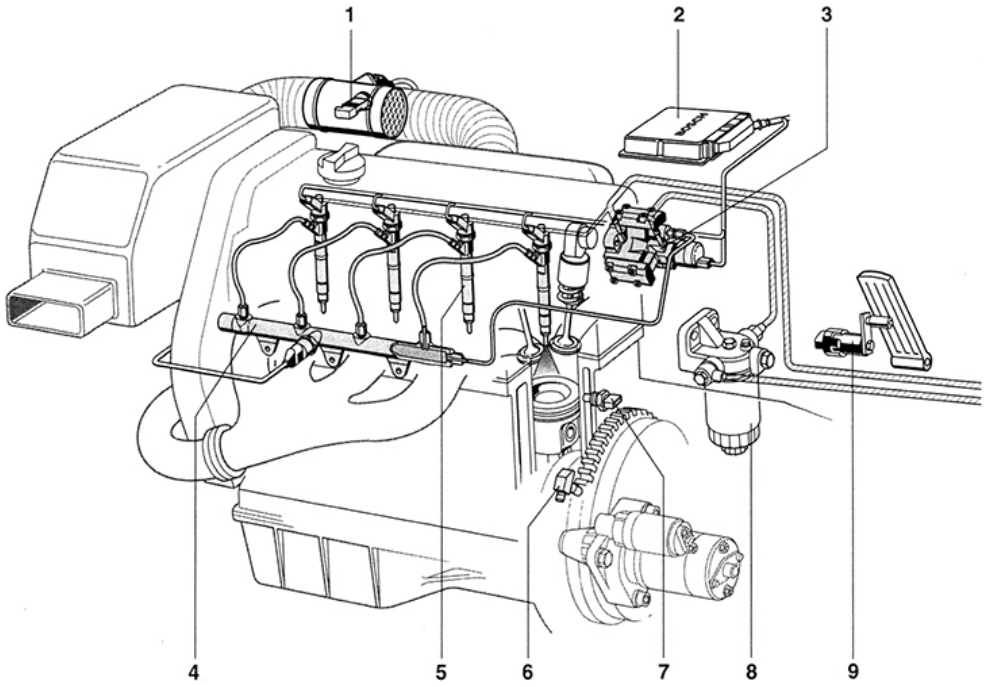


Figure 1.29 Common rail accumulator fuel-injection system on a four-cylinder automobile diesel.[25] (1) Air-mass flow meter; (2) Electronic control unit; (3) High-pressure pump; (4) High-pressure accumulator (rail); (5) Injectors; (6) Crankshaft speed sensor; (7) Coolant temperature sensor; (8) Fuel filter; (9) Accelerator-pedal sensor. (*Courtesy Robert Bosch GmbH and SAE*.)

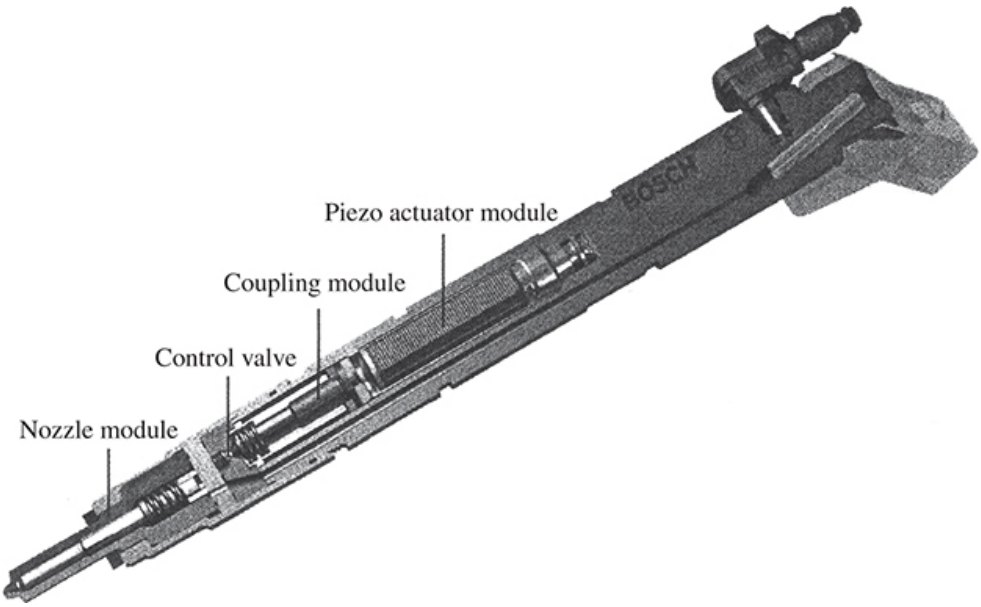


Figure 1.30 Bosch piezoelectric fuel injector that provides more precise fuel injection control on opening and closing, and finer atomization.[27] (*Courtesy Robert Bosch GmbH and SAE.*)

1.9 DIFFERENT TYPES OF DIESEL ENGINES

A large number of diesel engine configurations and designs are in common use. The largest marine and stationary power-generating diesels are two-stroke cycle engines and are discussed in the next section, 1.10. Small- and medium-size diesel engines use the four-stroke cycle. Because air capacity is an important constraint on the amount of fuel that can be burned in the diesel engine, and therefore on power, turbocharging is used extensively. Larger engines are almost always turbocharged. Small low-cost diesels are not usually turbocharged. The details of the engine design also vary significantly over the diesel size range. In particular, different combustion chamber geometries and fuel-injection characteristics are required to deal effectively with a major diesel engine design problem: achieving sufficiently rapid fuel-air mixing rates to complete the fuel-burning process in the short time available. Smaller engines run at higher maximum speeds. Thus, a wide variety of inlet port geometries, cylinder head and piston cavity or bowl shapes, and fuel-injection patterns are used to achieve the airflows and fuel

flows needed to accomplish fast enough combustion over the diesel size range.

Figure 1.31 shows a diesel engine typical of the heavy-duty truck application. The design shown is a six-cylinder in-line engine. The drawing indicates that diesel engines are generally substantially more rugged and heavier than SI engines because stress levels are higher due to the significantly higher pressure levels of the diesel cycle. The engine shown has a displacement of 12.9 liters, a compression ratio of 16.5, and is turbocharged. This type of diesel is called a *direct-injection* diesel since the fuel is injected directly into a combustion chamber above the piston crown. The combustion chamber shown is a "bowl-in-piston" design, which puts most of the clearance volume into a compact cavity in the piston crown. With this type of diesel engine, it is often necessary to use a swirling airflow rotating about the cylinder axis, which is created by suitable design of the inlet port and valve, to achieve fast enough fuel-air mixing and fuel-burning rates. The fuel injector, shown on the cylinder axis in the drawing, has a multihole nozzle. It typically has four to six holes in this application. The fuel jets move out radially from the nozzle holes close to the center of the piston bowl into the (swirling) airflow.

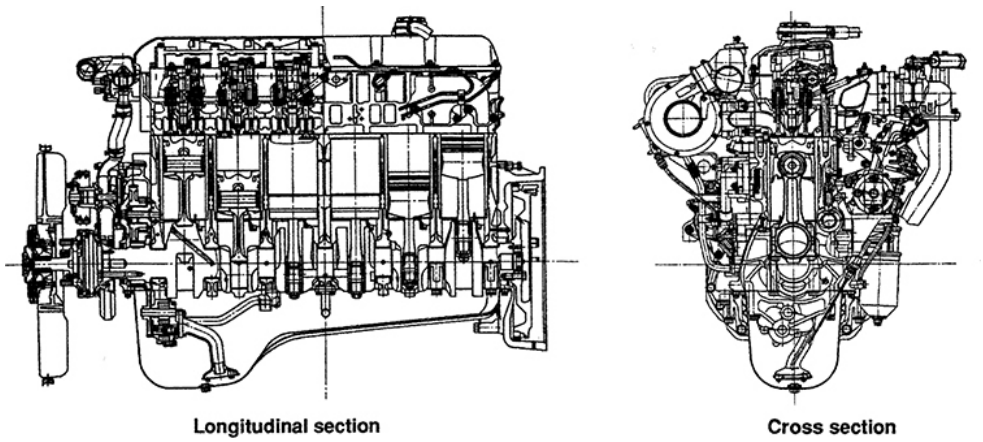


Figure 1.31 Six-cylinder 12.9-liter turbocharged direct-injection truck diesel engine: transverse and longitudinal sections. Bore 135 mm, stroke 150 mm, compression ratio 16:1, governed speed 2100 rev/min, maximum output 294 kW, maximum torque 1667 N · m at 1300 rev/min. [28]

Figure 1.32 shows a 2.2-liter four-cylinder high-speed direct-injection (HSDI) diesel typical of those used in automobiles. It has four valves per cylinder, with the fuel injector and bowl-in-piston combustion chamber centered on each cylinder axis. These types of engines are highly boosted to give high torque at low- to mid-engine speeds per unit displaced cylinder volume. The compressor exit air, cooled in a heat exchanger, enters the cylinder via intake ports that generate swirl about the cylinder axis. During the compression stroke, the flow of this swirling air into the reentrant bowl-in-piston significantly enhances the swirl, thereby achieving high rates of mixing of air with the injected fuel sprays close to top center. Maximum rated speeds (4500 to 5000 rev/min) are lower than maximum speeds of gasoline SI engines due to fuel-air mixing rate limitations. The compression ratio of these HSDI engines is 18 to 20:1, somewhat above the value that gives maximum engine efficiency. This is done to obtain higher air compression temperatures to enable cold engine starting at low cranking speeds. An electrically heated ceramic *glow plug* is usually inserted through the cylinder head in these size engines to assist the cold starting process.

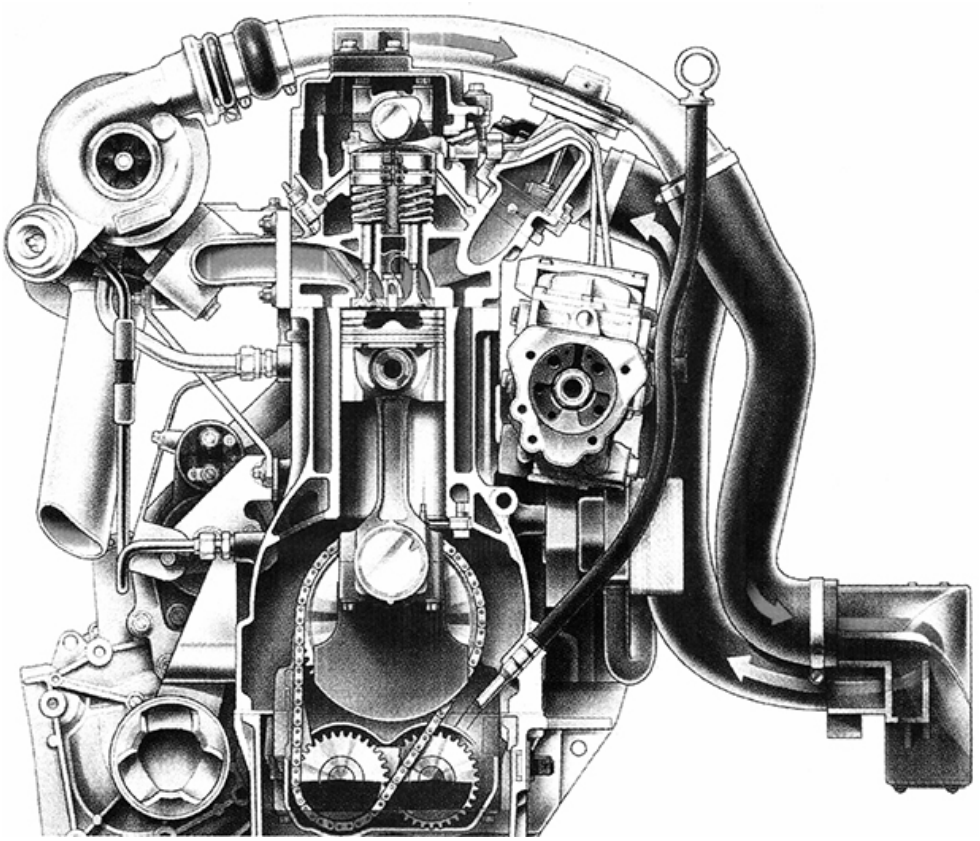


Figure 1.32 Small 2.2-liter four-cylinder high-speed direct-injection automobile diesel engine: maximum power 93 kW; maximum torque 285 N · m at 1750 rev/min; maximum boost 0.9 bar (gauge); compression ratio 18.5. (*Courtesy SAE and Saab*.)

The smallest diesels operate at higher engine speeds than do larger engines: hence the time available for burning the fuel is less and the fuel-injection and combustion systems must achieve faster fuel-air mixing rates. Figure 1.32 shows how this rapid combustion is realized in HSDI diesels. Historically, in the smallest sizes diesel engines, this can be accomplished by using an *indirect-injection* or prechamber type of diesel. Fuel is injected into an auxiliary combustion chamber that is separated from the main combustion chamber above the piston by a passageway or nozzle. During the latter stages of the compression process, air is forced through this nozzle from the cylinder into the prechamber at high velocity. Fuel is injected into the highly turbulent and often rapidly swirling flow in this prechamber, and very high fuel-air mixing rates are achieved. Combustion starts in the prechamber, and the resulting pressure rise in the prechamber forces burning gases, fuel, and air into the main chamber. Since this outflow is also extremely vigorous, rapid mixing then occurs in the main chamber as the burning jet exiting the prechamber mixes with the remaining air and combustion is completed. A glow plug is also shown in the auxiliary chamber; this plug is electrically heated during cold engine start-up to raise the temperature of the air charge and the fuel sufficiently to achieve autoignition. The compression ratio of this engine is high—around 20. Indirect-injection diesel engines require higher compression ratios than direct-injection engines to start adequately when cold.

1.10 TWO-STROKE CYCLE ENGINE OPERATION

The two-stroke engine is used at the small-size and very large-size ends of the engine market. In small sizes, the two-stroke cycle SI engine is cheap, compact and light, simple, and robust. This is the basis of its market appeal in mopeds, scooters, motorcycles, and snowmobiles, in portable devices

such as chain-saws and bush cutters, in agricultural and construction devices such as lawn mowers, disc saws, and snow blowers, in the outboard marine engine arena, and in light and in remotely piloted aircraft. The very large diesel engines used in marine and power-generation applications are also two-stroke cycle engines. These large internal combustion engines are the most efficient and cost effective prime movers currently available. The two-stroke diesel has also been used in the locomotive and in parts of the truck market. [29,30] The passenger-car and truck engine markets are now, however, dominated by four-stroke cycle engines.

Key operating features of the two-stroke cycle are its power stroke every crankshaft revolution and its scavenging of the burned gases from the engine cylinder with fresh charge. Doubling the number of power strokes per unit time relative to the four-stroke cycle increases the power output per unit displaced volume. It does not, however, increase by a factor of two. The outputs of two-stroke engines range from only 20% to 60% above those of equivalent-size four-stroke units. This lower increase in practice is a result of the poorer than ideal charging efficiency, that is, incomplete filling of the cylinder volume with fresh air due to incomplete scavenging of the residual burned gases. Doubling the number of power strokes per unit time also halves the intervals between combustion-generated pressure impulses. This results in a smoother crankshaft torque versus time profile.

The two-stroke cycle's process of scavenging the burned gases from the engine cylinder with fresh charge—its gas exchange process—has several consequences. First, charging losses are inevitable. Under higher load conditions, in a typical small two-stroke engine, some 20% or more of the fresh charge that enters the cylinder is lost due to short-circuiting to the exhaust. When the fuel is mixed with air prior to cylinder entry, this process results in very high hydrocarbon emissions and poor fuel consumption compared with the four-stroke cycle engine. However, as both exhaust and charging occur around BC, the exhaust and intake ports can be situated near the bottom end of the cylinder and can be covered and uncovered by a long-skirt piston (see Fig. 1.33). This simple geometric two-stroke cycle configuration obviates the need for valves and their actuating gear. It also substantially simplifies the engine structure and the production process, and significantly reduces engine cost.

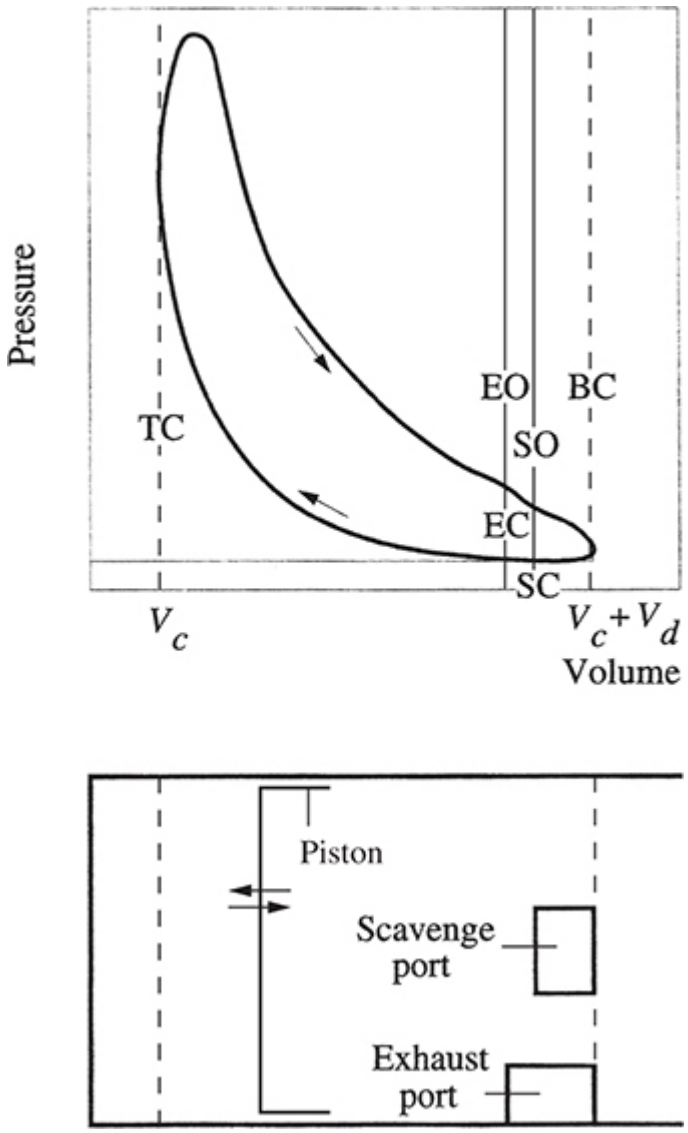


Figure 1.33 Cylinder-pressure versus cylinder-volume trace for a two-stroke cycle engine cylinder. Exhaust and transfer or scavenge ports are uncovered by the piston as shown.

In the two-stroke engine cycle, the compression, combustion, and expansion processes are similar to the equivalent four-stroke cycle processes; it is the intake and exhaust processes that are different (Fig. 1.33). The sequence of events in a port-scavenged two-stroke engine is illustrated in Fig. 1.34. In such engines, both exhaust and the scavenging (or transfer) ports are at the same end of the cylinder and are uncovered as the piston approaches BC (Fig. 1.34). After the exhaust ports open, the cylinder

pressure falls rapidly as burned gases flow out of the cylinder into the exhaust system, in a blowdown process as shown. The scavenging or transfer ports then open, and once the cylinder pressure p falls below the scavenging pressure p_i, fresh charge flows into the cylinder. Burned gases, displaced by this fresh charge, continue to flow out of the exhaust port (along with some of the fresh charge). Once the ports close as the piston starts the compression stroke, compression, fuel-injection and fuel-air mixing in direct-injection engines, combustion, and expansion processes proceed as in the equivalent four-stroke engine cycle. [29, 30] (Fresh charge is fuel vapor and air in engines where fuel and air are "premixed" before entry into the cylinder: the fresh charge is air if direct fuel injection is used.)

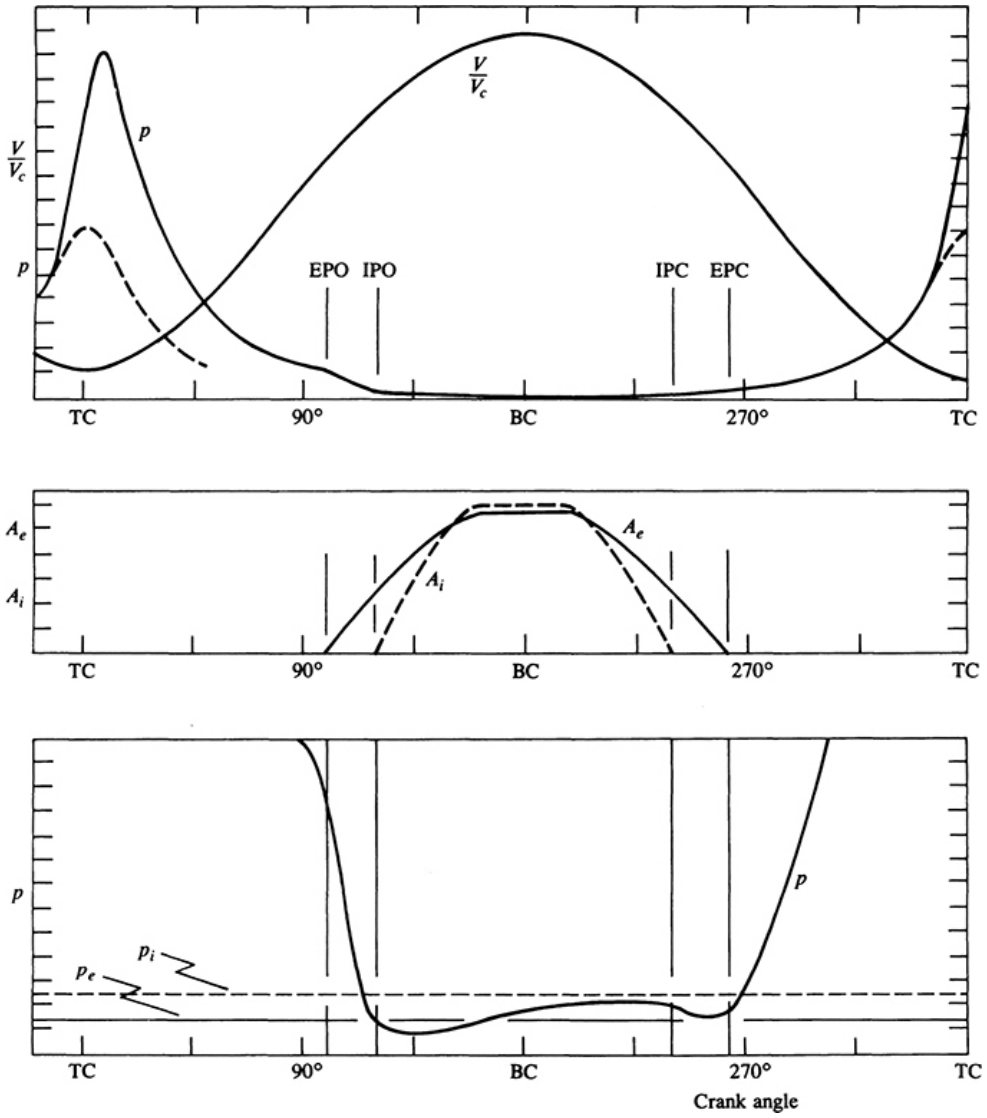


Figure 1.34 Sequence of events during expansion, gas exchange, and compression processes in a loop-scavenged two-stroke cycle compression-ignition engine. Cylinder volume/clearance volume V/V_c, cylinder pressure p, exhaust port open area A_e, and intake port open area A_i are plotted against crank angle.

Figure 1.35 shows how this two-stroke cycle is realized in a small gasoline engine.[31] This is an experimental single-cylinder crankcase-scavenged engine, with a pneumatic direct fuel injection system to avoid high hydrocarbon emissions and achieve good fuel consumption. Air flows into the crankcase, through Reed valves, as the piston travels up the cylinder. This air is compressed as the piston travels down. Air flows into the cylinder once the transfer or scavenging ports, which connect the crankcase to the cylinder, are uncovered. Fuel is injected after the transfer ports have been closed off. This engine employs a novel combustion approach—often called *activated radical combustion*. By retaining the appropriate amount of burned residual gas within the cylinder by restricting the flow out of the exhaust ports with a throttle valve, the in-cylinder unburned gas temperature at the end of compression can be raised and controlled. This in-cylinder mixture can then be made to autoignite at the appropriate point in the cycle—at the end of compression, just before TC. This spontaneous autoignition of the complete well-mixed fuel vapor, air, and burned residual in-cylinder charge is an alternative combustion process to the standard spark-ignited or diesel combustion processes. It is often called *homogeneous charge compression ignition* (HCCI) or *controlled autoignition* (CAI). This HCCI combustion concept is also being developed for four-stroke cycle engines. At high outputs, when effective scavenging of the burned gases becomes important, this two-stroke cycle engine reverts to the normal spark-ignition engine flame propagation process.

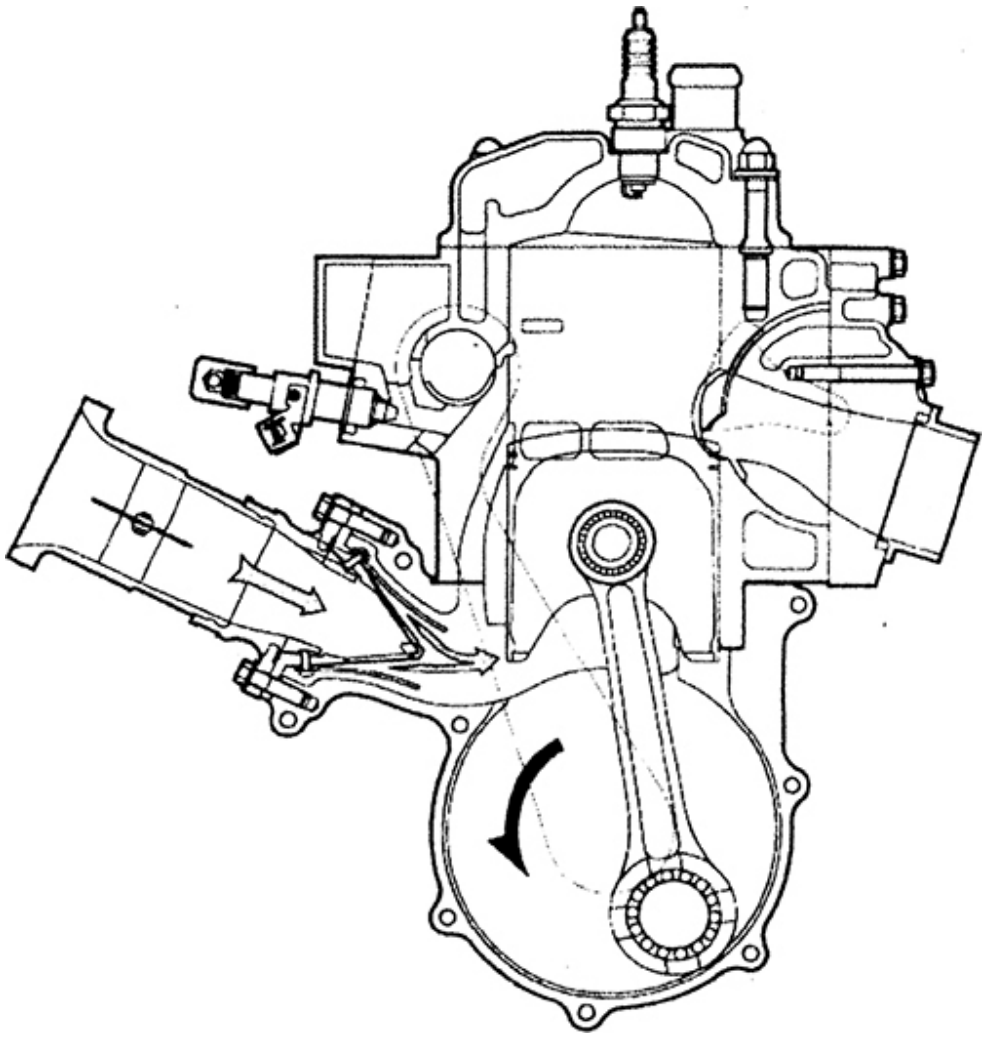


Figure 1.35 Honda experimental two-stroke cycle loop-scavenged engine with pneumatic direct in-cylinder fuel injection, employing an activated radical combustion process—controlled autoignition of the in-cylinder fuel-vapor, air, burned residual mixture just before top center. 402 cm^3 displacement, bore and stroke 80 mm, trapped compression ratio 6:1. Generates 33 kW at 6900 rev/min. [31]

Diesels, in the very large size engines, used for marine propulsion and electrical power generation, also operate using the two-stroke cycle. Figure 1.36 shows such a two-stroke cycle marine engine, available with 4 to 12 cylinders, with a maximum bore of 0.6 to 0.9 m and stroke of 2 to 3 m, which operates at speeds of about 100 rev/min. These engines are normally of the crosshead type shown to reduce piston side forces on the cylinder. The gas

exchange process is initiated by opening the exhaust valve in the cylinder head, followed by the piston uncovering the transfer ports at the lower end of the cylinder liner. The expanding exhaust gases leave the cylinder via the exhaust valve and manifold, and pass through the turbocharger turbine. Compressed air enters the cylinder via the transfer ports, continuing the scavenging process; the air is supplied from the turbocharger compressor and intercooler. At part load, electrically driven blowers cut in to compress the scavenge air. Because these large engines operate at low speed, the motion induced by the injected fuel jets is sufficient to mix the fuel with air and burn it in the time available. Thus, a simple open combustion chamber shape can be used, which achieves efficient combustion even with the low-quality heavy fuels used with these types of engines. The pistons are water cooled in these very large engines. The splash or jetted oil piston cooling used in small- and medium-size diesels is not adequate.

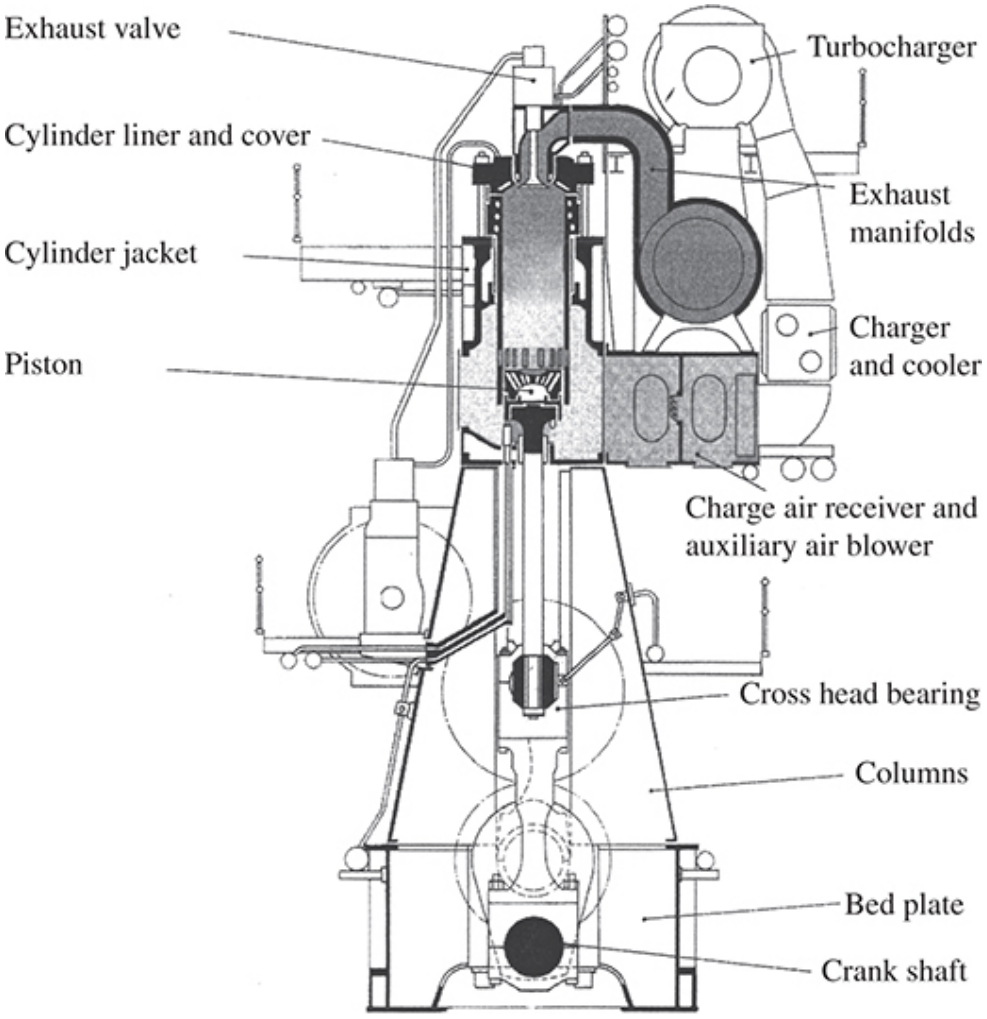


Figure 1.36 Cross-section of an IHI-Sulzer uniflow-scavenged large two-stroke cycle turbocharged diesel engine, developing 1590 kW per cylinder at 127 rev/min. Stroke to bore ratio about 2:5. [30]

1.11 FUELS

1.11.1 Gasoline and Diesel

Crude oil is the primary source or *feedstock* used today to produce transportation fuels, accounting for more than 95% of transport energy. The gasoline (or petrol) and diesel fuel produced from crude oil are the overwhelmingly dominant *energy carriers* used in SI and CI (or diesel) engines, respectively. Crude oil contains a large number of different hydrocarbons, and these compounds range from gases to liquids to waxes. In a refinery, the crude oil is physically separated by distillation into various fractions. Portions of these fractions are then chemically processed into fuels and other products. [32] Figure 1.37 shows the layout of a typical refinery. [33] The crude oil is first separated into various fractions (each with a higher boiling point range) referred to as *naphtha, distillate, gas oil*, and *residual oil*. The terms *light*, *middle*, or *heavy* break these out further by volatility. The terms *virgin* or *straight run* indicate that no chemical processing has been done to the fraction. As shown, virgin naphtha can be used directly as gasoline. Figure 1.37 also shows the different chemical processes typically used: alkylation, reforming, coking, and catalytic cracking. The refinery products are gasoline, kerosene and jet fuel, diesel, heating oil, burner fuel, residual fuel oil, coke, chemical feedstocks, and asphalt. On average, a refinery will refine 25 to 45% of the input crude oil into gasoline, 25 to 40% into diesel, jet fuel, and heating oil, 5 to 20% into heavy fuel oils, and the remaining 20% into other products.

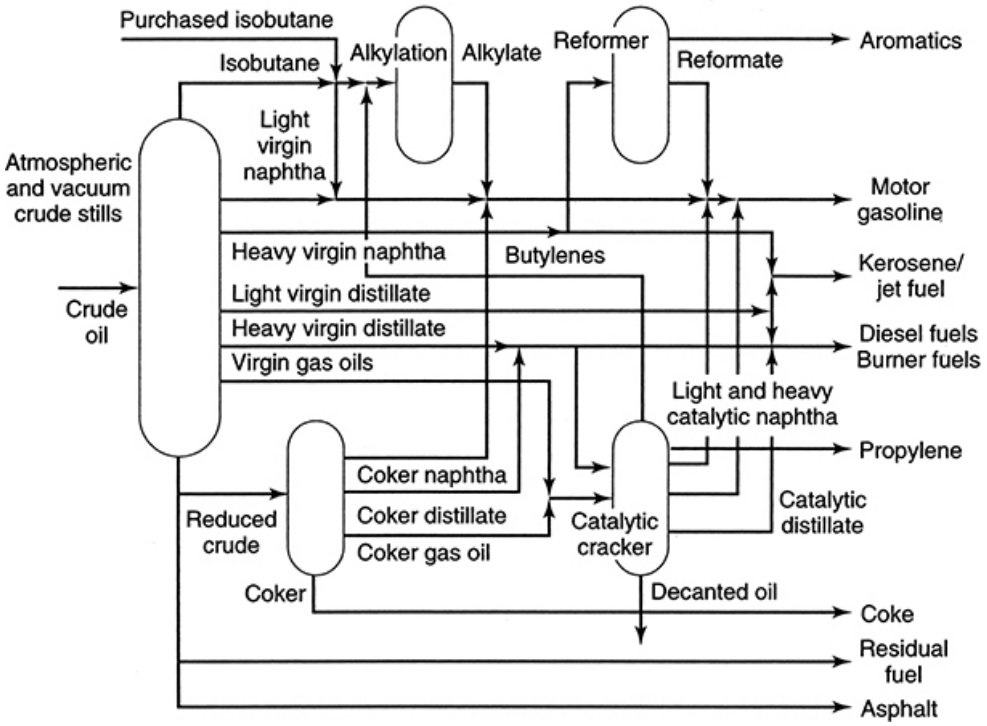


Figure 1.37 Schematic layout (simplified) of crude oil refinery process.[33]

Gasoline is a blend of hydrocarbons with boiling points ranging from about 25 to 200°C; diesel fuel is a blend of hydrocarbons with boiling points ranging from about 160 to 350°C. Chemical processing is used to convert one fraction into another to upgrade a given fraction so the refinery output meets each fuels' specifications. Alkylation is used to increase the fuel's molecular weight and knock resistance or octane number, [h] by adding alkyl radicals to gaseous hydrocarbon molecules to create branched paraffinic compounds (alkanes). Light olefin gases are reacted with isobutene in the presence of a catalyst. This process consumes relatively little energy. Catalytic cracking breaks up heavy oil molecules to produce a range of lighter products including aromatics and olefins for use in gasoline. The products of catalytic cracking go into high octane gasolines. The catalytic cracking reactions occur at high temperatures so considerable energy is consumed in this process. The reformer changes the molecular structure of specific streams such as naphtha to yield higher octane gasoline (e.g., conversion of paraffins into aromatic hydrocarbons, and straight chain hydrocarbons into branched hydrocarbons) in the presence of a catalyst. Considerable hydrogen is produced that is used elsewhere in the refinery.

The coker converts the heavy reduced crude fraction to naphtha and distillate fractions. The reduced crude is heated in an oven where the molecules undergo pyrolytic decomposition and recombination. While the average molecular weight of the fraction remains the same, a greater spectrum of components is produced. The heaviest component is coke, a solid largely carbonaceous material. [34]

Gasoline, or petrol, is the dominant SI engine fuel. It is a blend of light distillate hydrocarbons (paraffins, naphthenes, olefins, aromatics). Like all liquid hydrocarbons, it has a very high chemical energy density—energy per unit mass or volume. This chemical energy is stored in the carbon-hydrogen and carbon-carbon bonds and is released as thermal energy when these elements are oxidized by burning the fuel with air. In the United States, fuel requirements for SI engines are defined by the American Society for Testing and Materials in ASTM 4814. [35] The equivalent European Standard is EN 228. [36] The average molecular composition of a typical gasoline is close to C_7H_{13}; it includes hydrocarbon molecules containing between four and ten carbon atoms. The C:H atom ratio in commercial gasolines varies from about 1.6 to 2.2.

In terms of SI engine operation, the most critical gasoline properties are its volatility and its resistance to autoignition (or knock) during the latter part of the combustion process. A gasoline's volatility is characterized by its distillation curve, the volume fraction evaporated at atmospheric pressure as a function of temperature. Typically 10% of gasoline vaporizes below about 50°C (T_{10}), 50% vaporizes below 100°C (T_{50}), and 90% below 170°C (T_{90}). Winter gasolines are more volatile than summer gasolines. The fuel must contain enough highly volatile components to ensure rapid and low-emission cold starts. The lower temperature end of the distillation curve affects the fraction of the fuel injected into the port that vaporizes under these conditions. It also affects evaporative hydrocarbon emissions from the vehicle's fuel system. The upper end of the distillation curve is controlled to reduce hydrocarbon emissions and lubricating oil dilution with fuel.

A fuel's resistance to knock is defined by its octane rating. Two standard tests (see Sec. 9.6.3) define the *research octane number* and *motor octane number* of a fuel. The number displayed on gasoline pumps in the United States, the antiknock index, is the average of these. A typical regular (standard) gasoline would have a research octane number of 92, a motor

octane number of 82, with an antiknock index of 87. Historically, this antiknock index correlated the knock resistance behavior of fuels in engines in vehicles on the road. Modern engine antiknock requirements are better correlated by the research octane number, which is the fuel's octane number used in Europe. At least two quality gasolines are usually marketed: regular and one or more higher octane grades. Premium gasolines have higher antiknock ratings (up to 98 RON) and often higher levels of additives that are used to control deposit build-up tendencies in various parts of the engine and fuel system. Lead alkyl additives were once widely used to improve the antiknock resistance of gasolines. Since lead is highly toxic, and the catalytic converters used in engines' exhaust systems to reduce emissions (and the oxygen exhaust sensors used for modern engine control) are poisoned by lead, unleaded gasolines are now required in the developed world. While leaded gasoline is still available in some countries, their number is steadily decreasing.

Environmental authorities are imposing increasingly stringent regulations for fuels (gasoline and diesel) to ensure low hydrocarbon evaporative emissions and exhaust pollutant emissions. Thus *reformulated fuels* are now widely used. The main characteristics of reformulated gasolines are reduced vapor pressure (i.e., reduced low-end volatility), lower concentrations of aromatic compounds and benzene, and a lower final boiling point (T_{90}—temperature at which 90% of fuel is evaporated). In the United States, additives that inhibit deposit formation in the engine intake are also required. These reformulated gasolines usually contain oxygenated organic compounds [historically, methyl tertiary-butyl ether (MBTE), now–ethyl alcohol]. In some parts of the United States, these oxygenates are required in winter gasolines (at least 2.7% oxygen) to reduce engine carbon monoxide emissions. Year round, to encourage the use of biofuels, gasolines with up to 5, 10, or 15% ethanol can be marketed (amount depending on country or region). Both MBTE and ethanol have higher octane numbers than the base gasoline so they improve the fuels knock resistance. However, since MTBE can contaminate drinking water supplies, its use in gasolines has been discontinued. The sulfur levels in these clean unleaded gasolines are being reduced from historical levels [some 300 parts per million (ppm) by mass] to levels approaching 10 ppm. Sulfur is a catalyst poison that degrades the effectiveness of exhaust emission-control catalyst systems.

Diesel fuels contain many individual hydrocarbon compounds with boiling points ranging from about 180 to 370°C. As shown in Fig. 1.37, they are a primary product of the crude oil distillation process. To meet the growing demand for diesel fuel, refineries are adding increasing amounts of conversion products through cracking and coking of heavy oil. Diesel fuel is a mixture of paraffins, napthas, olefins, and aromatics; these have higher molecular weights (and different proportions) than these types of hydrocarbons in gasolines. The average molecular composition is $(CH_{1.8})_n$, and average molecular size is $C_{13}H_{24}$. The molecular weight range of the diesel hydrocarbons is about 170 to 200. The density of diesel fuel is important because fuel-injection systems are designed to deliver a specified *volume* of fuel whereas the combustion characteristics depend on the fuel/air *mass* ratio. The average diesel fuel density in Europe and the United States is about 0.84 kg/liter; the density, in other major geographic regions, varies between 0.81 and 0.86 kg/liter. Because the density of diesel fuel is higher than gasoline, the chemical energy content (heating value) *per unit volume* of fuel is about 10% higher. Other properties related to density are volatility and viscosity. Fuel volatility is defined by the distillation curve and the flash point. The distillation curve effectively defines the vaporization of the fuel inside the engine cylinder. The flash point is the minimum temperature to which the fuel must be heated to produce vapor that ignites in the presence of a flame. The viscosity of diesel fuel affects the fuel's performance in the fuel-injection system; it affects both the fuel pump and injector behavior, and the atomization process in the fuel injector nozzle holes. The low temperature flow characteristics of the fuel in the fuel system (especially the fuel filter) are a critical issue in regions with low winter temperatures where waxes can precipitate out.

Diesel fuel must have appropriate autoignition characteristics—that is, spontaneously ignite fast enough within the developing fuel sprays (see Sec. 1.8) to initiate combustion at the desired point in the engine cycle. This characteristic of diesel fuels is defined by the *cetane number*, which compares the autoignition characteristics of a diesel fuel with those of defined reference fuels (see Sec. 10.5.2). Typical cetane numbers are in the 40 to 55 range; the higher the cetane number, the easier (more rapid) is autoignition in the engine at around top center, just after injection starts. Fuel composition is adjusted to provide the appropriate cetane number; it can be

enhanced by ignition improving additives—active oxidizers such as alkyl nitrates. Diesel fuel property requirements are defined in ASTM D975. [37]

1.11.2 Alternative Fuels

Alternative fuels to the gasolines and diesel fuels produced from petroleum are in limited use in IC engines, and are being explored for expanded use in the future. Important driving forces are the need to reduce transportation's dependence on petroleum, and to reduce emissions of GHGs—especially CO_2. LPG and natural gas are in limited use as SI engine fuels in specific applications. Ethanol, produced from biomass, to date is used primarily as a gasoline blending agent. Alternative diesel fuels—for example, biodiesel (methyl esters made from rape seed or soybeans); also dimethyl ether (DME), Fischer-Tropsch gas-to-liquids fuels made from natural gas (or from coal)—are being produced and explored. Hydrogen is being examined as a possible longer-term SI engine fuel largely for its potential, once on the vehicles, to be a noncarbon-emitting energy carrier.

The more important properties of these alternative fuels, along with those of gasoline and diesel, are tabulated in App. D. An important property of these fuels relevant to their use in engines is their chemical energy content or heating value per unit volume of fuel-vapor/air mixture, which has just enough air for complete combustion (i.e., just enough for full oxidation of the fuel's carbon and hydrogen to CO_2 and H_2O—the stoichiometric mixture ratio). The basic objective of the SI engine's intake process is to fill each cylinder with such a mixture. In the diesel, this objective is met locally as the fuel is combusted. While the heating value of these different fuels *per unit mass of fuel* varies substantially, the differences in the composition of these fuels (and thus the amount of air each fuel requires for complete combustion) bring their heating values *per unit volume* of stoichiometric (chemically correct) mixture to surprisingly similar values. Thus SI engine outputs (i.e., maximum power per unit displaced volume with other parameters held fixed) over a wide range of fuels are closely comparable.

Note also that the combustion characteristics of these fuels when mixed with air in engines are similar (with some important differences in details—especially their anti-knock ratings). The exception is hydrogen which has a much higher flame speed than the hydrocarbon fuels.

Liquid petroleum gas , which consists of C_3 and C_4 hydrocarbons, is produced either by its removal from natural gas during the gas extraction process, or from the refining of crude oil. The fraction of LPG used in transportation varies widely (from a percent or so in the United States to significantly higher levels in some other countries), as does the composition of LPG. The dominant component is propane (C_3H_8): this can range from 30 to over 90%. Propylene, butanes, and ethane make up the remainder. In vehicles, LPG is stored as a compressed liquid at pressures between about 10 and 15 bar, in cylindrical on-board tanks, at ambient temperature. In most LPG fueling systems, the fuel is injected into the intake manifold in similar manner to multipoint injection gasoline systems. A common fuel rail and one injector valve for each cylinder inject the LPG (in liquid or gaseous form) into each intake manifold, continuously or intermittently. [13]

The components of LPG have higher octane ratings or knock resistance than gasoline, so higher engine compression ratios can be used. However, since the fuel is a gas with a lower molecular weight than gasoline, when mixed with air, the volume occupied by the fuel becomes more significant. LPG fuel therefore displaces more air than does gasoline vapor, and results in a modest loss in power. Air pollutant emissions from a LPG-fueled SI engine (prior to any catalyst in the exhaust) are usually lower than from gasoline-fueled engines.

Natural gas is a primary energy source and carrier. It is largely (80 to 98%) methane, CH_4. The remainder is ethane (1 to 8%), propane (up to 2%), with varying amounts of nitrogen (from a few to 10%). Thus it has a higher molar hydrogen:carbon ratio, so its CO_2 emissions on complete combustion are lower than those of gasoline and diesel by about 25%. However, methane itself is a GHG with a global warming potential about 20 times (per molecule) higher than CO_2. Natural gas leakage and unburned methane emissions would need to be carefully controlled if natural gas is used in transportation on a much larger scale.

Methane has a high octane number, or antiknock rating, though the other hydrocarbon species in natural gas reduce this somewhat depending on their relative amounts. Its combustion characteristics are comparable to those of gasoline.

Natural gas is, of course, a gas and thus storing adequate quantities storage on board the vehicle is a challenging problem. Storage as high-

pressure compressed gas (35 MPa) is expensive and requires a substantial volume even for limited vehicle range. At these conditions, the volumetric energy density (MJ/m ^{3}) of compressed natural gas is about one-quarter that of gasoline. Also, because it is a gas with low molecular weight (methane's is 16), when natural gas is mixed with air to combustible proportions, it displaces about 10% of the air (at fixed pressure and temperature) thus reducing the torque that can be produced per unit displaced cylinder volume. Similarly to LPG, natural gas is delivered to the engine intake manifold or intake port via a pressure regulator low pressure common rail, and injectors that inject natural gas intermittently into the intake. Because the fuel is a gas, mixture formation is easier to control since there are no liquid fuel drops and films to vaporize.

Alcohols, such as **ethanol** C_2H_6O and **methanol** CH_4O, can be used as SI engine fuels. Ethanol can be produced from biomass; methanol can be produced form natural gas. These alcohols contain oxygen, so their chemical energy content or heating value per unit mass is less than that of hydrocarbons (0.45 for methanol and 0.61 for ethanol). Heating values per unit mass of stoichiometric mixture are 5% less than gasoline.

These alcohols can be used as stand-alone fuels, or blended with gasoline at high or low fractions. These are designated E100 or M100 for ethanol or methanol alone. Blends with about 15% gasoline (e.g., E85 or M85) are used to broaden the volatility or vaporization characteristics of the single compound alcohol. Ethanol is the more practical of these two fuels. A disadvantage of methanol is its toxicity. Ethanol is used extensively as a transportation fuel in Brazil, where it is made from sugar cane. It is also made from corn (and can be more efficiently made from cellulosic biomass sources such as switchgrass and fast growing bushes and trees). Gasolines blended with up to 10% ethanol are used in the United States and elsewhere to both meet government requirements for oxygen content in "clean" lower-emission reformulated gasolines, and as an outlet for ethanol fuel.

Both ethanol and methanol have significantly higher antiknock or octane ratings than gasoline; the value depends on whether the fuels are tested or used as stand-alone fuels or in gasoline blends (see Table D4, App. D).

Hydrogen can be produced from natural gas, coal, biomass, and by electrolyzing water. Hydrogen for industrial use is currently largely made from steam reforming of natural gas. Storage of hydrogen on board vehicles

is a major challenge. As compressed gas, at 70 MPa (10,000 psi), about the maximum practical storage pressure, hydrogen has about one-third the energy density per unit volume of natural gas. Liquid hydrogen, at 20 K, has about one-fourth the energy density per unit volume of a hydrocarbon fuel. When mixed with air to stoichiometric proportions, the hydrogen in the fuel-air mixture occupies about 30% of the mixture volume, compared with less than 2% for gasoline vapor-air mixtures, reducing the power per unit displaced cylinder volume at constant conditions.

The combustion characteristics of hydrogen are substantially different to those of hydrocarbons, and thus gasoline. The laminar flame speed (a fundamental combustion characteristics of fuel-air mixtures) in hydrogen-air mixtures is several times higher than that of equivalent hydrocarbon-air mixtures due to the much faster diffusion of hydrogen species within the flame. Its flammability limits are broad (4 to 75% by volume at standard atmospheric conditions) and its self-ignition temperature is lower than that of gasoline. However, it has a higher knock resistance and octane number than do gasolines.

Several other fuel's pathways are potential alternatives to petroleum-based diesel fuel. Fuels made via the Fischer-Tropsch process from natural gas are being produced from natural gas reserves that are difficult to transport to the major natural gas markets. Fischer-Tropsch F-T fuels can also be made from coal. The F-T process catalytically combines CO and hydrogen, usually produced by reforming natural gas, to form CH_2 radicals that are combined to form larger hydrocarbon chains. These fuels are completely paraffinic, have very low levels of sulfur, and excellent autoignition characteristics (a high cetane number). They would thus be excellent diesel engine fuels.

Other potential diesel fuels are DME C_2H_6O, and biodiesel. DME is a gas at ambient pressure and temperature; it can be liquefied at about 5 bar pressure. It burns easily in a diesel engine, and its oxygen component helps inhibit the formation of soot and diesel smoke. Vegetable oils, such as rapeseed methyl ester and soybean methyl ester, can be processed to produce biodiesel fuels. Their potential attractiveness is that they are produced from biomass and thus are a possible low GHG emitting source of diesel fuel.

PROBLEMS

1.1 Describe the major functions of the following reciprocating engine components: piston, connecting rod, crankshaft, cams and camshaft, valves, intake, and exhaust manifolds.

1.2 Indicate on an appropriate sketch the different forces that act on the piston, and the direction of these forces, during the engine's expansion stroke with the piston, connecting rod, and crank in the positions shown in Fig. 1.1. Explain how the reciprocating engine geometry generates torque. Illustrate on a sketch.

1.3 List six important differences between the design and operating characteristics of spark-ignition and compression-ignition (diesel) engines. Start with the more fundamental of these.

1.4 Indicate the approximate crank angle at which the following events in the four-stroke and two-stroke internal combustion engine cycles occur on a line representing the full cycle (720° for the four-stroke cycle; 360° for the two-stroke cycle): bottom- and top-center crank positions, inlet and exhaust valve or port opening and closing, start of combustion process, end of combustion process, maximum cylinder pressure.

1.5 Explain briefly the following differences between a standard automobile spark-ignition engine and a truck diesel engine:

(a) Where the fuel is injected and why

(b) How the load is varied at fixed speed

(c) How the combustion process starts, develops, and ends

(d) How the fuels are different and why

(e) How the in-cylinder pressure varies as a function of crank angle (draw qualitatively the pressure traces for both engines in the same graph showing their relative magnitudes and when approximately the combustion starts and ends in each case).

1.6 The two-stroke cycle has twice as many power strokes per crank revolution as the four-stroke cycle. However, two-stroke cycle engine power outputs per unit displaced volume are less than twice the power output of an

equivalent four-stroke cycle engine at the same engine speed. Suggest reasons why this potential advantage of the two-stroke cycle is offset in practice.

1.7 Suggest reasons why multicylinder engines prove more attractive than single-cylinder engines once the total engine displaced volume exceeds a few hundred cubic centimeters.

1.8 The gasoline spark-ignition engine dominates the light-duty vehicle market in the United States. Briefly discuss the following:

(a) For light-duty vehicles, what other competing "prime mover" technologies might be attractive? Think both in terms of fuel used and the means by which the fuel's energy is converted to useful work.

(b) In what ways is each of these fuels or technologies better (or worse) than a gasoline spark-ignition engine? Think both in terms of a single vehicle's performance and the "big picture" resulting from having many such vehicles. You do not need to be quantitative.

(c) Now imagine that you are ready to buy your first car. Which of the factors you listed above are most important to you? Which "engine" and energy source would you choose?

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[a] The gas turbine is also, by this definition, an "internal combustion engine." Conventionally, however, the term is used for spark-ignition and compression-ignition engines. The operating principles of gas turbines are fundamentally different, and they are not discussed in this book.

[b] The conversion of the carbon in the biomass source to CO_2 is often regarded as "carbon neutral" since that carbon came from CO_2 in the atmosphere. This topic is the subject of ongoing research.

[c] In the remainder of the book, these terms will often be abbreviated by SI and CI, respectively.

[d] These crank positions are also referred to as top-dead-center (TDC) and bottom-dead-center (BDC).

[e] It is primarily for this reason that two-stroke SI engines are at a disadvantage because the lost fresh charge contains fuel and air.

[f] In practice, the intake and compression processes of a firing engine and a motored engine are not the same due to the presence of burned gases from the previous cycle under firing conditions.

[g] MBT timing was traditionally defined as the minimum spark advance for best torque. Since the torque first increases and then decreases as spark timing is advanced, the definition used here is more precise.

[h] See Sec. 9.6.3 .

CHAPTER 2

Engine Design and Operating Parameters

2.1 IMPORTANT ENGINE CHARACTERISTICS

In this chapter, the basic geometrical relationships and the parameters commonly used to quantify engine operation are developed. The characteristics important to an engine user are:

1. The engine's maximum performance over its full operating speed range
2. The engine's fuel consumption within this operating range and the cost of the required fuel
3. The engine's noise and air pollutant emissions within this operating range
4. The initial cost of the engine and its installation
5. The reliability and durability of the engine, its maintenance requirements, and how these affect engine availability and operating costs

These factors control total engine operating costs—usually the primary consideration of the user—and whether the engine in operation can satisfy environmental regulations. This book is concerned primarily with the performance, efficiency, and emissions characteristics of engines; the

omission of the other factors listed above does not, in any way, reduce their importance.

Engine performance is more precisely defined by:

1. The maximum torque available at each speed over the useful engine operating range
2. The maximum engine speed, and thus the maximum power
3. The range of speed and torque (or power) over which engine operation is satisfactory

The following definitions are commonly used to define maximum performance of the engine:

Maximum rated power . The highest power an engine is allowed to develop for short periods of operation.

Normal rated power . The highest power an engine is allowed to develop in continuous operation.

Rated speed . The crankshaft rotational speed at which rated power is developed.

2.2 GEOMETRICAL RELATIONSHIPS FOR RECIPROCATING ENGINES

Useful insights can be obtained from analyzing the geometry of the piston-in-cylinder, connecting rod, and crank mechanism. For example, the cylinder volume as a function of time effectively defines the in-cylinder gas density and amount of gas compression or expansion. The cylinder head, liner, and piston surface areas enter linearly into wall heat transfer calculations. Piston motion drives most engine gas flows, and piston acceleration at higher engine speeds plays a significant role in the periodic inertia forces that this reciprocating engine mechanism produces.

The following parameters define the basic geometry of a reciprocating engine (Fig. 2.1):

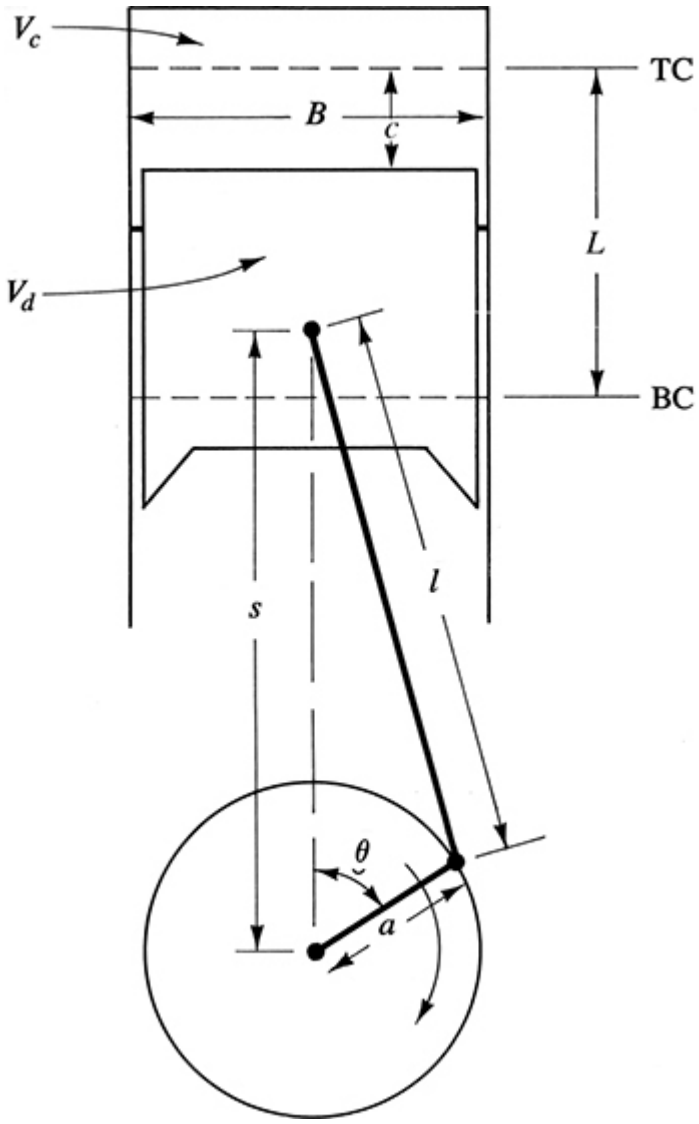


Figure 2.1 Geometry of cylinder, piston, connecting rod, and crankshaft where B = bore, L = stroke, l = connecting rod length, a = crank radius, θ = crank angle.

Compression ratio r_c:

$$r_c = \frac{\text{maximum cylinder volume}}{\text{minimum cylinder volume}} = \frac{V_d + V_c}{V_c} \tag{2.1}$$

where V_d is the displaced or swept volume and V_c is the clearance volume.
Ratio of cylinder bore to piston stroke:

$$R_{bs} = \frac{B}{L} \tag{2.2}$$

Ratio of connecting rod length to crank radius:

$$R = \frac{l}{a} \tag{2.3}$$

In addition, the stroke and crank radius are related by

$$L = 2a$$

Typical values of these parameters are: r_c = 9 to 12 for spark-ignition (SI) engines and r_c = 14 to 22 for compression-ignition (CI) engines; B/L = 0.8 to 1.2 for small- and medium-size engines, decreasing to about 0.5 for large slow-speed CI engines; R = 3 to 3.5 for small- and medium-size engines, increasing to 5 to 9 for large slow-speed CI engines.

The cylinder volume V at any crank position θ is the clearance volume plus the swept volume:

$$V = V_c + \frac{\pi B^2}{4} c \tag{2.4}$$

where c is the piston clearance from its top center position. It can be found from s, the distance between the crank axis and the piston pin axis (Fig. 2.1), which is given by

$$s = a\cos\theta + (l^2 - a^2\sin^2\theta)^{1/2} \tag{2.5}$$

The piston clearance height c, which is ($\ell + a$) − s, is given by

$$c = \frac{L}{2}\left[R + 1 - \cos\theta - (R^2 - \sin^2\theta)^{1/2}\right] \tag{2.6}$$

The angle θ, defined as shown in Fig. 2.1, is called the *crank angle*. Equation (2.4) with the above definitions can be rearranged:

$$\frac{V}{V_c} = 1 + \frac{1}{2}(r_c - 1)[R + 1 - \cos\theta - (R^2 - \sin^2\theta)^{1/2}] \tag{2.7}$$

The combustion chamber surface area A_s at any crank position θ is given by

$$A_s = A_{ch} + A_p + \pi Bc \tag{2.8}$$

where A_{ch} is the cylinder head surface area and A_p is the piston crown surface area. For flat-topped pistons, $A_p = \pi B^2/4$. Using Eq. (2.6) , Eq. (2.8) can be rearranged:

$$A_s = A_{ch} + A_p + \frac{\pi BL}{2}\left[R + 1 - \cos\theta - (R^2 - \sin^2\theta)^{1/2}\right] \tag{2.9}$$

An important characteristic engine speed is the *mean piston speed* $\overline{S}_p$:

$$\overline{S}_p = 2LN \tag{2.10}$$

where N is the rotational speed of the crankshaft. Mean piston speed is often a more appropriate parameter than crank rotational speed for correlating engine behavior as a function of speed. For example, gas-flow velocities in the intake and the cylinder all scale with $\overline{S}_p$. The *instantaneous piston velocity* S_p is obtained from

$$S_p = \frac{ds}{dt} \tag{2.11}$$

The piston velocity is zero at the beginning of the stroke, reaches a maximum near the middle of the stroke, and decreases to zero at the end of the stroke. Differentiation of Eq. (2.5) and substitution gives

$$\frac{S_p}{\overline{S}_p} = \frac{\pi}{2}\sin\theta\left[1 + \frac{\cos\theta}{(R^2 - \sin^2\theta)^{1/2}}\right] \tag{2.12}$$

Figure 2.2 shows how S_p, and the piston displacement, vary over each stroke for $R = 3.5$. Equation (2.12) can be approximated (with error less than 5%) by the equation

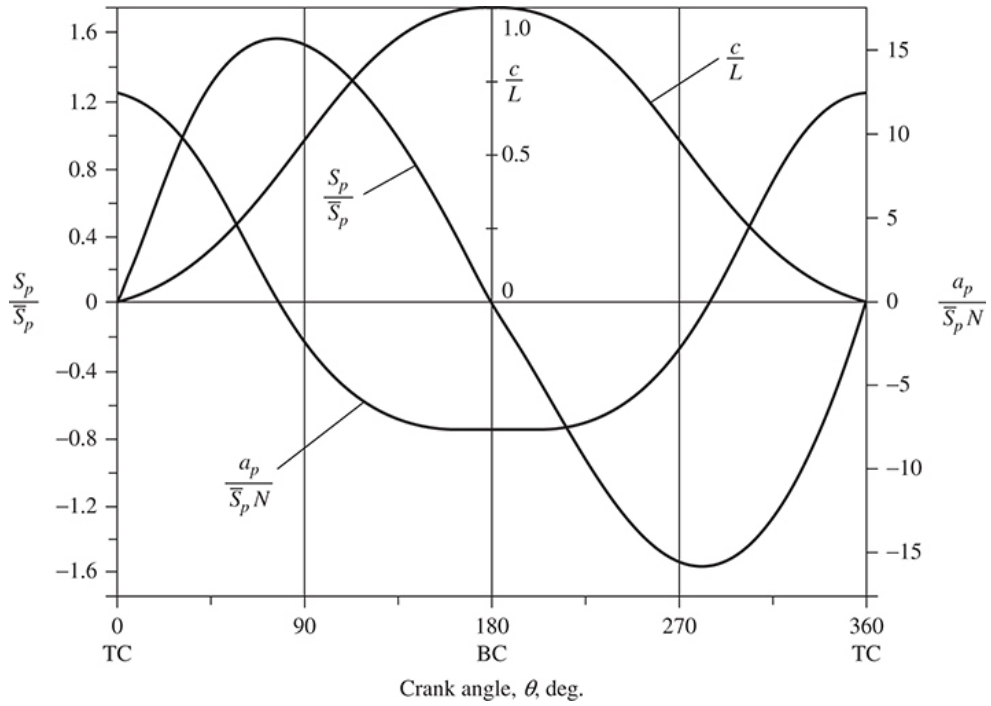


Figure 2.2 Piston displacement from top center TC, c/L (where L is the stroke), instantaneous piston speed/mean piston speed $S_p/\bar{S}_p$, and piston acceleration $a_p/(\bar{S}_p N)$, as a function of crank angle for $R = 3.5$.

$$\frac{S_p}{\bar{S}_p} = \frac{\pi}{2}\sin\theta\left(1 + \frac{1}{R}\cos\theta\right) \tag{2.13}$$

Resistance to gas flow into the engine cylinder limits the maximum mean piston speed to about 15 m/s, independent of engine size (the flow becomes choked). Automobile SI engines typically operate with maximum mean piston speeds in the range 14 to 20 m/s. Values between 10 and 14 are typical of diesel engines, where maximum speed is limited by the fuel-air mixing process.

The piston's acceleration is obtained by differentiating Eq. (2.12) or (2.13). The latter leads to this expression for a_p:

$$a_p = \pi^2 \bar{S}_p N\left(\cos\theta + \frac{1}{R}\cos 2\theta\right) \tag{2.14}$$

Figure 2.2 shows how $a_p/(\bar{S}_p N)$ varies with crank angle for one crankshaft revolution.

Note that the piston motion during each stroke in Fig. 2.2 is not as straightforward as it may appear. From TC to 90° after TC, the piston does not quite get to half stroke ($c/L = 0.5$), and the piston velocity S_p reaches a maximum at about 74° ATC (not 90° ATC). Only as ℓ/a becomes large (typically, for small- and mid-size engines, it is 3 to 3.5) does the piston motion during the second half of the downward (or upward) stroke become "symmetric" with the first half of the stroke.

In practical reciprocating engines, two modifications to this axisymmetric geometry are sometimes used: (1) piston pin offset and (2) crankshaft axis offset. The piston pin is frequently offset by about 1.5% of the bore toward the major thrust side of the liner. (The major thrust side is the left-hand side for the arrangement shown in Fig. 2.1 with clockwise crank rotation: it is the side of the liner against which the piston is pushed by the compression force in the connecting rod during the power or expansion stroke.) Piston pin offset reduces the piston impact with the liner close to top center as the piston transitions from the minor thrust side, (right-hand side in Fig. 2.1) to the major thrust side between the compression and expansion strokes. This piston transition with pin offset occurs earlier, before TC, when the forces acting on the piston are lower. Piston pin offset can be incorporated into engine geometry analysis. Its effect on cylinder volume is small (maximum error at mid-stroke is about 0.4%).

The crankshaft axis may be offset about 15% of the bore toward the major thrust side of the cylinder (to the left of the symmetric on-cylinder-axis position shown in Fig. 2.1). This reduces the sideways force the piston exerts on the cylinder liner during the expansion stroke and thus reduces engine friction. It also reduces the piston speed during the earlier part of the expansion stroke, effectively completing combustion sooner and improving engine efficiency. This change in the crankshaft axis location can also be incorporated into engine geometry analysis. Note that offsetting the crank axis changes the compression ratio slightly.

2.3 FORCES IN RECIPROCATING MECHANISM

The engine's reciprocating mechanism generates torque about the crankshaft axis through the force generated in the connecting rod. Figure 2.3 shows the forces acting on the piston, and in the connecting rod. Cylinder gas pressure pushes down on the piston crown; crankcase gas pressure acts on the piston underside. Friction acts to retard the piston motion via the friction force F on the piston ring and skirt surfaces. During the expansion stroke, the cylinder gas pressure force dominates, creating a compressive reaction force R_c in the connecting rod. An axial force balance, downward, on the piston yields

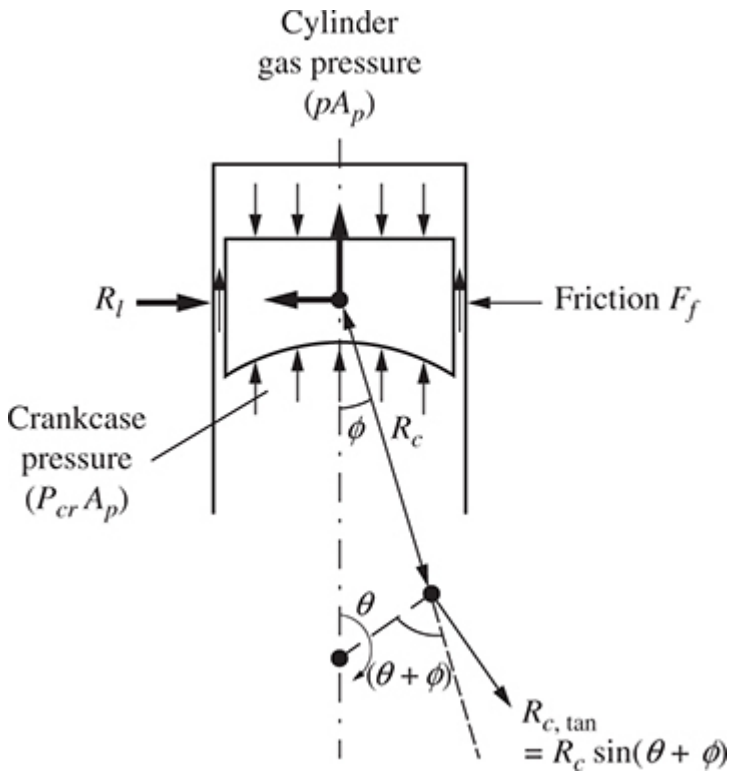


Figure 2.3 The forces acting on the piston and connecting rod, and the resultant crankshaft torque, during expansion stroke.

$$m_p a_p = \Sigma F_{\text{axial}} = (p - p_{cr})A_p - F - R_c \cos\phi \tag{2.15}$$

where m_p is the mass of the piston, rings, piston pin, and upper part of the connecting rod. [a] Note that $\ell \sin\phi = a \sin\phi$ from Fig. 2.1. If the cylinder gas

pressure and crankcase pressure are known, and the friction force can be estimated, with a_p obtained from Eq. (2.14) , the connecting rod force R_c can be calculated.

A transverse force balance relates the reaction force the liner exerts on the piston, R_l, to R_c:

$$R_l = R_c \sin\phi \tag{2.16}$$

This liner reaction force has an important effect on engine friction.

The component of the connecting rod force perpendicular to the crank radius generates torque about the crank axis. This torque, T, is therefore given by

$$T = aR_c \sin(\theta + \phi) \tag{2.17}$$

Using the fact that $a \sin\theta = \ell \sin\phi$, and expanding $\sin(\theta + \phi)$, yields

$$\sin(\theta+\phi) = \sin\theta\left(1-\frac{a^2}{l^2}\sin^2\theta\right)^{1/2} + \frac{a}{\ell}\sin\theta\cos\theta$$

Equation (2.15) , with Eq. (2.14) , allows the connecting rod force R_c as a function of crank angle to be calculated for a given engine geometry and speed, from known cylinder pressure, crankcase pressure, and friction force profiles. The torque each piston and connecting rod exert on the crankshaft, as a function of crank angle, can then be obtained from Eq. (2.17) and is given by

$$T = a\sin\theta\left[(p-p_{cc})A_p - F - m_p a_p\right]\left[1+\frac{a}{\ell}\cos\theta\frac{1}{\left(1-\frac{a^2}{\ell^2}\sin^2\theta\right)^{1/2}}\right] \tag{2.18}$$

Figure 2.4 illustrates how the different time-varying forces on the piston, and piston motion, combine to produce an oscillating torque at the crankshaft through the four-stroke cycle of a given cylinder in an SI engine. Note that a/ℓ is about 0.3, so the effect of the a^2/ℓ^2 term is small. At low engine speeds, the piston inertia force $m_p a_p$ is small relative to gas pressure forces. At high

engine speeds, the inertia forces are much more important (see Sec. 15.7.2). The relative importance of the piston friction forces depends on engine load —that is, the magnitude of the cylinder pressure force—and speed. Note that the torque pulse each cycle rises from zero at TC to a maximum slightly after the crank angle of peak pressure, and decays to close to zero by mid-expansion stroke. For a multicylinder engine, the time-varying crankshaft torque is obtained by superposing curves like those for a single cylinder in Fig. 2.4, with an appropriate phase difference (720°/ n_c, where n_c is the number of cylinders).

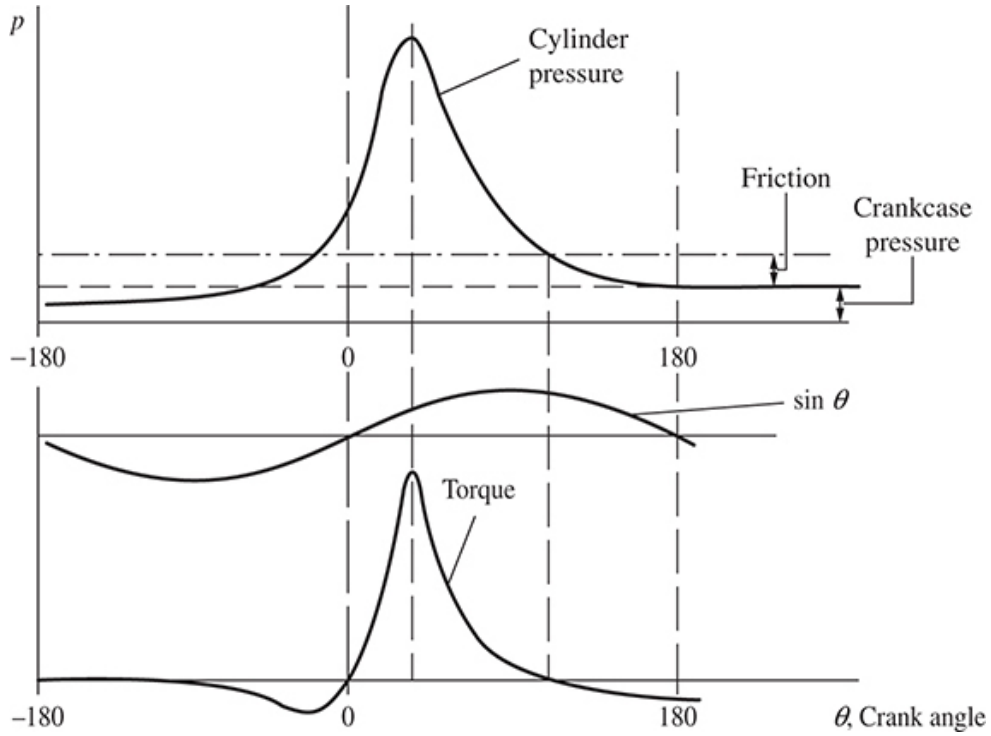


Figure 2.4 Forces acting on the piston (cylinder pressure, crankcase pressure, friction), and the crankshaft torque, as a function of crank angle for an SI engine at low engine speed. At high engine speed, the piston inertia force $m_p a_p$ is significant.

2.4 BRAKE TORQUE AND POWER

The torque an engine delivers through its driveshaft is normally measured with a dynamometer. The engine is mounted on a test bed and the shaft is

connected to the dynamometer rotor. Figure 2.5 illustrates the operating principle of a dynamometer. The rotor is coupled electromagnetically, hydraulically, or by mechanical friction to a stator, which is supported in low friction bearings. The stator is balanced with the rotor stationary. The torque exerted on the stator with the rotor turning is measured by balancing the stator with weights, springs, or pneumatic means. The dynamometer usually has sufficient inertia that this procedure with a multicylinder engine sums the torque contributions from each cylinder, and measures the time-averaged torque.

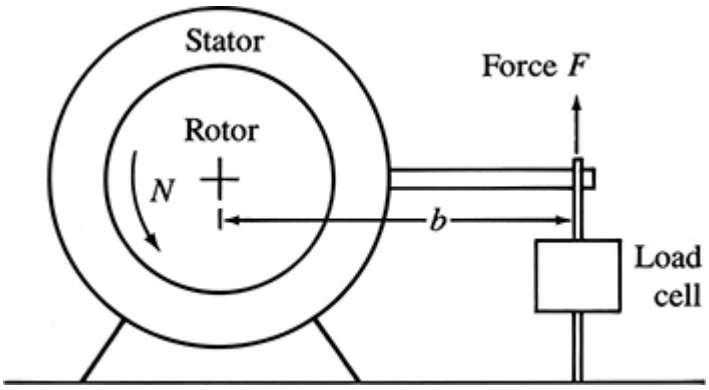


Figure 2.5 Schematic of principle of operation of dynamometer.

Using the notation in Fig. 2.5, if the average torque exerted by the engine is T

$$T = Fb \tag{2.19}$$

The power P delivered by the engine and absorbed by the dynamometer is the product of torque and angular speed:

$$P = 2\pi NT \tag{2.20a}$$

where N is the crankshaft rotational speed. In SI units

$$P(\text{kW}) = 2\pi N(\text{rev/s})T(\text{N}\cdot\text{m})\times 10^{-3} \tag{2.20b}$$

or in U.S. units

$$P(\text{hp}) = \frac{N(\text{rev/min})T(\text{lbf}\cdot\text{ft})}{5252} \qquad (2.20c)$$

The value of engine power measured as described above is called the *brake power* P_b. This power is the usable power delivered by the engine to the load—in this case, a "brake."

Torque is a measure of an engine's ability to do work; power is the rate at which work is done. These are important practical parameters, and their maximum values define an engine's performance. *Load* (e.g., mid load) is often used to denote where in an engine's torque range, zero to maximum, the engine is operating.

2.5 INDICATED WORK PER CYCLE

Work is produced inside each engine cylinder as gas pressure pushes the piston downward during the expansion stroke. Pressure data for the gas in the cylinder over the operating cycle of the engine can be used to calculate this work transfer from the gas to the piston. The cylinder pressure and corresponding cylinder volume throughout the engine cycle can be plotted on a *p-V* diagram as shown in Fig. 2.6. The *indicated work per cycle* $W_{c,i}$ [b] (per cylinder) is obtained by integrating around the curve to obtain the area enclosed on the diagram:

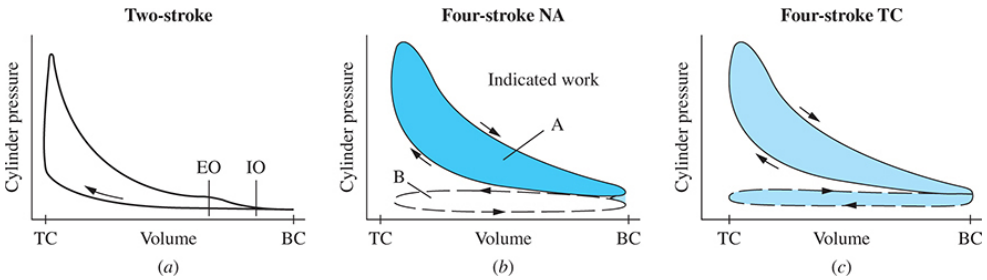


Figure 2.6 Examples of *p-V* diagrams illustrating the indicated work per cycle, for (*a*) a two-stroke cycle engine; (*b*) a naturally-aspirated four-stroke cycle engine at part load; (*c*) a turbocharged four-stroke cycle engine at high load.

$$W_{c,i} = \oint p\, dV \tag{2.21}$$

With two-stroke cycles (Fig. 2.6 *a*), the application of Eq. (2.21) is straightforward.[1]

With the addition of inlet and exhaust strokes for the four-stroke cycle, two definitions of indicated output are in common use. These are:

1. *Gross indicated work per cycle* $W_{c,\mathrm{ig}}$: Work delivered to the piston over the compression and expansion strokes only.
2. *Net indicated work per cycle* $W_{c,\mathrm{in}}$: Work delivered to the piston over the entire four-stroke cycle.

Figures 2.6 *b* and *c* show *p-V* diagrams for a naturally-aspirated and a turbocharged four-stroke cycle engine. The blue-colored areas in each figure show the commonly used definition of indicated work per cycle for these two types of engines. The work transfer between the piston and the cylinder gases during the inlet and exhaust strokes is called the *pumping work* W_p (see Chaps. 5 and 13). This pumping work transfer will be *to* the in-cylinder gases when the cylinder pressure during the intake stroke is less than the cylinder pressure during the exhaust stroke (area B in Fig. 2.6 *b*). This is the situation with naturally-aspirated engines. The pumping work transfer will be *from* the in-cylinder gases to the piston if the exhaust stroke pressure is lower than the intake pressure, which is normally the case with highly loaded turbocharged engines (Fig. 2.6*c*). Pumping work transfer to the in-cylinder gases reduces the work transferred to the crankshaft, and constitutes one component of engine friction (this work is dissipated pumping the exhaust gas out and pulling fresh charge into the cylinder). [c] Thus the gross indicated work per cycle (i.e., $\int pdV$ over the compression and expansion strokes, area A in Fig. 2.6*b*) is often used with naturally-aspirated engines since it represents the maximum amount of work produced inside the engine. With turbocharged engines where the pumping work can be positive (i.e., to the piston) and therefore adds to the positive work transfer produced during the compression and expansion strokes, the net indicated work per cycle definition—$\int pdV$ over the full four-stroke cycle—is usually used. Note that with turbocharged engine operation there are still frictional gas-flow losses during the exhaust and intake strokes. However, at higher engine loads, with

turbocharged engines, the difference between the compressor outlet pressure and turbine inlet pressure is often sufficiently large to more than offset these losses.

Increasingly, the net definition of imep is being used to quantify the work transferred to the piston because the integral of Eq. (2.21) over the full two revolutions—four strokes—better represents the impacts of the in-cylinder processes per cycle. The increasing use of variable valve control in engines makes the separation between energy conversion and transfer processes during the compression and expansion strokes, and the gas exchange processes, more complex. Then the friction work [and friction mean effective pressure (fmep)] is the sum of the mechanical "rubbing" friction and the work required to drive the essential engine accessories. Figure 2.7 shows how these indicated and brake definitions of work per cycle (and mep) fit with the reciprocating piston-in-cylinder engine geometry and work transfers.

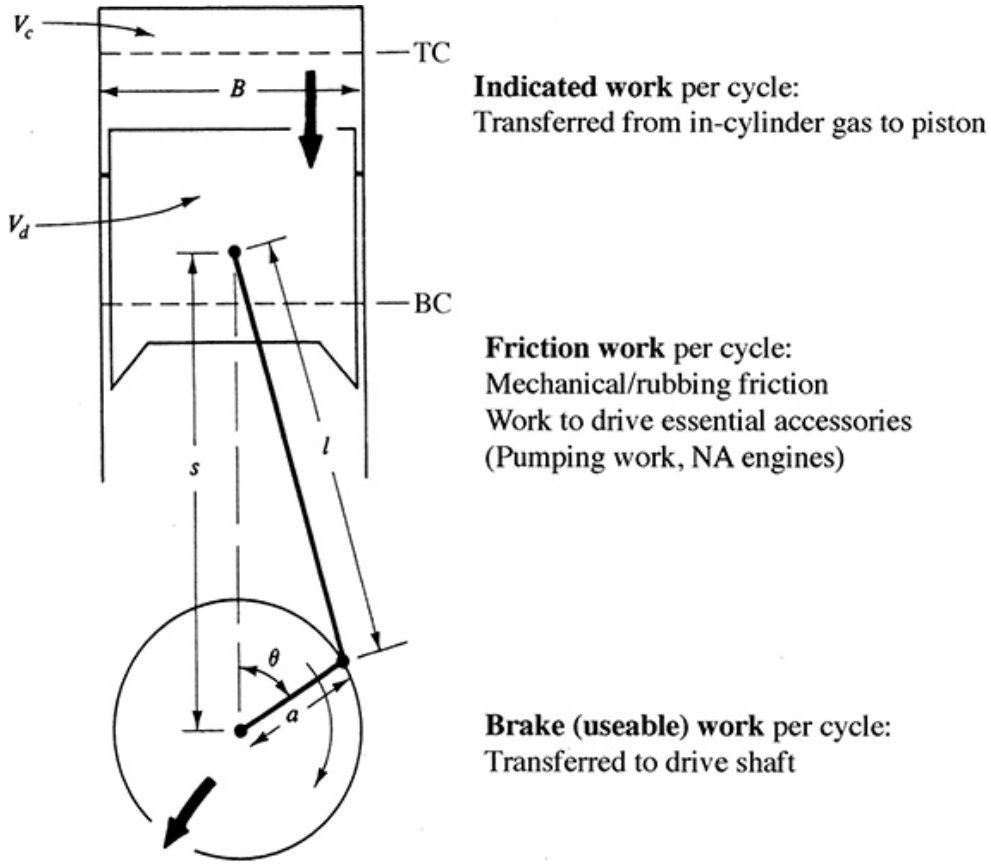


Figure 2.7 Schematic of how indicated work, friction work, and brake work transfers relate to the reciprocating engine geometry.

The power per cylinder is related to the indicated work per cycle by

$$P_i = \frac{W_{c,i} N}{n_R} \tag{2.22}$$

where n_R is the number of crank revolutions for each power stroke. For four-stroke cycles, n_R equals 2; for two-stroke cycles, n_R equals 1. This power is the indicated power; that is, the rate of work transfer from the gas within the cylinder to the piston. It differs from the brake power by the power absorbed (dissipated) in overcoming engine friction, driving engine accessories, and (in the case of naturally-aspirated engines) the pumping power. In discussing indicated quantities of the four-stroke cycle engine, such as work per cycle or power, the definition used for "indicated" (i.e., gross or net) *should always be explicitly stated*.

The indicated work per cycle (or power) represents the sum of the useful work (or power) available at the drive shaft plus the work (or power) required to overcome all the engine's frictional losses. The standard engine test codes define procedures for measuring brake power and friction power (the friction power test provides a close approximation to the total lost power in the engine). [2] Thus the sum of brake power and friction power provides an alternative way of estimating indicated power or work per cycle. The terms *brake* and *indicated* are used to describe other parameters such as mep, specific fuel consumption, and specific emissions (see the following sections) in a manner similar to that used for work per cycle and power.

2.6 MECHANICAL EFFICIENCY

In naturally-aspirated engines, a part of the (gross) indicated work per cycle or power is used to expel exhaust gases and induct fresh charge. In all engines, a portion of the work per cycle is used to overcome the friction of the bearings, pistons, and other mechanical components of the engine, and to drive the essential engine accessories. All of these power requirements are grouped together and called *friction power* P_f. [d] Thus,

$$P_i = P_b + P_f \tag{2.23}$$

Friction power is difficult to determine accurately. One common approach is to drive or motor the engine with a dynamometer (i.e., operate the engine without firing it) and measure the power that has to be supplied by the dynamometer to overcome *all* the frictional losses. The engine speed, throttle setting, oil and water temperatures, and ambient conditions are kept the same in the motored test as under firing conditions. The major sources of inaccuracy with this method are that gas pressure forces on the piston and rings are lower in the motored test than when the engine is firing, the oil temperatures on the cylinder wall are also lower under motoring conditions, and the motored exhaust gas (air) is more dense than the burned exhaust gas when firing. These differences between firing engine friction and this measured motoring friction both add and subtract: it turns out that the difference between motored and fired engine friction is modest.

The ratio of the brake (or useful) power delivered by the engine to the indicated power is called the *mechanical efficiency* η_m. It is a measure of the relative importance of engine friction.

$$\eta_m = \frac{P_b}{P_{ig}} = 1 - \frac{P_f}{P_{ig}} \tag{2.24}$$

Since in naturally-aspirated engines, the friction power includes the power required to pump gas into and out of the engine, mechanical efficiency in such SI engines depends on throttle position as well as engine design details and engine speed. Typical values of mechanical efficiency for a modern automotive engine at wide-open or full throttle are close to 90% at speeds below about 1800 to 2400 rev/min (30 to 40 rev/s), decreasing to 75% at the engine's maximum rated speed. As the engine is throttled, mechanical efficiency decreases, eventually to zero at idle operation where the brake output is negligible.

2.7 MEAN EFFECTIVE PRESSURE

While torque is an important measure of a particular engine's ability to do work, it depends on engine size. A more useful relative engine performance measure is obtained by dividing the work per cycle by the cylinder volume displaced per cycle. The parameter so obtained has units of force per unit area and is called the *mean effective pressure*. Since, from Eq. (2.22) ,

$$\text{Work per cycle} = \frac{Pn_R}{N}$$

where n_R is the number of crank revolutions for each power stroke per cylinder (two for four-stroke cycles; one for two-stroke cycles), then

$$\text{mep} = \frac{Pn_R}{V_d N} \tag{2.25a}$$

In SI and U.S. units, respectively,

$$\text{mep(kPa)} = \frac{P(\text{kW})n_R \times 10^3}{V_d(\text{dm}^3)N(\text{rev/s})} \tag{2.25b}$$

$$\text{mep(lb/in}^2) = \frac{P(hp)n_R \times 396{,}000}{V_d(\text{in}^3)N(\text{rev/min})} \tag{2.25c}$$

Both indicated, brake, and friction mep can be calculated from the indicated, brake, and friction work per cycle or power. Brake mep can also be expressed in terms of torque using Eq. (2.20) : effectively it is the volume normalized torque, multiplied by 4π for four-stroke cycle engines.

$$\text{bmep(kPa)} = \frac{6.28n_R T(\text{N}\cdot\text{m})}{V_d(\text{dm}^3)} \tag{2.26a}$$

or

$$\text{bmep(lb/in}^2) = \frac{75.4n_R T(\text{lb}_f\cdot\text{ft})}{V_d(\text{in}^3)} \tag{2.26b}$$

Best brake mep values of good engine designs of a given type of engine, at maximum torque and maximum power, are well established, and are essentially the same over a wide range of engine sizes. Thus, the actual bmep that a particular engine develops can be compared with this norm, and the effectiveness with which the engine designer has used the engine's displaced volume can be assessed. Also, for design calculations, the engine displacement required to provide a given torque or power, at a specified speed, can be estimated by assuming appropriate values for bmep for that particular type of engine.

Typical best values for bmep are as follows: For naturally-aspirated four-stroke SI engines, maximum values are in the range of 1050 to 1250 kPa at the engine speed where maximum torque is obtained (at a mean piston speed of about 11 m/s). Two-valve-per-cylinder engines are at the low end of this range, four-valve-per-cylinder engines with variable valve timing at the high end. At the maximum rated power, bmep values are some 10% lower than these bmep values corresponding to maximum torque. For turbocharged automotive SI engines, the maximum bmep is in the 1800 to 2400 kPa range. For naturally-aspirated four-stroke diesels, the maximum bmep is in the 800 to 900 kPa range, with bmep at the maximum rated power of about 750 kPa. Turbocharged four-stroke diesel maximum bmep values are typically in the range of 1600 to 2400 kPa with the higher values for turbocharged after-cooled engines. At maximum rated power, bmep is between about 1400 and 2200 kPa. Two-stroke cycle SI engines have maximum bmep values of 400 to 750 kPa (small single-cylinder engines at the low end, larger multicylinder engines at the high end). Larger two-stroke cycle diesels have comparable performance to four-stroke cycle engines. Large low-speed marine and power-generating two-stroke cycle engines can achieve bmep values of about 1750 kPa. [e]

An example of how the above engine performance parameters can be used to initiate an engine design is given below.

EXAMPLE 2.1

A four-cylinder four-valve-per-cylinder naturally-aspirated automotive SI engine is being designed to provide a maximum torque of 195 N·m in the mid-speed range (~4000 rev/min). Estimate the required engine

displacement, bore, and stroke, and the maximum brake power the engine will deliver.

Equation (2.26 *a*) relates torque and mep. Assume that 1200 kPa is an appropriate value for bmep at the maximum engine torque point for this type of engine. Equation (2.26 *a*) gives

$$V(\mathrm{dm}^3) = \frac{6.28 n_R T_{\max}(\mathrm{N\cdot m})}{\mathrm{bmep}_{\max}(\mathrm{kPa})} = \frac{6.28 \times 2 \times 195}{1200} = 2\,\mathrm{dm}^3$$

For a four-cylinder engine, the displaced volume, bore, and stroke are related by

$$V_d = 4 \times \frac{\pi}{4} B^2 L$$

Assume $B = L$; this gives $B = L = 86$ mm.

The maximum rated engine speed can be estimated from an appropriate value for the maximum mean piston speed, 17 m/s (see Sec. 2.2):

$$\bar{S}_{p\max} = 2LN_{\max} \rightarrow N_{\max} = 17/(2 \times 0.086) = 99\ \mathrm{rev/s}\ (5900\ \mathrm{rev/min})$$

The maximum brake power can be estimated from the typical bmep value at maximum power, 1080 kPa (about 10% less than 1200 kPa), using Eq. (2.25b) :

$$P_{\max}(\mathrm{kW}) = \frac{\mathrm{bmep(kPa)} V(\mathrm{dm}^3) N_{\max}(\mathrm{rev/s})}{N_R \times 10^3} = \frac{1080 \times 2 \times 99}{2 \times 10^3} = 107\ \mathrm{kW}$$

2.8 SPECIFIC FUEL CONSUMPTION AND EFFICIENCY

In engine tests, the fuel consumption is measured as a flow rate—mass flow per unit time $\dot{m}_f$. A more useful parameter is the *specific fuel consumption* (sfc)—the fuel flow rate per unit power output. It is a normalized parameter

(i.e., independent of engine size), and measures how efficiently an engine is using the fuel supplied to produce work at a specific operating condition:

$$\text{sfc} = \frac{\dot{m}_f}{P} \tag{2.27}$$

With units,

$$\text{sfc(mg/J)} = \frac{\dot{m}_f\,(\text{g/s})}{P(\text{kW})} \tag{2.28a}$$

or

$$\text{sfc(g/kW}\cdot\text{h)} = \frac{\dot{m}_f\,(\text{g/h})}{P(\text{kW})} = 608.3\ \text{sfc}(\text{lb}_\text{m}/\text{hp}\cdot\text{h}) \tag{2.28b}$$

or

$$\text{sfc}(\text{lb}_\text{m}/\text{hp}\cdot\text{h}) = \frac{\dot{m}_f\,(\text{lb}_\text{m}/\text{h})}{P(\text{hp})} = 1.644\times10^{3}\text{sfc(g/kW}\cdot\text{h)} \tag{2.28c}$$

Low values of sfc are obviously desirable. For SI engines, typical best values of brake-specific fuel consumption are about 65 μg/J = 235 g/kW·h = 0.41 lb $_m$/hp·h. For CI engines, best values are lower, and in larger engines can go below 50 μg/J = 180 g/kW·h = 0.29 lb $_m$/hp · h.

The specific fuel consumption has units. A dimensionless parameter that relates the desired engine output (work per cycle or power) to the necessary input (fuel flow) has more fundamental value. The ratio of the work produced per cycle to the amount of fuel chemical energy supplied per cycle that can be released in the combustion process is commonly used for this purpose. It is a measure of the engine's efficiency. The fuel energy supplied that can be released by combustion is given by the mass of fuel supplied to the engine per cycle times the heating value of the fuel. The heating value of a fuel, Q_{HV}, defines its chemical energy content. It is determined in a standardized test procedure in which a known mass of fuel is fully burned with air, and the chemical energy released by the combustion process is absorbed as thermal

energy in a calorimeter as the combustion products cool down to their original temperature.

This measure of an engine's "efficiency," called the *fuel conversion efficiency*[f] η_f, is given by

$$\eta_f = \frac{W_c}{m_f Q_{HV}} = \frac{(Pn_R/N)}{(\dot{m}_f n_R/N)Q_{HV}} = \frac{P}{\dot{m}_f Q_{HV}} \tag{2.29}$$

where m_f is the mass of fuel inducted per cycle. Substitution for $P/\dot{m}_f$ from Eq. (2.27) gives

$$\eta_f = \frac{1}{\text{sfc}\, Q_{HV}} \tag{2.30a}$$

or with units

$$\eta_f = \frac{1}{\text{sfc(mg/J)}Q_{HV}\,\text{(MJ/kg)}} \tag{2.30b}$$

$$\eta_f = \frac{3600}{\text{sfc}\left(\text{g/kW}\cdot\text{h}\right)Q_{HV}\left(\text{MJ/kg}\right)} \tag{2.30c}$$

$$\eta_f = \frac{2545}{\text{sfc}(\text{lb}_\text{m}/\text{hp}\cdot\text{h})Q_{HV}\,(\text{Btu/lb}_\text{m})} \tag{2.30d}$$

Typical heating values for the commercial hydrocarbon fuels used in engines are in the range of 42 to 44 MJ/kg (18,000 to 19,000 Btu/lb $_m$). Thus, specific fuel consumption is inversely proportional to fuel conversion efficiency for normal hydrocarbon fuels.

Note that the fuel energy supplied to the engine per cycle is not fully released as sensible or thermal energy in the combustion process because the actual combustion process is incomplete. When enough air is present in the cylinder to oxidize the fuel completely, almost all (about 95% or more) of this fuel energy supplied is transferred as thermal energy to the working fluid. When insufficient air is present to oxidize the fuel completely, lack of

oxygen prevents this supplied fuel energy from being fully released. This topic is discussed in more detail in Secs. 3.5 and 4.9.4.

2.9 AIR/FUEL AND FUEL/AIR RATIOS

In engine testing, both the air mass flow rate $\dot{m}_a$ and the fuel mass flow rate $\dot{m}_f$ are normally measured. The ratio of these flow rates is useful when defining engine operating conditions:

$$\text{Air/fuel ratio } (A/F) = \frac{\dot{m}_a}{\dot{m}_f} \tag{2.31}$$

$$\text{Fuel/air ratio } (F/A) = \frac{\dot{m}_f}{\dot{m}_a} \tag{2.32}$$

The normal operating range for a conventional SI engine using gasoline fuel is $12 \leq A/F \leq 15$ ($0.067 \leq F/A \leq 0.083$); for CI engines with diesel fuel, it is $20 \leq A/F \leq 80$ ($0.013 \leq F/A \leq 0.05$).

2.10 VOLUMETRIC EFFICIENCY

The intake system—the air filter, intake duct, throttle plate (in an SI engine), intake manifold, intake port, intake valve—restricts the amount of air that an engine of given displacement can induct. The parameter used to measure the effectiveness of an engine's induction process is the *volumetric efficiency* η_v. Volumetric efficiency is only used with four-stroke cycle engines that have a distinct induction process. It is defined as the volume flow rate of air into the intake system (at defined air conditions) divided by the rate at which volume is displaced by the piston:

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,i} V_d N} \tag{2.33a}$$

where $\rho_{a,i}$ is the inlet air density. An alternative equivalent definition for volumetric efficiency is

$$\eta_v = \frac{m_a}{\rho_{a,i} V_d} \tag{2.33b}$$

where m_a is the mass of air inducted into the cylinder per cycle.

With naturally-aspirated engines, the inlet density is usually taken to be atmosphere air density (in which case η_v measures the air-flow performance of the entire intake system), or may be taken as the air density in the inlet manifold (in which case η_v measures the air-flow performance of the inlet port and valve only). Typical maximum values of η_v for naturally-aspirated engines are about 90%. Volumetric efficiency is discussed more fully in Sec. 6.2 . It is not normally used with turbocharged engines.

Two-stroke cycle engines, where intake and exhaust processes overlap during scavenging, use different parameters to quantify the cylinder charging process (see Sec. 6.6.2).[1]

2.11 SPECIFIC POWER, SPECIFC WEIGHT, AND SPECIFIC VOLUME

The *specific power*

$$\text{Specific power} = P_{b,\max}/V_d \tag{2.34}$$

is often used as a normalized measure of an engine's maximum power. Here V_d is the total engine displaced volume. Typical best values for SI engines are 40 to 65 kW/liter (lower end, two valves per cylinder; higher end, four valve with variable valve control), about 80 kW/liter for turbocharged SI engines, about 30 kW/liter for naturally-aspirated truck diesels, and about 40 to 50 kW/liter for turbocharged diesels (higher values for passenger car diesels).

Engine weight and bulk volume for a given maximum power are important in many applications. Two parameters useful for comparing these attributes

from one engine to another are:

$$\text{Specific weight} = \frac{\text{engine weight}}{\text{maximum power}} \tag{2.35}$$

$$\text{Specific volume} = \frac{\text{engine volume}}{\text{maximum power}} \tag{2.36}$$

For these parameters to be useful in engine comparisons, a consistent definition of what components and auxiliaries are included in the term *engine* must be adhered to. These parameters indicate the effectiveness with which the engine designer has used the engine materials and packaged the engine components.

2.12 CORRECTION FACTORS FOR POWER AND VOLUMETRIC EFFICIENCY

The pressure, humidity, and temperature of the ambient air inducted into an engine, at a given engine speed, affect the air mass flow rate and the power output. Correction factors are used to adjust measured wide-open-throttle power and volumetric efficiency values to standard atmospheric conditions to provide a more accurate basis for comparisons between engines. Typical standard ambient conditions used are:

Dry air pressure	Water vapor pressure	Temperature
736.6 mmHg	9.65 mmHg	29.4°C
29.00 inHg	0.38 inHg	85°F

The basis for the correction factor for maximum power and torque is the equation for one-dimensional steady compressible air flow through an orifice or flow restriction of effective area A_E (see App. C). The flow resistances in the intake system, port, and valve effectively act as such a flow restriction. Thus

$$\dot{m} = \frac{A_E p_0}{\sqrt{RT_0}} \left\{ \frac{2\gamma}{\gamma - 1} \left[\left(\frac{p}{p_0} \right)^{2/\gamma} - \left(\frac{p}{p_0} \right)^{(\gamma+1)/\gamma} \right] \right\}^{1/2} \tag{2.37}$$

In deriving this equation, it has been assumed that air is an ideal gas with gas constant R and that the ratio of specific heats ($c_p / c_v = \gamma$) is a constant; p_0 and T_0 are the total pressure and temperature upstream of the restriction and p is the pressure at the throat of the restriction.

If, in the engine, p / p_0 is assumed constant at wide-open throttle, then for a given intake system and engine, the mass flow rate of dry air $\dot{m}_a$ at higher engine speeds varies as

$$\dot{m}_a \propto \frac{p_0}{\sqrt{T_0}} \tag{2.38}$$

For mixtures containing the proper amount of fuel to use all the air available (and thus provide maximum power), the indicated power at full throttle P_i will be proportional to $\dot{m}_a$, the *dry* air-flow rate. Thus if

$$P_{i,s} = C_F P_{i,m} \tag{2.39}$$

where the subscripts s and m denote values at the standard and measured conditions, respectively, the correction factor C_F is given by

$$C_F = \frac{p_{s,d}}{p_m - p_{v,m}} \left(\frac{T_m}{T_s} \right)^{1/2} \tag{2.40}$$

where

$p_{s,d}$ = standard dry-air absolute pressure
p_m = measured ambient-air absolute pressure
$p_{v,m}$ = measured ambient-water vapor partial pressure
T_m = measured ambient temperature, K
T_s = standard ambient temperature, K

The maximum *brake* power is corrected by using Eq. (2.40) to correct the *indicated* power, and making the assumption that friction power is unchanged. Thus

$$P_{b,s} = C_F P_{i,m} - P_{f,m} \tag{2.41}$$

Volumetric efficiency is proportional to $\dot{m}_a / \rho_a$ [see Eq. (2.33)]. Since ρ_a is proportional to p/T, the correction factor for volumetric efficiency, C'_F is

$$C'_F = \frac{\eta_{v,s}}{\eta_{v,m}} = \left(\frac{T_s}{T_m}\right)^{1/2} \tag{2.42}$$

2.13 SPECIFIC EMISSIONS AND EMISSIONS INDEX

Levels of emissions of oxides of nitrogen (nitric oxide, NO, and nitrogen dioxide, NO_2, usually grouped together as NO_x), carbon monoxide (CO), unburned hydrocarbons (HC), and particulates are important engine operating characteristics.

The concentrations of gaseous emissions in the engine exhaust gases are usually measured in parts per million or percent by volume (which corresponds to the mole fraction multiplied by 10^6 or by 10^2, respectively). Normalized indicators of emissions levels are also useful, however, and two of these are in common use. *Specific emissions* are the mass flow rates of pollutant per unit power output:

$$\text{sNO}_x = \frac{\dot{m}_{\text{NO}_x}}{P} \tag{2.43a}$$

$$\text{sCO} = \frac{\dot{m}_{\text{CO}}}{P} \tag{2.43b}$$

$$\text{sHC} = \frac{\dot{m}_{\text{HC}}}{P} \tag{2.43c}$$

$$\text{sPart} = \frac{\dot{m}_{\text{part}}}{P} \tag{2.43d}$$

Indicated and brake-specific emissions can be defined. Units in common use are μg/J, g/kW·h, and g/hp·h.

Alternatively, emission rates can be normalized by the fuel flow rate. An *emission index* (EI) is commonly used, for example,

$$\text{EI}_{\text{NO}_x} = \frac{\dot{m}_{\text{NO}_x}(\text{g/s})}{\dot{m}_f(\text{kg/s})} \tag{2.44}$$

with similar expressions for CO, HC, and particulates.

2.14 RELATIONSHIPS BETWEEN PERFORMANCE PARAMETERS

The importance of the parameters defined in Secs. 2.4 to 2.10 to engine performance becomes evident when power, torque, and mep are expressed in terms of these parameters. From the definitions of engine brake power [Eq. (2.20)], mep [Eq. (2.25)], fuel conversion efficiency [Eq. (2.29)], mechanical efficiency [Eq. (2.24)], fuel/air ratio [Eq. (2.32)], and volumetric efficiency [Eq. (2.33)], the following relationships between engine performance parameters can be developed. For brake power P_b,

$$P_b = \frac{\eta_{f,b} m_a N Q_{\text{HV}} (F/A)}{n_R} \tag{2.45}$$

For four-stroke cycle engines, volumetric efficiency can be introduced:

$$P_b = \frac{\eta_v \eta_{f,b} N V_d Q_{\text{HV}} \rho_{a,i} (F/A)}{2} \tag{2.46}$$

Note that since $\eta_{f,b} = \eta_m \eta_{f,i}$, the indicated power P_i is given by equations identical to Eqs. (2.45) and (2.46) with $\eta_{f,b}$ replaced by $\eta_{f,i}$.

For torque T,

$$T = \frac{\eta_v \eta_{f,b} V_d Q_{HV} \rho_{a,i} (F/A)}{4\pi} \tag{2.47}$$

For brake or indicated mep,

$$\text{mep} = \eta_v n_f Q_{HV} \rho_{a,i} (F/A) \tag{2.48}$$

with the fuel conversion efficiency $\eta_{f,b}$ or $\eta_{f,i}$, respectively.

The power per unit piston area, often called the *specific power*, is a measure of the engine designer's success in using the available piston area regardless of cylinder size. From Eq. (2.46) , the specific power is

$$\frac{P}{A_p} = \frac{\eta_v \eta_f N L \rho_{a,i} (F/A) Q_{HV}}{2} \tag{2.49}$$

with η_f being $\eta_{f,b}$ or $\eta_{f,i}$ for brake or indicated specific power. The mean piston speed can be introduced with Eq. (2.10) to give

$$\frac{P}{A_p} = \frac{\eta_v \eta_f \rho_{a,i} Q_{HV} \bar{S}_p}{4} \tag{2.50}$$

Combining Eqs. (2.48) and (2.50), we see that specific power is proportional to the product of mep and mean piston speed.

These relationships illustrate the direct importance to (brake) engine performance of:

1. High volumetric efficiency and intake air density to maximize amount of air inducted into each cylinder
2. Maximum fuel/air ratio that can be burned in each cylinder to fully utilize that air
3. High indicated fuel conversion efficiency to use the fuel's chemical energy effectively

4. Low engine friction to minimize losses and achieve a high mechanical efficiency
5. For power, high mean piston speed consistent with high volumetric efficiency

The brake fuel conversion efficiency $\eta_{f,b}$ of an engine is the product of the indicated fuel conversion efficiency $\eta_{f,i}$ and the mechanical efficiency η_m. The indicated fuel conversion efficiency defines the fraction of the fuel's chemical energy that is transferred to the piston as work. The mechanical efficiency defines that fraction of this work transferred to the piston (the indicated work) that is used to overcome the friction in the engine. Both efficient energy conversion and low friction are important to achieving a high brake fuel-conversion efficiency (low brake-specific fuel consumption). Figure 2.8 shows how these parameters $\eta_{f,i}$, η_m, and $\eta_{f,b}$ vary over the operating regime or map for a naturally-aspirated SI engine. The top line in each figure is the wide-open-throttle brake mean effective pressure. The engine's operating regime beneath the WOT line shows contours of constant indicated fuel conversion efficiency (Fig. 2.8 *a*), mechanical engineering (Fig. 2.8 *b*), and brake fuel conversion efficiency (Fig. 2.8 *c*). The indicated efficiency is essentially constant at about 40% over most of the map. The mechanical efficiency, however, varies from about 0.9 at high load and low speed, to zero at engine idle. The brake fuel conversion efficiency decreases steadily, as load or output is reduced, from a maximum of 35% at 2000 to 2500 rev/min, to zero at idle. It scales, essentially, with the mechanical efficiency underlining the importance of low engine friction.

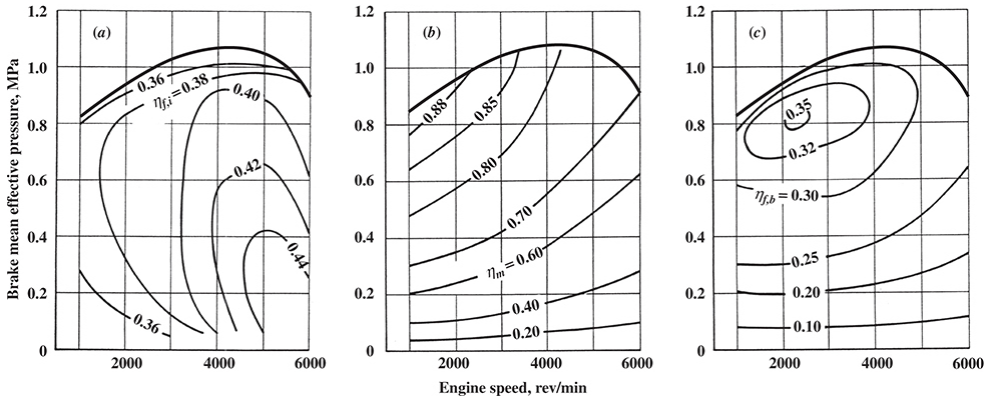


Figure 2.8 Performance map for a modern spark-ignition engine. Top line in each figure shows wide-open-throttle brake mean effective pressure as a function of engine speed. Figures show: (*a*) lines of constant (gross) indicated fuel-conversion efficiency; (*b*) lines of constant mechanical efficiency; (*c*) lines of constant brake fuel conversion efficiency. Data from Ref. 4. Note $\eta_{f,b} = \eta_m \eta_{f,i}$.

2.15 ENGINE DESIGN AND PERFORMANCE DATA

Figures 2.9 and 2.10 show the maximum torque and bmep, and the maximum brake power, as a function of speed for a typical naturally-aspirated light-duty-vehicle SI engine and a turbocharged truck diesel engine. Engines can, of course, operate at any point below these maximum output curves. Engine ratings usually indicate the maximum power and speed at which manufacturers expect their products to give satisfactory economy, reliability, and durability under service conditions. Maximum torque, and the speed at which it is achieved, is also usually given. [g] For the naturally-aspirated gasoline engine, the torque (and bmep) at maximum power is lower than the maximum torque by about 10%, since at higher speed, the volumetric efficiency is lower (due to increased flow resistances) and friction is higher. For the turbocharged diesel engine, the shape of the maximum torque/bmep versus speed curve depends on the relative sizing of the engine and the turbocharger. Usually, as shown in Fig. 2.10, a relatively smaller turbocharger is used to provide high torque at low to mid engine speeds. A relatively larger turbocharger would raise the high speed end of the torque curve, and lower the low speed end.

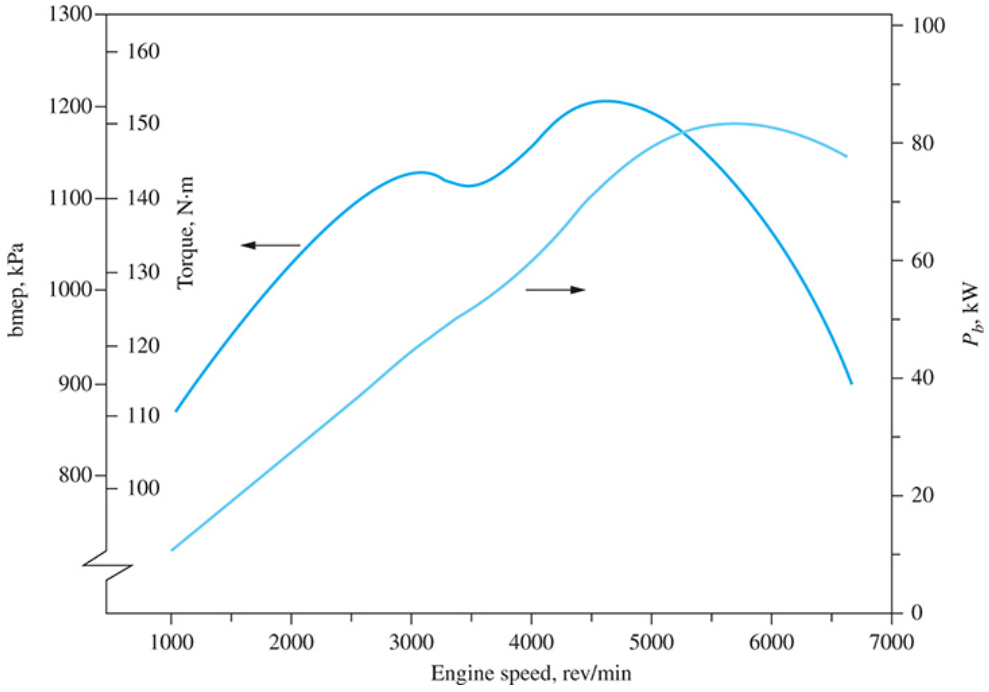


Figure 2.9 Wide-open-throttle torque, bmep, and power versus speed for a typical automobile SI engine. DaimlerChrysler four-cylinder in-line 1.6-liter engine: four valves per cylinder, bore = 77 mm, stroke = 85.8 mm, compression ratio = 10.5, fixed valve timing. [5]

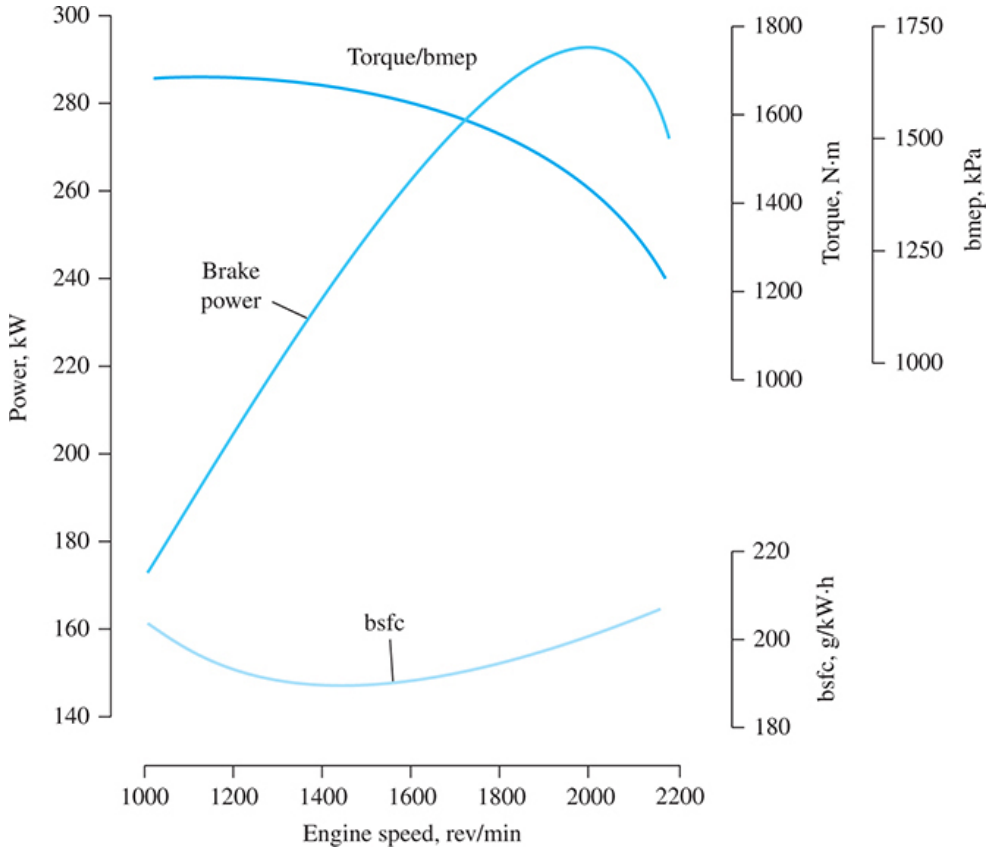


Figure 2.10 Full-load torque, bmep, and power versus speed for a typical turbocharged and intercooled direct-injection truck diesel engine. Hino K13C in-line six-cylinder engine; 12.9-liter displacement, bore = 135 mm, stroke = 150 mm, compression ratio = 16.5. [6]

Since both power and torque depend on displaced volume, for comparative analyses between engines of different displacements in a given engine category, normalized performance parameters are more useful. The following measures, at the operating points indicated, have most significance: [7]

1. At maximum rated power:

   Mean piston speed . Measures comparative success in reducing resistance to air flow, handling loads due to inertia of the reciprocating parts, and engine friction.

Brake mean effective pressure . In naturally-aspirated engines, bmep is not stress limited. It reflects the product of volumetric efficiency (ability to induct air), fuel/air ratio (effectiveness of air utilization in combustion), indicated fuel conversion efficiency, and reducing engine friction. In boosted (supercharged or turbocharged) engines, bmep indicates, also, the degree of success in handling higher gas pressures and thermal loading.

Power per unit piston area . Measures the effectiveness with which the piston area is used, regardless of cylinder size.

Specific weight . Indicates relative economy with which materials are used.

Specific volume . Indicates relative effectiveness with which the space occupied by the engine has been utilized.

2. At all speeds at which the engine will be used with full throttle or with maximum fuel-pump setting:

Brake mean effective pressure . Measures ability to obtain/provide high air flow and use that air effectively to generate torque.

3. In all useful regimes of operation, and especially in those regimes where the engine is run for long periods of time:

Brake specific fuel consumption or fuel conversion efficiency.

Brake specific emissions .

Typical performance data for SI and diesel engines over the normal production size range are summarized in Table 2.1. The four-stroke cycle dominates except in the smallest and largest engine sizes. The larger engines are turbocharged and/or supercharged. The maximum rated engine speed decreases as engine size increases, maintaining the maximum mean piston speed in the range of about 10 to 20 m/s. The maximum brake mep for turbocharged engines is significantly higher than for naturally-aspirated engines. Because the maximum fuel/air ratio for SI engines is higher than for diesels, their naturally-aspirated maximum bmep levels (at equal boost pressures) are higher. As engine size increases, brake specific fuel

consumption decreases and fuel conversion efficiency increases, due to reduced importance of friction and heat losses. For the largest diesel engines, brake fuel conversion efficiencies of up to 55% (and indicated fuel conversion efficiencies approaching 60%) can be obtained.

TABLE 2.1 Typical design and operating data for internal combustion engines [1, 4, 8]

					Rated maximum				
	Operating cycle	Compression ratio	Bore, m	Stroke/ bore	Speed, rev/min	bmep, bar	Power per unit volume kW/dm³	Weight/ power, kg/kW	Best bsfc, g/kW·h
Spark-ignition engines:									
Small (e.g., motorcycles)	2S	7–11	0.05–0.08	1.2–0.9	4500–14,000	5–12	40–160	2–0.4	380
	4S	9–12			5000–13,000	9–13	50–150	2.5–0.5	320
Passenger cars (NA)	4S	10–12	0.07–0.1	1.1–0.9	5000–8000	11–13	40–80	2–0.8	240
(TC,CAC)	4S	9–11			5000–7500	16–24	60–110	1.5–0.5	230
Trucks (NA)	4S	9–11	0.09–0.13	1.2–0.7	2000–3500	8–10	20–35	5.0–3.0	240
Large gas engines	4S	9–12	0.22–0.45	1.1–1.4	300–1000	7–14	30–40	7–3	200
Wankel engine	4S	10	0.65 dm³ per chamber		8500	10.5	140	~ 1	280
Diesel engines:									
Passenger cars (NA)	4S	19–24	0.075–0.1	1.2–0.9	3500–5000	7–9	20–35	4–2	240
(TC)	4S	18–22			3500–4500	12–20	35–55	3–1.3	220
Trucks (NA)	4S	16–18	0.1–0.15	1.3–0.8	2000–4000	7–10	15–25	6–3	210
(TC,CAC)	4S	15–18	0.1–0.15	1.3–0.8	1800–3200	18–24	30–45	4–3	200
Locomotive, industrial	4S, 2S	12–18	0.15–0.4	1.1–1.3	425–1800	7–23	5–20	6–18	180
Large engines, marine, power generation	2S	10–12	0.4–1	1.2–3	110–400	10–20	2–8	20–50	160

Abbreviations: 2S, two stroke; 4S, four stroke; NA, naturally-aspirated; TC, turbocharged; CAC, charge-air cooling.

2.16 VEHICLE POWER REQUIREMENTS

When an engine is driving a vehicle, it must provide enough power to overcome the resistances to vehicle motion and accelerate the vehicle, as well as overcome the losses in the transmission and driveline. The vehicle resistances and the inertia force, illustrated in Fig. 2.11, are: tire rolling resistance F_R, vehicle aerodynamic drag F_D, vehicle acceleration F_a (negative when decelerating), gravity when climbing a grade F_g (negative when descending a grade), and any braking force F_b. The power required at the wheels to drive the vehicle (P_v) is, therefore,

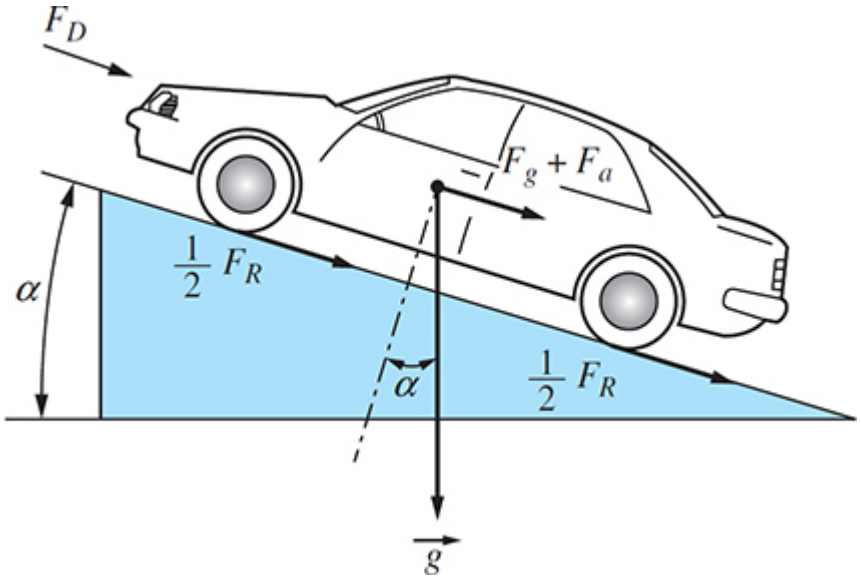


Figure 2.11 Forces resisting the motion of a vehicle climbing a grade.

$$P_v = (F_R + F_D + F_a + F_g + F_b)S_v \tag{2.51}$$

This vehicle power is the engine power multiplied by the combined transmission and driveline efficiency. These individual resistances (forces) are given by:

$$F_R = C_R m_v g \cos\alpha \tag{2.52a}$$

$$F_D = \frac{1}{2}\rho_a C_D A_v S_v^2 \tag{2.52b}$$

$$F_a = m_v a_v \tag{2.52c}$$

$$F_g = m_v g \sin\alpha \tag{2.52d}$$

where

C_R = coefficient of rolling resistance ($0.01 < C_R < 0.02$)[8]
m_v = mass of vehicle (curb mass plus payload; in U.S. units W_v = vehicle weight in lb_f)

g = acceleration due to gravity
α = grade angle
ρ_a = ambient air density
C_D = drag coefficient (for cars: $0.2 < C_D \leq 0.5$); average about 0.33[8]
A_v = frontal area of vehicle ($\approx 0.9 \times$ vehicle height $\times$ width)[8]
S_v = vehicle speed
a_v = vehicle acceleration

A part-load power level useful as a reference point for testing automobile engines is the power required to drive the vehicle on a level road at a steady speed. Called *road-load power*, this power overcomes the tire rolling resistance and the aerodynamic drag of the vehicle. Rolling resistance and drag coefficients, C_R and C_D, respectively, are determined empirically. Equations (2.51) and (2.52) give the formula for road-load power P_r as

$$P_r = (C_R m_v g + \frac{1}{2}\rho_a C_D A_v S_v^2) S_v \tag{2.53}$$

With the quantities in the units indicated:

$$P_r(\text{kW}) = \left[2.73 C_R m_v(\text{kg}) + 0.0126 C_D A_v(\text{m}^2) S^2{}_v(\text{km/h})^2\right] S_v(\text{km/h}) \times 10^{-3} \tag{2.54a}$$

or

$$P_r(\text{hp}) = \frac{\left[C_R W_v(\text{lb}_f) + 0.0025 C_D A_v(\text{ft}^2) S^2{}_v(\text{mi/h})^2\right] S_v(\text{mi/h})}{375} \tag{2.54b}$$

PROBLEMS

2.1 Explain why the brake mean effective pressure of a naturally-aspirated diesel engine is lower than that of a naturally-aspirated spark-ignition engine. Explain why the bmep is lower at the maximum rated power for a given engine than the bmep at the maximum torque.

2.2 Calculate the mean piston speed, bmep, and specific power of: **(a)** the spark-ignition engines in Figs. 1.11 and 1.13 at their maximum rated power

and torque; and **(b)** the mean piston speed, bmep, and specific power of the diesel engines in Figs. 1.31 and 1.32 at their maximum rated power and torque. Briefly explain any significant differences between these two types of engine.

2.3 Develop an equation for the power required to drive a vehicle at constant speed up a hill of angle α, in terms of vehicle speed, mass, frontal area, drag coefficient, coefficient of rolling resistance, α, and acceleration due to gravity. Calculate this power when the car mass is 1500 kg, the hill angle is 15 degrees, and the vehicle speed is 50 mi/h (80 km/h).

2.4 The spark-ignition engine in Fig. 1.4 is operating at a mean piston speed of 10 m/s. The measured air flow is 60 g/s. Calculate the volumetric efficiency (based on atmospheric conditions).

2.5 The diesel engine of Fig. 1.31 is operating with a mean piston speed of 8 m/s. Calculate the air flow if the volumetric efficiency is 0.92. If (F/A) is 0.05, what is the fuel flow rate and the mass of fuel injected per cylinder per cycle?

2.6 The brake fuel conversion efficiency of a spark-ignition engine is 0.3, and varies little with fuel type. Calculate the brake specific fuel consumption for isooctane, gasoline, ethanol, methanol, and hydrogen (relevant data are in App. D).

2.7 You are doing a preliminary design study of a turbocharged four-stroke diesel engine. The maximum rated power is limited by stress considerations to a brake mean effective pressure of 1200 kPa and maximum value of the mean piston speed of 12 m/s.

(a) Derive an equation relating the engine inlet pressure (pressure in the intake manifold at the turbocharger compressor exit) to the fuel/air ratio at this maximum rated power operating point. Other reciprocating engine parameters (e.g., volumetric efficiency, fuel conversion efficiency, bmep, etc.) appear in this equation also.

(b) The maximum rated brake power requirement for this engine is 400 kW. Estimate appropriate values for number of cylinders, cylinder bore, and

stroke, and determine the maximum rated speed of this preliminary engine design.

(c) If the pressure ratio across the compressor is 2, estimate the overall fuel/air and air/fuel ratios at the maximum rated power. Assume appropriate values for any other parameters you may need.

2.8 The brake fuel conversion efficiency of an engine is 0.3. The mechanical efficiency is 0.8. The combustion efficiency is 0.94. The heat losses to the coolant and oil are 60 kW. The fuel chemical energy entering the engine per unit time, $\dot{m}_f Q_{\mathrm{HV}}$, is 190 kW. What percentage of this energy becomes **(a)** brake work; **(b)** friction work; **(c)** heat losses; **(d)** exhaust chemical energy; and **(e)** exhaust sensible energy?

2.9 A modern naturally-aspirated spark-ignition engine has an indicated fuel conversion efficiency $\eta_{f,i}$ of 38% over its operating range. At 2000 rev/min, the mechanical friction mean effective pressure (mfmep) is 90 kPa and the accessory (water, oil, and fuel pump) friction (afmep) is 20 kPa; both are independent of load. The engine operates at stoichiometric and MBT spark timing at all intake manifold pressures. The exhaust pressure is p_e, fuel heating value Q_{HV}, fuel/air ratio, intake system air density $\rho_{a,i}$ just downstream of the throttle, and the volumetric efficiency η_v is based on $\rho_{a,i}$.

(a) Develop an expression for the brake fuel conversion efficiency as a function of $\eta_{f,i}$ mfmep, afmep, intake manifold pressure p_i, exhaust manifold pressure p_e, fuel heating value Q_{HV}, fuel/air ratio, intake system air density $\rho_{a,i}$ just downstream of the throttle, and volumetric efficiency η_v is based on $\rho_{a,i}$.

(b) Evaluate the engine's mechanical efficiency and brake fuel conversion efficiency, at an intake manifold pressure of 0.4, 0.7, and 1.0 bar.

(c) Assuming that the mechanical and accessory friction values remain the same at idle, determine the intake manifold pressure at idle.

2.10 In the reciprocating engine, during the power or expansion stroke, the gas pressure force acting on the piston is transmitted to the crankshaft via the connecting rod. List the forces acting on the piston during this part of the operating cycle. Show the direction of the forces acting on the piston on a

sketch of the piston, cylinder, connecting rod, crank arrangement. Write out the force balance for the piston **(a)** along the cylinder axis and **(b)** transverse to the cylinder axis in the plane containing the connecting rod. (You are not asked to manipulate or solve these equations.)

2.11 Several velocities, time, and length scales are useful in understanding what goes on inside engines. Make estimates of the following quantities for a 1.6-liter displacement four-cylinder spark-ignition engine, operating at wide-open throttle at 2500 rev/min.

(a) The mean piston speed and the maximum piston speed.

(b) The maximum charge velocity in the intake port (the port area is about 20% of the piston area).

(c) The time occupied by one engine operating cycle, the intake process, the compression process, the combustion process, the expansion process, and the exhaust process. (*Note*: The word *process* is used here, not the word *stroke*.)

(d) The average velocity with which the flame travels across the combustion chamber.

(e) The length of the intake system (the intake port, the manifold runner, etc.) that is filled by one cylinder charge just before the intake valve opens and this charge enters the cylinder (i.e., how far back from the intake valve, in centimeters, does one cylinder volume of gas extend into the intake system).

(f) The length of exhaust system filled by one cylinder charge after it exits the cylinder (assume an average exhaust gas temperature of 425°C).

You will have to make several appropriate geometric assumptions. The calculations are straightforward, and only approximate answers are required.

2.12 A four-cylinder, four-stroke diesel engine is being designed. A bore of 100 mm and a stroke of 120 mm have been selected and the operating speed of interest is 1500 rev/min. A turbocharging system is envisioned that will supply inlet manifold air at 2.0 bar and 380 K. The volumetric efficiency is expected to be 90%. The indicated fuel conversion efficiency has been estimated at 55%, and an air/fuel ratio of 28:1 is to be used. The friction mean effective pressure is expected to be 2.2 bar. The heating value of the fuel is 42.5 MJ/kg.

Estimate:

(a) Mass flow rate of air into the engine (kg/s)

(b) Mass of fuel burned per cylinder per cycle (mg)

(c) The indicated work done per cylinder per cycle (kJ)

(d) The brake mean effective pressure (kPa)

(e) The engine shaft power output (kW)

(f) The brake specific fuel consumption (g/kW·h)

(g) The brake fuel conversion efficiency

2.13 An automobile engine must provide enough power to overcome the vehicle's resistances. These resistances include the rolling resistance due to friction between the tires and the road, aerodynamic drag, gravity (when traveling up a hill), as well as the inertia of the vehicle while accelerating. For a Ford Taurus vehicle that weighs 1515 kg and has a relatively flat maximum engine torque versus speed curve of 247 N·m for medium engine speeds, estimate the following:

(a) The road load power (power required to drive the vehicle on a level road at steady speed) when the vehicle speed is 50 km/h and 100 km/h. How does the relative importance of each resistance change as vehicle speed increases?

(b) The power required to drive the vehicle up a 20% gradient hill at a steady speed of 50 km/h.

(c) The minimum time required to accelerate the vehicle from 40 to 80 km/h on a level road with the vehicle in third gear. (In third gear for this vehicle, the ratio of engine to wheel speed is 3.27:1.)

Make appropriate estimates for tire rolling resistance and car drag coefficient and geometry (frontal area, wheel radius).

2.14 At 45° after top center on the expansion stroke, the pressure in the engine's cylinder is 1000 kPa. The bore and stroke are 80 mm, the compression ratio is 10, the con rod length: crank radius is 3.5, the crankcase pressure is 100 kPa, the axial friction force on the piston is 65 N. The mass of the piston (and piston pin and half the con rod) is 0.57 kg. The engine speed is 2500 rev/min.

(a) Calculate the forces acting on the piston, the compression force in the connecting rod, and the torque delivered to the crankshaft.

(b) At this same crank position during the intake stroke, the pressure in the cylinder is 0.5. The friction force is about the same. Estimate the force in the connecting rod under these conditions.

2.15 Laboratory project: Disassembly of a multi-cylinder production gasoline.

During the lab session you will disassemble an engine, and make and record the following measurements:

(a) Cylinder bore (at top center and bottom center)

Connecting rod length (pin center to pin center)

Stroke

Piston top land diameter

Piston top land height

Piston top ring groove dimensions

Piston top ring thickness, height, and circumference

Piston diameter, skirt center to skirt center

Overall piston height

(b) Intake and exhaust valve head diameter

Intake and exhaust valve seat diameter (mean value)

Intake and exhaust valve stem diameter

Angle between the valve stems

Maximum intake and exhaust valve lift

Minimum intake port cross-sectional area

Mean intake port length

Mean intake manifold "diameter"

Mean intake manifold runner length

(c) Do some hand-drawn, carefully proportioned sketches to illustrate the geometry of the piston, con rod, etc., in the cylinder, and a cross-section through the cylinder head and ports.

(d) Calculate these geometric ratios:

Bore/stroke

Mean clearance height/bore

Connecting rod length/crank radius

Intake valve seat diameter/bore

Maximum intake valve lift/valve diameter

Intake valve head diameter/mean seat diameter

Exhaust valve seat diameter/bore

Maximum exhaust valve lift/valve diameter

Exhaust valve head diameter/mean seat diameter

Volume of one intake port plus one intake manifold runner/displaced volume of one cylinder

(e) What in addition to the components that were part of the engine you disassembled is needed for that engine to operate continuously? Write out a list.

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[a] The mass of the upper part of the connecting rod is often taken as the mass above the center of gravity of the connecting rod.

[b] The term indicated is used because such *p-V* diagrams used to be generated directly with a device called an engine indicator.

[c] With many two-stroke engine concepts, there is a piston pumping work term associated with compressing the scavenging air in the crankcase.[1]

[d] The various components of friction are examined in detail in Chap. 13 .

[e] Note that these typical best values of bmep for different types of engines increase over time as engines are improved. The average improvement is on the order of 1% per year. [3]

[f] This empirically defined engine efficiency has previously been called thermal efficiency or enthalpy efficiency. The term fuel conversion efficiency is preferred because it describes this quantity more precisely, and distinguishes it clearly from other definitions of engine efficiency, which will be developed in Sec. 3.6 . Note that there are several different definitions of heating value (see Sec. 3.5). The numerical values do not normally differ by more than a few percent, however. In this text, the lower heating value at constant pressure is used in evaluating the fuel conversion efficiency.

[g] The dip in the WOT torque curve at 3500 rev/min is due to intake tuning effects (see Sec. 6.2.6).

CHAPTER 3

Thermochemistry of Fuel-Air Mixtures

3.1 CHARACTERIZATION OF FLAMES

Combustion of the fuel-air mixture inside the engine cylinder is one of the key processes that controls engine power, efficiency, and emissions. Some background in relevant combustion phenomena is therefore a necessary preliminary to understanding engine operation. These combustion phenomena are different for the two main types of engines—spark-ignition and diesel—which are the subject of this book. In spark-ignition (SI) engines, the fuel is normally well mixed with air, either in the engine's intake ports or within the engine's cylinders. Following the compression of this fuel-air mixture, an electrical discharge initiates the combustion process; a flame develops from the "kernel" created by the spark discharge, and propagates across the cylinder to the combustion chamber walls. At the walls, the flame is "quenched" or extinguished as heat transfer and destruction of active species at the wall become the dominant processes. An undesirable combustion phenomenon—the "spontaneous'' ignition of a substantial mass of fuel-air mixture ahead of the flame, before the flame can propagate through this mixture (which is called the end gas)—can also occur. This autoignition or self-explosion combustion phenomenon is the cause of spark–ignition engine knock which due to the high pressures generated, can lead to engine damage.

In the diesel engine, the fuel is injected into the cylinder into air already at high pressure and temperature, near the end of the compression stroke. The autoignition, or self-ignition, of portions of the developing mixture of already

injected and vaporized fuel with this hot air starts the combustion process, which spreads rapidly. Burning then proceeds as fuel and air mix to the appropriate composition for combustion to take place. Thus, fuel-air mixing plays a controlling role in the diesel combustion process.

Chapters 3 and 4 focus on the thermochemistry of combustion, that is, the composition and thermodynamic properties of the pre- and postcombustion working fluids in engines and the energy changes associated with the combustion processes that take place inside the engine cylinder. Later chapters (9 and 10) deal with the phenomenological aspects of engine combustion, that is, the details of the physical and chemical processes by which the fuel-air mixture is converted to burned products. At this point it is useful to review the key combustion phenomena that occur in engines to provide an appropriate background for the material that follows. More detailed information on these combustion phenomena can be found in texts on combustion such as those of Borman and Ragland [1] and Glassman. [2]

The combustion process is a fast exothermic gas-phase reaction where oxygen is a critical reactant. A flame is the region within which the combustion reaction takes place, which can propagate subsonically through space; motion of the flame relative into the unburned 82gas is the important feature. The existence of a propagating flame implies that the reaction is confined to a zone that is small in thickness compared to the dimensions of the apparatus—in our case, the engine combustion chamber. This flame characteristic of spatial propagation is the result of the strong coupling between chemical reaction, the transport processes of mass diffusion and heat conduction, and fluid flow. The generation of heat and active species accelerate the chemical reaction; the supply of fresh reactants, governed by the convection velocity, limits the reaction. When these processes are in balance, a steady-state flame results.

Flames are usually classified according to the following overall characteristics. The first of these has to do with the composition of the reactants as they enter the reaction zone. If the fuel (in vapor form) and oxidizer are essentially uniformly mixed together, the flame is designated as *premixed*. If the reactants are not premixed and must mix together in the region where reaction takes place, the flame is called a *diffusion* flame because the mixing is accomplished by the diffusion of fuel and air into the reaction zone. The second means of classification relates to the basic character of the gas flow through the reaction zone: whether it is *laminar* or

turbulent. In laminar flow, mixing and transport occur through molecular processes. Laminar flows occur only at low Reynolds number. The Reynolds number (fluid density × velocity × length scale/viscosity) is the ratio of inertial to viscous forces. In turbulent flows, mixing and transport are enhanced (usually by a substantial factor) by the macroscopic relative motion of eddies or lumps of fluid, which are the characteristic feature of turbulent (high Reynolds number) flows. A third area of classification is whether the flame is *steady* or *unsteady*. The distinguishing feature here is whether the flame structure and motion change with time. The final characterizing feature is the *initial phase* of the reactants—gas, liquid, or solid.

Flames in engines are unsteady, an obvious consequence of the internal combustion engine's operating cycle. Engine flames are turbulent. Only with substantial augmentation of laminar transport processes by turbulent convection can mixing and burning rates and flame-propagation rates be made fast enough to complete the engine combustion process within the time available.

The conventional spark-ignition flame is thus a premixed unsteady turbulent flame, and the fuel-air mixture through which the flame propagates is in the gaseous state. The structure of this premixed flame is illustrated in Fig. 3.1. Within the thin flame sheet (about 0.2 mm thick), the processes that dominate are laminar, that is, governed by molecular diffusion, as illustrated in Fig. 3.1 *a*. The combustion reactions that release the fuel's chemical energy occur in the downstream part of the flame (zone II). The thermal energy released there is both conducted upstream into zone I, the preheating zone, where the temperature of the as yet unreacted fuel-air mixture flowing through the zone steadily rises, and is also carried away downstream in the high temperature burned gases flowing out of the flame into zone III. Effectively, the temperature of the unburned mixture must be raised above an *ignition temperature*, T_i, before significant chemical reaction and energy release can occur. Note that the flame structure illustrated shows unburned fuel-air mixture flowing into the flame from the left; the observer is thus moving with the flame. Within the spark-ignition engine, the flame propagates across the combustion chamber moving into the unburned mixture ahead of it. This spark-ignition engine flame is a turbulent premixed flame. Figure 3.1 *b* shows several short time scale realizations of a turbulent premixed flame: the fuel-air mixture flows out of the tube as shown, and through the "wrinkled" thin flame sheet. The turbulent fluid motion convects, distorts, and stretches

this flame sheet producing a "laminar-like wrinkled flame." Locally, at each point of this turbulent flame sheet, the laminar flame structure shown in Fig. 3.1 *a* pertains.

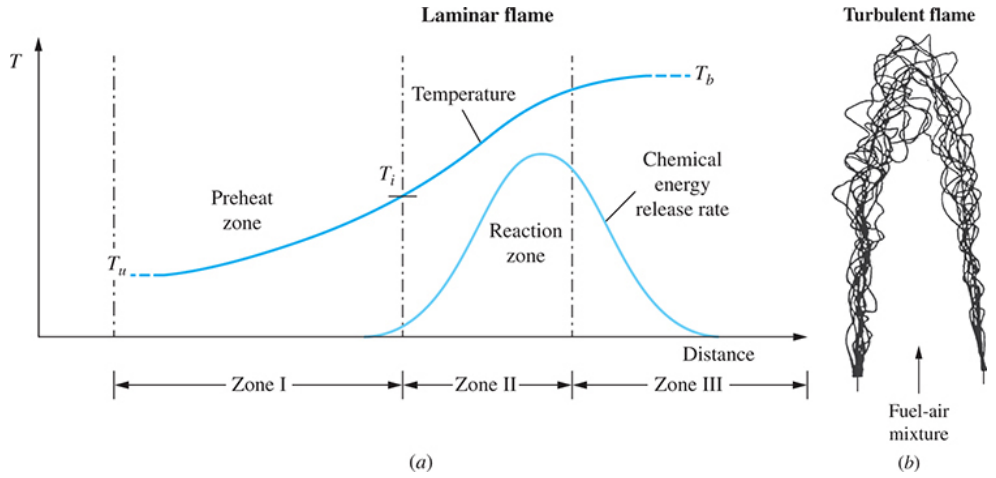


Figure 3.1 Premixed flame structure: (*a*) Temperature and chemical energy release-rate profiles through a premixed laminar flame; (*b*) multiple short time-exposure realizations of a turbulent premixed flame.

The diesel engine combustion process is predominantly an unsteady turbulent diffusion flame, and the fuel is initially in the liquid phase. Figure 3.2 illustrates the structure of this diffusion flame. In diffusion flames, the fuel and air are initially separate, as shown in Fig. 3.2 *a*. Fuel vapor [produced within each diesel engine fuel spray (see Fig. 1.25) by evaporation of the liquid fuel drops] flows into the inside of the flame; air flows into the flame from the outside (Fig. 3.2 *b*). Chemical reaction occurs where the temperatures are highest; that is, where there is just enough air to fully release the fuel's chemical energy—the stoichiometric mixture composition. As shown, the combustion products diffuse away from the diffusion flame on the outside into the surrounding air, and on the inside where the downstream flow within the spray/flame boundary also moves these products away from the flame. In the diesel combustion process, the diffusion flame which surrounds each fuel spray, is also turbulent; the thin flame is made irregular in shape (i.e., wrinkled) by the turbulent flow. Locally it retains the structure shown in Fig. 3.2 *b*.

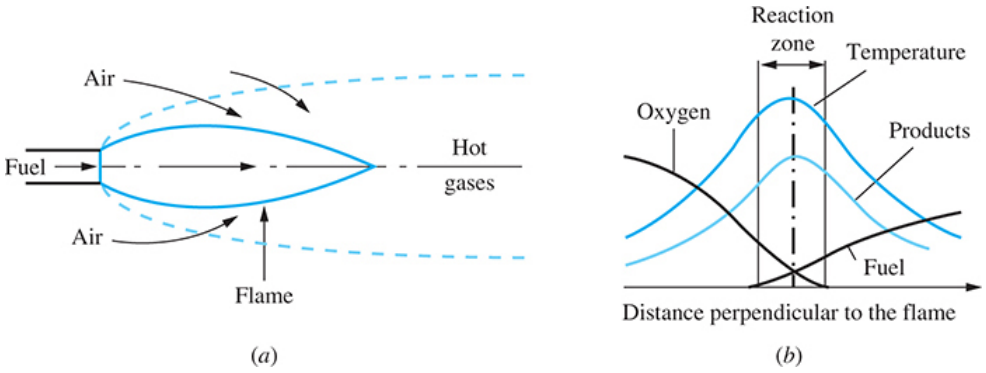


Figure 3.2 Diffusion flame structure: (*a*) Fuel, air, and flame location in a jet diffusion flame; (*b*) concentration profiles of fuel, oxidizer, products, and temperature, through the flame.

Both these engine flames are extremely complicated because they involve the coupling of the multispecies chemical mechanism by which fuel and oxidizer react to form products, with the turbulent convective transport process. The diesel combustion process is even more complicated than the spark-ignition engine combustion process because vaporization of liquid fuel and the fuel-air mixing process are involved too. Chapters 9 and 10 contain a more detailed discussion of the SI engine and diesel combustion processes, respectively. This chapter reviews the basic thermodynamic and chemical composition aspects of engine combustion.

3.2 IDEAL GAS MODEL

The gas species that make up the working fluids in internal combustion engines (e.g., oxygen, nitrogen, fuel vapor, carbon dioxide, water vapor, etc.) can usually be treated as ideal gases. The relationships between the thermodynamic properties of an ideal gas and of ideal gas mixtures are reviewed in App. B. There can be found the various forms of the ideal gas law:

$$pV = mRT = m\frac{\tilde{R}}{M}T = n\tilde{R}T \tag{3.1}$$

where p is the pressure, V the volume, m the mass of gas, R the gas constant for the gas, T the temperature, $\tilde{R}$ the universal gas constant, M the molecular weight, and n the number of moles. Relations for evaluating the specific internal energy u, enthalpy h, and entropy s, specific heats at constant volume c_v and constant pressure c_p($c_p = c_v + R$) on a per unit mass basis and on a per mole basis (where the notation $\tilde{u}, \tilde{h}, \tilde{s}, \tilde{c}_v$, and $\tilde{c}_p$ is used) of an ideal gas, are developed there. Also given are equations for calculating the thermodynamic properties of mixtures of ideal gases.

3.3 COMPOSITION OF AIR AND FUELS

Normally in engines, fuels are burned with air. Dry air is a mixture of gases that has a representative composition by volume of 20.95% oxygen, 78.09% nitrogen, 0.93% argon, and trace amounts of carbon dioxide, neon, helium, methane, and other gases. Table 3.1 shows the relative proportions of the major constituents of dry air.[3]

TABLE 3.1 Principle constituents of dry air

Gas	ppm by volume	Molecular weight	Mole fraction	Molar ratio
O_2	209,480	31.998	0.2095	1
N_2	780,840	28.012	0.7905	3.773
Ar	9300	39.948		
CO_2	404	44.009		
Air	1,000,000	28.962	1.0000	4.773

In combustion, oxygen is the reactive component of air. It is usually sufficiently accurate to regard air as consisting of 21% oxygen and 79% inert gases taken as nitrogen (often called atmospheric or apparent nitrogen). For each mole of oxygen in air, there are

$$\frac{1-0.2095}{0.2095} = 3.773$$

moles of atmospheric nitrogen. The molecular weight of air is obtained from Table 3.1 with Eq. (B.17) as 28.962, usually approximated by 29. Because

atmospheric nitrogen contains traces of other species, its molecular weight is slightly different from that of pure molecular nitrogen, that is,

$$M_{aN_2} = \frac{28.962 - 0.2095 \times 31.998}{1 - 0.2095} = 28.16$$

In the following sections, *nitrogen* will refer to atmospheric nitrogen and a molecular weight of 28.16 will be used. An air composition of 3.773 moles of nitrogen per mole of oxygen will be assumed.

The density of dry air can be obtained from Eq. (3.1) with $R = 8314.3$ J/kmol·K and $M = 28.962$:

$$\rho(\text{kg/m}^3) = \frac{3.483 \times 10^{-3}\, p(\text{Pa})}{T(K)} \tag{3.2a}$$

or

$$\rho(\text{lb}_m/\text{ft}^3) = \frac{2.699\, p(\text{lb}_f/\text{in}^2)}{T(^\circ R)} \tag{3.2b}$$

Thus, the value for the density of dry air at 1 atmosphere (1.0133×10^5 Pa, 14.696 lb$_f$/in^2) and 25°C (77°F) is 1.184 kg/m^3 (0.0739 lb$_m$/ft^3).

Actual air contains water vapor, the amount depending on temperature, and degree of saturation. Typically the proportion by mass is about 1%, though it can rise to about 4% under extreme conditions. The *relative humidity* compares the water vapor content of air with that required to saturate. It is defined as:

> The ratio of the partial pressure of the water vapor actually present to the saturation pressure at the same temperature.

Water vapor content is measured with a wet- and dry-bulb psychrometer. This consists of two thermometers exposed to a stream of moist air. The dry-bulb temperature is the temperature of the air. The bulb of the other thermometer is wetted by a wick in contact with a water reservoir. The wet-bulb temperature is lower than the dry-bulb temperature due to evaporation of water from the wick. It is a good approximation to assume that the wet-bulb temperature is the adiabatic saturation temperature. Water vapor

pressure can be obtained from observed wet- and dry-bulb temperatures and a psychrometric chart such as Fig. 3.3. [4] The effect of humidity on the properties of air is given in Fig. 3.4. [5]

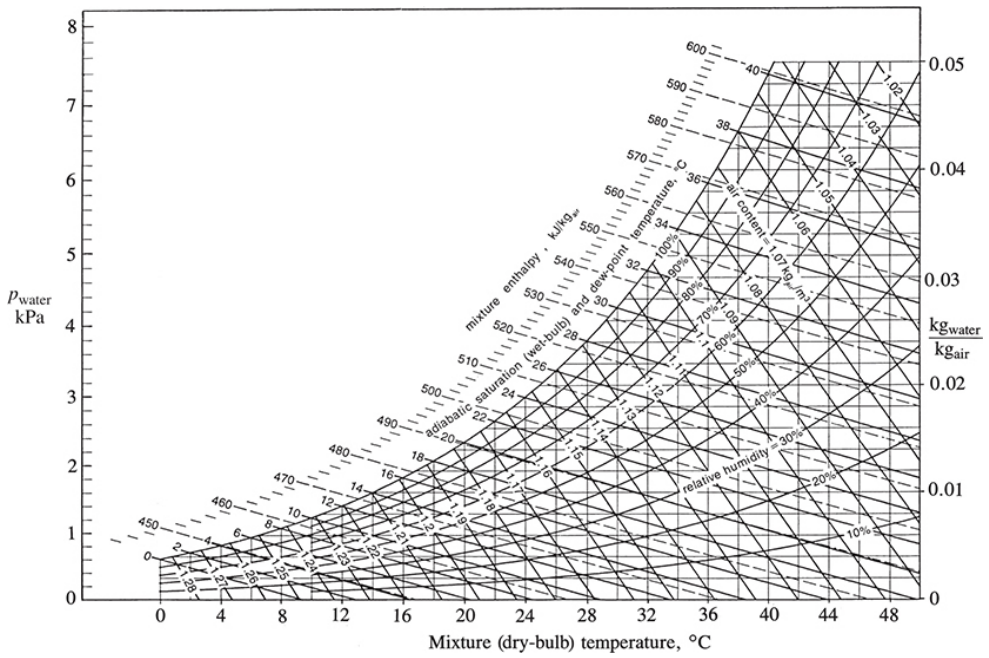


Figure 3.3 Psychrometric chart for air-water mixtures at 1 atmosphere. (*From Reynolds.* [4])

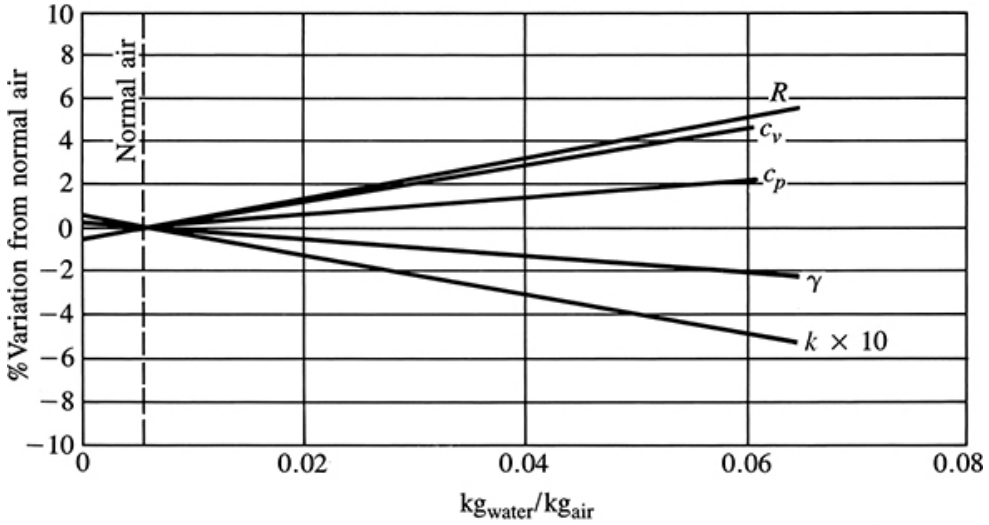


Figure 3.4 Effect of humidity on properties of air: R is the gas constant; c_v and c_p are specific heats at constant volume and pressure, respectively; γ =

$c_p/_v$;k is the thermal conductivity. (*From Taylor*. [5])

The fuels most commonly used in internal combustion engines (gasoline or petrol, and diesel) are blends of many different hydrocarbon compounds obtained by refining petroleum or crude oil. These fuels are predominantly carbon and hydrogen (typically about 86% carbon and 14% hydrogen by weight). In some regions of the world, diesel fuels can contain up to about 1% sulfur. Other fuels of interest are alcohols (which contain oxygen), [a] gaseous fuels (natural gas and liquid petroleum gas), and single hydrocarbon compounds (e.g., methane, propane, isooctane) that are often used in engine research. Properties of the more common internal combustion engine fuels are summarized in App. D.

Some knowledge of the different classes of organic compounds and their molecular structure is necessary in order to understand combustion mechanisms. [6] The different classes are as follows:

Alkyl Compounds

Single-bonded open-chain saturated hydrocarbon molecules, that is, no more hydrogen can be added. For the larger molecules, different isomers —straight-chain and branched-chain configurations—exist. These are called normal (*n*-) and iso compounds, respectively. Examples: CH_4, methane; C_2H_6, ethane; C_3H_8, propane; C_8H_{18}, *n*-octane and isooctane.



There are several “isooctanes,” depending on the relative position of the branches. By isooctane is usually meant the molecule 2,2,4-trimethylpentane, indicating five carbon atoms in the straight chain (pentane) with three methyl (CH_4) branches located, respectively, at C-atoms 2, 2, and 4. Radical molecules deficient in one hydrogen atom take the name methyl, ethyl, propyl, etc.

Single-bond (no double-bond) ring hydrocarbons: Unsaturated, since ring can be broken and additional hydrogen added. Examples: C_3H_6, cyclopropane (three C-atom ring); C_4H_8, cyclobutane (four C-atom ring); C_5H_{10}, cyclopentane (five C-atom ring).



Open-chain hydrocarbons containing a double bond; hence they are unsaturated. Examples: C_2H_4, ethene (or ethylene); C_3H_6, propene (or

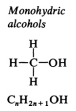
propylene); C_4H_8, butene (or butylene). From butene upward, several structural isomers are possible depending on the location of the double bond in the basic carbon chain. Straight- and branched-chain structures exist. Diolefins contain two double bonds.

Open-chain unsaturated hydrocarbons containing one carbon-carbon triple bond. First member is acetylene, H–C≡C–H. Additional members of the alkyne series comprise open-chain molecules, similar to higher alkenes but with each double bond replaced by a triple bond.

Aromatics

Building block for aromatic hydrocarbons is the benzene (C_6H_6) ring structure shown. This ring structure is very stable and accommodates additional $–CH_2$ and $–CH_3$ groups in side chains and not by ring expansion. Examples: C_7H_8, toluene; C_8H_{10}, xylene (several structural arrangements). More complex aromatic hydrocarbons incorporate ethyl, propyl, and heavier alkyl side chains in a variety of structural arrangements.

Alcohols



In these organic compounds, one hydroxyl (−OH) group is substituted for one hydrogen atom. Thus methane becomes methyl alcohol, CH_3OH (also called methanol); ethane becomes ethyl alcohol, C_2H_5OH (ethanol); etc.

3.4 COMBUSTION STOICHIOMETRY

This section develops relations between the composition of the reactants (fuel and air) in a combustible mixture and the composition of the products. Since these relations depend only on the conservation of mass of each chemical element in the reactants, only the relative elemental composition of the fuel and the relative proportions of fuel and air are needed.

If sufficient oxygen is available, a hydrocarbon fuel can be completely oxidized. The carbon in the fuel is then converted to carbon dioxide, CO_2,

and the hydrogen to water, H_2O. For example, consider the overall chemical equation for the complete combustion of one mole of propane C_3H_8:

$$C_3H_8 + aO_2 = bCO_2 + cH_2O \tag{3.3}$$

A carbon balance between the reactants and products gives $b = 3$. A hydrogen balance gives $2c = 8$, or $c = 4$. An oxygen balance gives $2b + c = 10 = 2a$, or $a = 5$. Thus Eq. (3.3) becomes

$$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O \tag{3.4}$$

Note that Eq. (3.4) only relates the elemental composition of the reactant and product species; it does not indicate the chemical mechanism by which combustion proceeds, which is much more complex.

Air contains nitrogen, but when the products are at relatively low temperatures, the nitrogen is not significantly affected by the reaction. Consider the complete combustion of a general hydrocarbon fuel of average molecular composition C_aH_b with air. The overall complete combustion equation is

$$C_aH_b + \left(a + \frac{b}{4}\right)(O_2 + 3.773N_2) = aCO_2 + \frac{b}{2}H_2O + 3.773\left(a + \frac{b}{4}\right)N_2 \tag{3.5}$$

Note that only the *ratios* of the numbers in front of the symbol for each chemical species are defined by Eq. (3.5) that is, only the relative proportions on a molar basis are obtained. Thus the fuel composition could have been written as CH_y where $y = b/a$.

Equation (3.5) defines the *stoichiometric* (or chemically correct or theoretical) proportions of fuel and air: that is, there is just enough oxygen for conversion of all the fuel into completely oxidized products. The stoichiometric air/fuel or fuel/air ratios (see Sec. 2.9) depend on fuel composition. From Eq. (3.5) :

$$\left(\frac{A}{F}\right)_s = \left(\frac{F}{A}\right)_s^{-1} = \frac{(1 + y/4)(32 + 3.773 \times 28.16)}{12.011 + 1.008y} = \frac{34.56(4 + y)}{12.011 + 1.008y} \tag{3.6}$$

The molecular weights of oxygen, atmospheric nitrogen, atomic carbon, and atomic hydrogen are, respectively, 32, 28.16, 12.011, and 1.008. $(A/F)_s$ depends only on y; Fig. 3.5 shows the variation in $(A/F)_s$ as y varies from 1 (e.g., benzene) to 4 (methane).

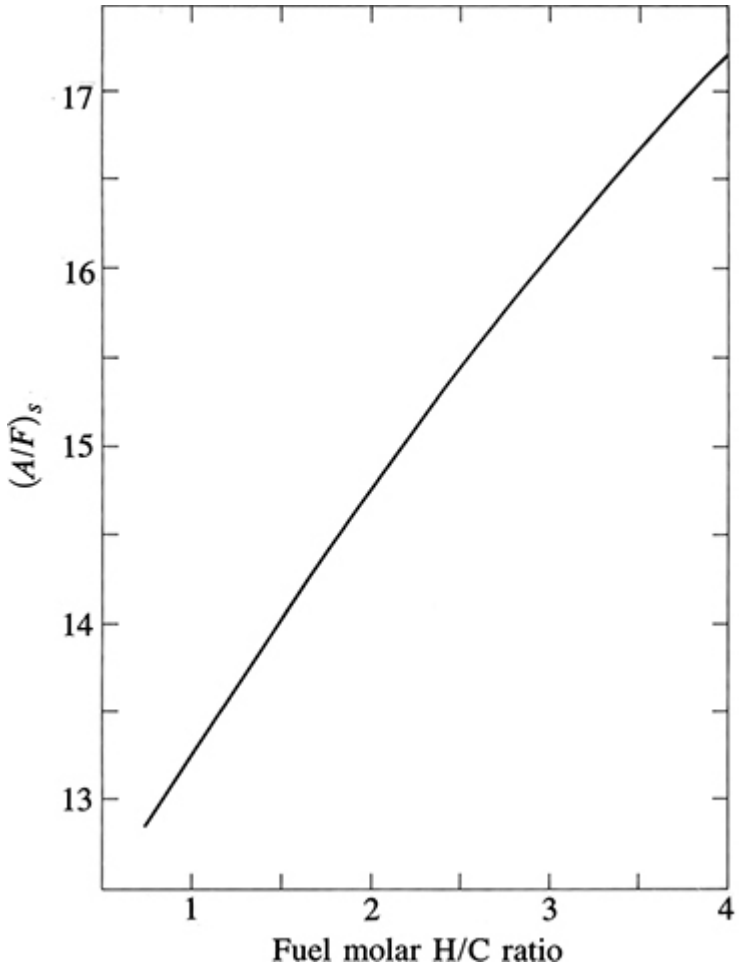


Figure 3.5 Stoichiometric air/fuel ratio for air-hydrocarbon fuel mixtures as a function of fuel molar H/C ratio.

EXAMPLE 3.1

A hydrocarbon fuel of composition 84.1% by mass C and 15.9% by mass H has a molecular weight of 114.15. Determine the number of moles of air required for stoichiometric combustion and the number of moles of products produced per mole of fuel. Calculate $(A/F)_s$, $(F/A)_s$, and the molecular weights of the reactants and the products.

Assume a fuel composition C_aH_b. The molecular weight relation gives

$$114.15 = 12.011a + 1.008b$$

The gravimetric analysis of the fuel gives

$$\frac{b}{a} = \frac{15.9/1.008}{84.1/12.011} = 2.25$$

Thus,

$$a = 8 \quad b = 18^b$$

The fuel is octane C_8H_{18}. Equation (3.5) then becomes

$$\overset{\text{Fuel}}{C_8H_{18}} + 12.5(O_2 + 3.773N_2) = 8CO_2 + \overset{\text{Products}}{9H_2O} + 47.16N_2$$

In moles:

$$1 + 12.5(1 + 3.773) = 8 + 9 + 47.16$$
$$1 + 59.66 = 64.16$$

Relative mass:

$$114.15 + 59.66 \times 28.96 = 8 \times 44.01 + 9 \times 18.02 + 47.16 \times 28.16$$
$$114.5 + 1727.8 = 1842.3$$

Per unit mass fuel:

$$1 + 15.14 = 16.4$$

Thus for stoichiometric combustion, 1 mole of fuel requires 59.66 moles of air and produces 64.16 moles of products. The stoichiometric $(A/F)_s$ is 15.14 and $(F/A)_s$ is 0.0661.

The molecular weights of the reactants M_R and products M_P are

$$M_R = \frac{1}{n}\sum n_i M_i = \frac{1}{60.66}(1\times 114.15 + 59.66\times 28.96)$$
$$M_p = \frac{1}{n}\sum n_i M_i = \frac{1}{64.16}(8\times 44.01 + 9\times 18.02 + 47.16\times 28.16)$$

Thus,

$$M_R = 30.36 \quad M_p = 28.71$$

Fuel-air mixtures with more than or less than the stoichiometric air requirement can be burned. With excess air or fuel-lean combustion, the extra air appears in the products in essentially unchanged form. For example, the combustion of isooctane with 25% excess air, or 1.25 times the stoichiometric air requirement, gives

$$C_8H_{18} + 1.25\times 12.5(O_2 + 3.773N_2) = 8CO_2 + 9H_2O + 3.13O_2 + 58.95N_2 \qquad (3.7)$$

With less than the stoichiometric air requirement, that is, with fuel-rich combustion, there is insufficient oxygen to oxidize fully the fuel C and H to CO_2 and H_2O. The products are a mixture of CO_2 and H_2O, with carbon monoxide CO and hydrogen H_2 (as well as N_2). The product composition cannot be determined from an element balance alone, and an additional assumption about the chemical composition of the product species must be made (see Secs. 4.2 and 4.9.2).

Because the composition of the combustion products is significantly different for fuel-lean and fuel-rich mixtures, and because the stoichiometric fuel/air ratio depends on fuel composition, the ratio of the actual fuel/air ratio to the stoichiometric ratio (or the inverse of this ratio) is a more informative parameter for defining mixture composition. The *fuel/air equivalence ratio* ϕ,

$$\phi = \frac{(F/A)_{\text{actual}}}{(F/A)_s} \qquad (3.8)$$

will be used throughout this text for this purpose: The inverse of ϕ, the *relative air/fuel ratio* λ,

$$\lambda = \phi^{-1} = \frac{(A/F)_{\text{actual}}}{(A/F)_s} \tag{3.9}$$

is also used extensively. These parameters define the following fuel-air mixture stoichiometry regimes:

For fuel-lean mixtures: $\phi < 1, \lambda > 1$
For stoichiometric mixtures: $\phi = \lambda = 1$
For fuel-rich mixtures: $\phi > 1, \lambda < 1$

When the fuel contains oxygen (e.g., with alcohols), the procedure for determining the overall combustion equation is the same except that fuel oxygen is included in the oxygen balance between reactants and products. For methyl alcohol (methanol), CH_3OH, the stoichiometric combustion equation is

$$CH_3OH + 1.5(O_2 + 3.773N_2) = CO_2 + 2H_2O + 5.66N_2 \tag{3.10}$$

and $(A/F)_s = 6.47$. For ethyl alcohol (ethanol), C_2H_5OH, the stoichiometric combustion equation is

$$C_2H_5OH + 3(O_2 + 3.773N_2) = 2CO_2 + 3H_2O + 11.32N_2 \tag{3.11}$$

and $(A/F)_s = 9.00$.

If there are significant amounts of sulfur in the fuel, the appropriate oxidation product for determining the stoichiometric air and fuel proportions is sulfur dioxide, SO_2.

For hydrogen fuel, the stoichiometric equation is

$$H_2 + \frac{1}{2}(O_2 + 3.773N_2) = H_2O + 1.887N_2 \tag{3.12}$$

and the stoichiometric (A/F) ratio is 34.3.

Note that the actual composition of the products of combustion in Eqs. (3.7) and (3.10) to (3.12) may not occur in practice. At normal combustion temperatures, significant dissociation of CO_2 and of H_2O occurs (see Sec. 3.7.1). At lower temperatures, how close recombination brings the product

composition to that indicated by these overall chemical equations depends on the rate of cooling of the product gases. More general relationships for the composition of unburned and burned gas mixtures are developed in Chap. 4 .

The stoichiometric (A/F) and (F/A) ratios of common fuels and representative single hydrocarbon and other compounds are given in App. D along with other fuel data. Figure 3.5 shows how this stoichiometric $(A/F)_s$ varies with fuel H:C ratio.

3.5 THE FIRST LAW OF THERMODYNAMICS AND COMBUSTION [c]

3.5.1 Energy and Enthalpy Balances

In a combustion process, fuel and oxidizer react to produce products of different composition. The actual path by which this transformation takes place is understood only for relatively simple fuels such as hydrogen, methane, *n*-heptane, isooctane, toluene, methanol, and ethanol. For fuel mixtures such as gasoline, or with more complicated molecular structure, the details are not well defined. Nonetheless, the first law of thermodynamics can be used to relate the end states of mixtures undergoing a combustion process; its application does not require that the details of the process be known.

The first law of thermodynamics relates changes in internal energy (or enthalpy) to heat and work transfer interactions. In applying the first law to a system whose chemical composition changes, care must be exercised in relating the reference states at which zero internal energy or enthalpy for each species or groups of species are assigned. We are not free, when chemical reactions occur, to choose independently the zero internal energy or enthalpy reference states of chemical substances transformed into each other by reaction.

Consider a system of mass m that changes its composition from reactants to products by chemical reaction as indicated in Fig. 3.6. Applying the first law to the system between its initial and final states gives

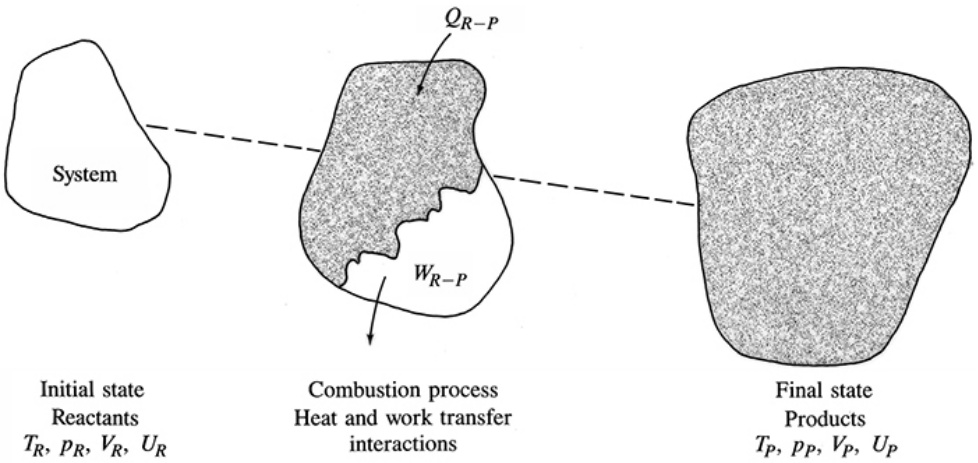


Figure 3.6 System changing from reactants to products for first law analysis.

$$Q_{R-P} - W_{R-P} = U_P - U_R \tag{3.13}$$

Heat transfer Q_{R-P}, and work transfer W_{R-P} due to normal force displacements, may occur across the system boundary. The standard thermodynamic sign convention for each energy transfer interaction—positive for heat transfer *to* the system and positive for work transfer *from* the system—is used.

We will consider a series of special processes: first, a *constant volume* process where the initial and final temperatures are the same, T'. Then Eq. (3.13) becomes

$$Q_{R-P} = U'_P - U'_R = (\Delta U)_{V,T'} \tag{3.14}$$

The internal energy of the system has changed by an amount $(\Delta U)_{V,T'}$, which can be measured or calculated. Since combustion processes are exothermic, Q_{R-P} and $(\Delta U)_{V,T'}$ are negative; therefore, the system's internal energy decreases. If Eq. (3.14) is expressed per mole of fuel, then $(\Delta U)_{V,T'}$ is known as the increase in internal energy at constant volume, and - $(\Delta U)_{V,T'}$ is known as the *heat of reaction at constant volume* at temperature T'.

Next, consider a *constant pressure* process where the initial and final temperatures are the same, T'. For a constant pressure process

$$W_{R-P} = \int_R^P p\, dV = p(V_P - V_R) \tag{3.15}$$

so Eq. (3.13) becomes

$$Q_{R-P} - p(V_P' - V_R') = U_P' - U_R'$$

or

$$\begin{aligned} Q_{R-P} &= p(U_P' - pV_P') - U_R' + pV_R') \\ &= H_P' - H_R' = (\Delta H)_{p,T'} \end{aligned} \tag{3.16}$$

The enthalpy of the system has changed by an amount $(\Delta H)_{p,T'}$, which can be measured or calculated. Again, for combustion reactions, $(\Delta H)_{p,T'}$ is a negative quantity. If Eq. (3.16) is written per mole of fuel, $(\Delta H)_{p,T'}$ is called the increase in enthalpy at constant pressure and $-(\Delta H)_{p,T'}$ is called the *heat of reaction at constant pressure* at T'.

These processes can be displayed, respectively, on the internal energy or enthalpy versus temperature plot shown schematically in Fig. 3.7. If U (or H) for the reactants is arbitrarily assigned a value U_R° (or H_R°) (or $(\Delta H)_{p,T_0}$) at some reference temperature $(\Delta U)_{V,T_0}$, then the value of $U(T)$ or $H(T)$ fixes the relationship between $U(T)$ or $H(T)$, respectively, for the products and the reactants. Note that the slope of these lines (the specific heat at constant volume or pressure, if the vertical axis is expressed per unit mass or per mole) increases with increasing temperature; also, the magnitude of $(\Delta U)_{V,T'}$ or $(\Delta H)_{p,T'}$ decreases with increasing temperature because c_v (or c_p) for the products is greater than that for the reactants.

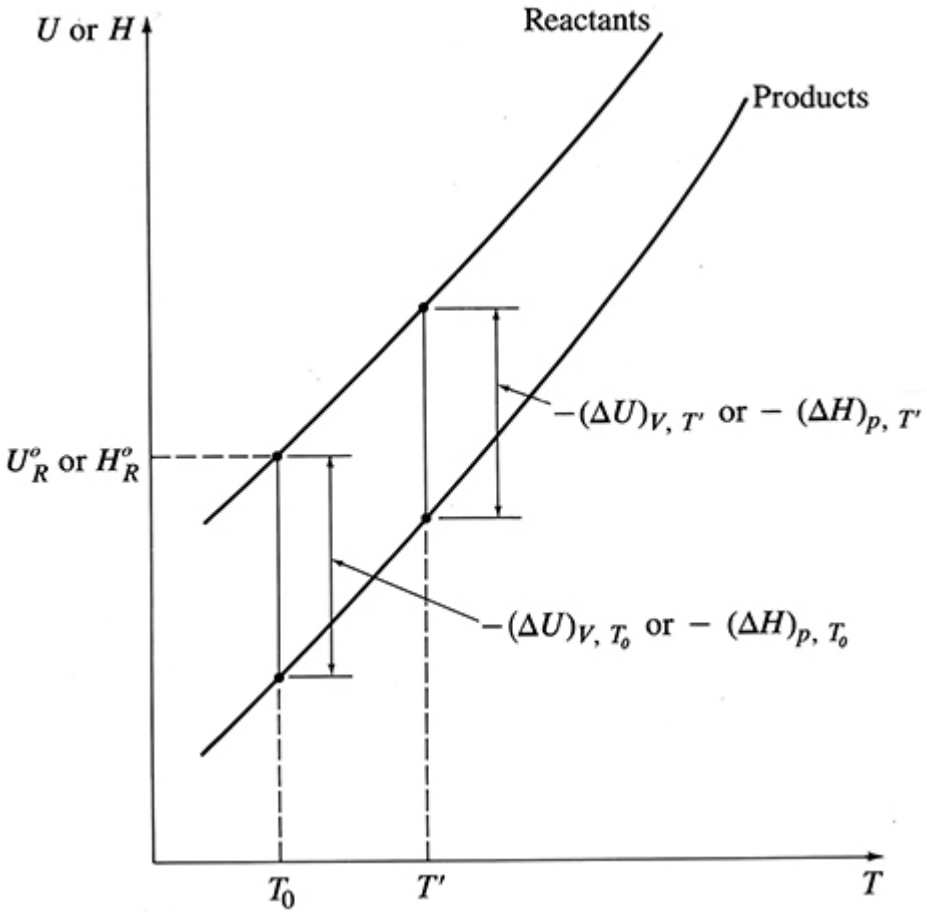


Figure 3.7 Schematic plot of internal energy (U) or enthalpy (H) of reactants and products as a function of temperature.

The difference between $(\Delta H)_{p,T'}$ and $(\Delta U)_{V,T'}$ is

The difference between $(\Delta H)_{p,T'}$ and $(\Delta U)_{V,T'}$ is

$$(\Delta H)_{p,T'} - (\Delta U)_{V,T'} = p(V_P - V_R) \tag{3.17}$$

Only if the volumes of the products and reactants in the constant pressure process are the same, are $(\Delta H)_{p,T'}$ and $(\Delta U)_{V,T'}$ equal. If all the reactant and product species are ideal gases, then the ideal gas law Eq. (3.1) gives

$$(\Delta H)_{p,T'} - (\Delta U)_{V,T'} = \tilde{R}(n_P' - n_R')T' \tag{3.18}$$

Note that any inert gases do not contribute to $(n_P' - n_R')$.

With a hydrocarbon fuel, one of the products, H_2O, can be in the gaseous or liquid phase. The internal energy (or enthalpy) of the products in the constant volume or constant pressure processes described above in Fig. 3.7 will depend on the relative proportions of the water in the gaseous and liquid phases. The limiting cases of all vapor and all liquid are shown in Fig. 3.8 *a* for a *U*-*T* plot. The internal energy differences between the curves is

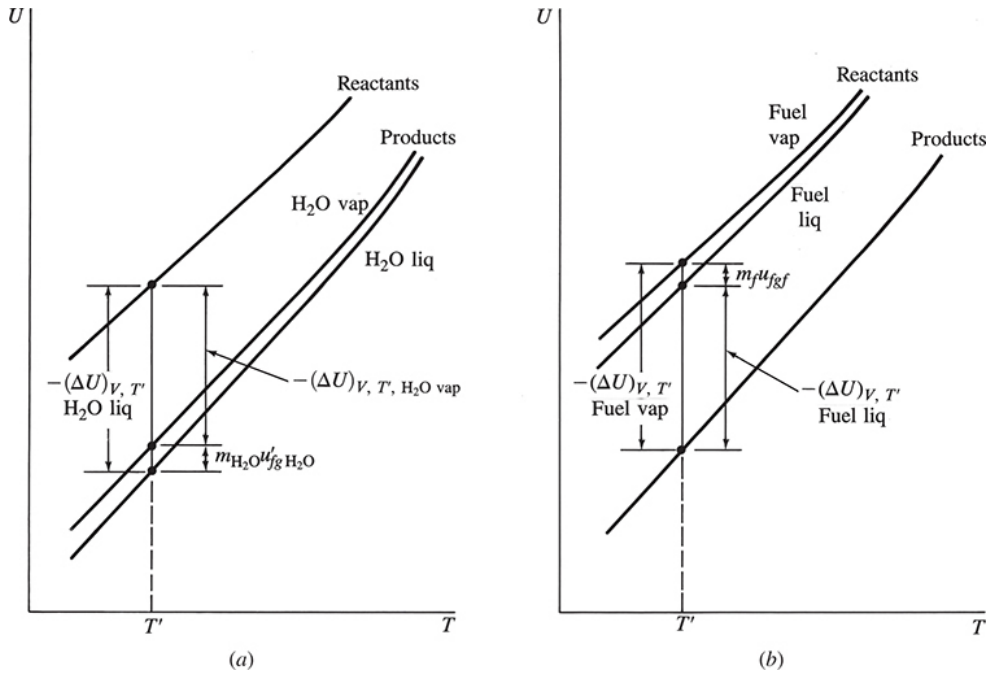


Figure 3.8 Schematic plots of internal energy of reactants and products as a function of temperature: (*a*) Effect of water in products as either vapor or liquid; (*b*) effect of fuel in reactants as either vapor or liquid.

$$\left|(\Delta U)_{V,T',\mathrm{H_2O\,liq}}\right| - \left|(\Delta U)_{V,T',\mathrm{H_2O\,vap}}\right| = m_{\mathrm{H_2O}}\, u'_{fg\,\mathrm{H_2O}} \tag{3.19}$$

where $m_{\mathrm{H_2O}}$ is the mass of water in the products and $u'_{fg\,\mathrm{H_2O}}$ is the internal energy of vaporization of water at the temperature and pressure of the products. Similar curves and relationships apply for enthalpy:

$$\left|(\Delta H)_{p,T',\mathrm{H_2O\,liq}}\right| - \left|(\Delta H)_{p,T',\mathrm{H_2O\,vap}}\right| = m_{\mathrm{H_2O}}\, h'_{fg\,\mathrm{H_2O}} \tag{3.20}$$

For some fuels, the reactants may contain the fuel as either liquid or vapor. The *U–T* (or *H–T*) line for the reactants with the fuel as liquid or as vapor will be different, as indicated in Fig. 3.8 *b*. The vertical distance between the two reactant curves is $m_f u_{fg\,f}$ (or $m_f h_{fg\,f}$) where the subscript *f* denotes fuel.

3.5.2 Enthalpies of Formation

For fuels that are single hydrocarbon compounds, or where the precise fuel composition is known, the internal energies or enthalpies of the reactants and the products can be related through the enthalpies of formation of the reactants and products.

The *enthalpy of formation* $\Delta\tilde{h}_f^\circ$ of a chemical compound is the enthalpy increase associated with the reaction of forming one mole of the given compound from its elements, with each substance in its thermodynamic standard state at the given temperature.

The *standard state* is the state at one atmosphere pressure and the temperature under consideration. We will denote the standard state by the superscript °.

Since thermodynamic calculations are made as a difference between an initial and a final state, it is necessary to select a *datum state* to which all other thermodynamic states can be referred. While a number of datum states have been used in the literature, the most common datum is 298.15 K (25°C) and 1 atmosphere. We will use this datum throughout this text. Elements at their *reference state* are arbitrarily assigned zero enthalpy at the datum temperature. The reference state of each element is its stable standard state: for example, for oxygen at 298.15 K, the reference state is O_2 gas.

Enthalpies of formation are tabulated as a function of temperature for all commonly occurring species. For inorganic compounds, the *JANAF Thermochemical Tables* are the primary reference source. [8] These tables include values of the molar specific heat at constant pressure, standard entropy, standard Gibbs free energy (called free energy in the tables), standard enthalpy, enthalpy of formation and Gibbs free energy of formation, and $\log_{10}$ equilibrium constant for the formation of each species from its elements. Some primary references for thermodynamic data on fuel compounds are Maxwell, [9] Rossini et al., [10] and Stull et al . [11] Enthalpies of

formation of species relevant to hydrocarbon fuel combustion are tabulated in Table 3.2. Selected values of thermodynamic properties of relevant species are tabulated in App. D.

TABLE 3.2 Standard enthalpies of formation

Species	State*	$\Delta\tilde{h}_f^\circ$, MJ/kmol
O_2	Gas	0
N_2	Gas	0
H_2	Gas	0
C	Gas	0
CO_2	Gas	−393.52
H_2O	Gas	−241.83
H_2O	Liquid	−285.84
CO	Gas	−110.54
CH_4	Gas	−74.87
C_3H_8	Gas	−103.85
CH_3OH	Gas	−201.17
CH_3OH	Liquid	−238.58
C_8H_{18}†	Gas	−224.1
C_8H_{18}†	Liquid	−259.3

*At 298.15 K (25°C) and 1 atm.
†Isooctane.

For a given combustion reaction, the enthalpy of the products at the standard state relative to the enthalpy datum is then given by

$$H_P^\circ = \sum_{\text{products}} n_i \Delta\tilde{h}_{f,i}^\circ \qquad (3.21a)$$

and the enthalpy of the reactants is given by

$$H_R^\circ = \sum_{\text{reactants}} n_i \Delta \tilde{h}_{f,i}^\circ \qquad (3.21b)$$

The enthalpy increase, $(\Delta H)_{p,T_0}$, is then obtained from the difference $(H_P^\circ - H_R^\circ)$. The internal energy increase can be obtained with Eq. (3.17) .

EXAMPLE 3.2

Calculate the enthalpy of the products and reactants, and the enthalpy increase and internal energy increase of the reaction, of a stoichiometric mixture of methane and oxygen at 298.15 K.

The stoichiometric reaction is

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

For H_2O gas, from Table 3.2 and Eqs. (3.21 *a* and *b*):

$$H_P^\circ = -393.52 + 2(-241.83) = -877.18 \text{ MJ/kmol } CH_4$$

For H_2O liquid:

$$H_P^\circ = -393.52 + 2(-285.84) = -965.20 \text{ MJ/kmol } CH_4$$
$$H_R^\circ = -74.87 \text{ MJ/kmol } CH_4$$

Hence for H_2O gas:

$$(\Delta H)_P^\circ = -877.18 + 74.87 = -802.31 \text{ MJ/kmol } CH_4$$

and for H_2O liquid:

$$(\Delta H)_P^\circ = -965.20 + 74.87 = -890.33 \text{ MJ/kmol } CH_4$$

Use Eq. (3.18) to find $(\Delta U)_V^\circ$. With H_2O gas, the number of moles of reactants and products are equal, so

$$(\Delta U)_V^\circ = (\Delta H)_P^\circ = -802.3 \text{ MJ/kmol } CH_4$$

For H_2O liquid:

$$(\Delta H)_V^\circ = -890.33 - 8.3143 \times 10^{-3}(1-3)298.15 \text{ MJ/kmol } CH_4$$
$$(\Delta U)_V^\circ = -885.4 \text{ MJ/kmol } CH_4$$

Note that the presence of nitrogen in the mixture or oxygen in excess of the stoichiometric amount would not change any of these calculations.

3.5.3 Heating Values

For fuels where the precise fuel composition is not known, the enthalpy of the reactants cannot be determined from the enthalpies of formation of the reactant species. The *heating value* of the fuel is then measured directly.

The heating value Q_{HV} or calorific value of a fuel is the magnitude of the heat of reaction at constant pressure or at constant volume at a standard temperature [usually 25°C (77°F)] for the complete combustion of unit mass of fuel. Thus

$$Q_{HV_p} = -(\Delta H)_{p,T_0} \tag{3.22a}$$

and

$$Q_{HV_v} = -(\Delta U)_{v,T_0} \tag{3.22b}$$

Complete combustion means that all carbon is converted to CO_2, all hydrogen is converted to H_2O, and any sulfur present is converted to SO_2. The heating value is usually expressed in joules per kilogram or joules per kilomole of fuel (British thermal units per pound-mass or British thermal units per pound-mole). It is therefore unnecessary to specify how much oxidant was mixed with the fuel, though this must exceed the stoichiometric requirement. It is immaterial whether the oxidant is air or oxygen.

For fuels containing hydrogen, we have shown that whether the H_2O in the products is in the liquid or gaseous phase affects the value of the heat of reaction. The term *higher heating value* Q_{HHV} (or gross heating value) is used when the H_2O formed is all condensed to the liquid phase; the term

lower heating value Q_{LHV} (or net heating value) is used when the H_2O formed is all in the vapor phase. The two heating values at constant pressure are related by

$$Q_{\text{HHV}_p} = Q_{\text{LHV}_p} + \left(\frac{m_{\text{H}_2\text{O}}}{m_f}\right) h_{fg\text{H}_2\text{O}} \tag{3.23}$$

where $(m_{\text{H}_2\text{O}}/m_f)$ is the ratio of mass of H_2O produced to mass of fuel burned. A similar expression with $u_{fg\text{H}_2\text{O}}$ replacing $h_{fg\text{H}_2\text{O}}$ applies for the higher and lower heating value at constant volume. The difference between these two heating values for hydrocarbon-based fuels is in the 4 to 11% range, depending on C:H ratio.

The heating value at constant pressure is the more commonly used one; often the qualification "at constant pressure" is omitted. The difference between the heating values at constant pressure and constant volume is small.

Heating values [d] of fuels are measured in calorimeters. For gaseous fuels, it is most convenient and accurate to use a continuous-flow atmosphere pressure calorimeter. The entering gaseous fuel is saturated with water vapor and mixed with sufficient saturated air for complete combustion at the reference temperature. The mixture is burned in a burner and the combustion products cooled with water-cooled metal tube coils to close to the inlet temperature. The heat transferred to the cooling water is calculated from the measured water flow rate and water temperature rise. The heating value determined by this process is the higher heating value at constant pressure.

For liquid and solid fuels, it is more satisfactory to burn the fuel with oxygen under pressure at constant volume in a calorimeter. A sample of the fuel is placed in the calorimeter, which is a stainless-steel container immersed in cooling water at the standard temperature. Sufficient water is placed in the calorimeter to ensure that the water produced in the combustion process will condense. Oxygen at 30 atmospheres is admitted into the calorimeter. A length of firing cotton is suspended into the sample from an electrically heated wire filament to act as a source of ignition. When combustion is complete, and the combustion products and calorimeter have thermally equilibrated, the temperature rise of the calorimeter and its cooling water is measured. The heating value determined by this process is the higher heating value at constant volume.

The heating values of common fuels are tabulated with other fuel data in App. D. The following example illustrates how the enthalpy of a reactant mixture relative to the enthalpy datum we have defined can be determined from the measured heating value of the fuel.

EXAMPLE 3.3

Liquid kerosene fuel with a lower heating value (determined in a calorimeter) of 43.2 MJ/kg and average molar H/C ratio of 2 is mixed with the stoichiometric air requirement at 298.15 K. Calculate the enthalpy of the reactant mixture relative to the datum of zero enthalpy for C, O_2, N_2, and H_3 at 298.15 K.

The combustion equation per mole of fuel can be written as

$$CH_2 + \tfrac{3}{2}(O_2 + 3.773N_2) = CO_2 + H_2O + 5.660N_2$$

or

$$14\text{ kg fuel} + \left.\begin{matrix}7.160\text{ kmol}\\207.4\text{ kg}\end{matrix}\right\}\text{air} \rightarrow \left.\begin{matrix}7.66\text{ kmol}\\221.4\text{ kg}\end{matrix}\right\}\text{products}$$

where $M = 28.962$ for atmospheric air.

The heating value given is at constant volume, $-(\Delta U)^\circ_V$. $-(\Delta H)^\circ_p$ is obtained from Eq. (3.18) , noting that the fuel is in the liquid phase:

$$(\Delta H)^\circ_p = -43.2 + 8.3143\times10^{-3}(7.66 - 7.160)\times\frac{298.15}{14}$$
$$= -43.2 + 0.09 = -43.1\text{ MJ/kg fuel}$$

The enthalpy of the products per kilogram of mixture is found from the enthalpies of formation of CO_2 and H_2O vapor given in Table 3.2:

$$h_p = \frac{1\times(-393.52) + 1\times(-241.83)}{221.4}$$
$$= -2.87\text{ MJ/kg}$$

The enthalpy of the reactants per kilogram of mixture is then

$$h_R = h_P - (\Delta h)_p^\circ = -2.87 + \frac{43.1 \times 14}{221.4} = -0.145 \text{ MJ/kg}$$

3.5.4 Adiabatic Combustion Processes

We now use the relationships developed above to examine two other special processes important in engine analysis: constant-volume and constant-pressure adiabatic combustion. For an adiabatic constant-volume process, Eq. (3.13) becomes

$$U_P - U_R = 0 \tag{3.24}$$

when U_P and U_R are evaluated relative to the same datum (e.g., the enthalpies of C, O_2, N_2, and H_2 are zero at 298.15 K, the datum used throughout this text).

Frequently, however, the data on internal energy or enthalpy for species and reactant or product mixtures that are available give internal energies or enthalpies relative to the species or mixture value at some reference temperature T_0: e.g., $U(T) - U(T_0)$ or $H(T) - H(T_0)$ are tabulated or graphed. Since

$$U_P(T_0) - U_R(T_0) = (\Delta U)_{V,T_0}$$

it follows from Eq. (3.24) that

$$[U_P(T) - U_P(T_0)] - [U_R(T) - U_R(T_0)] = -(\Delta U)_{V,T_0} \tag{3.25}$$

relates the product and reactant states. Figure 3.9 illustrates the adiabatic constant-volume combustion process on a U-T diagram. Given the initial state of the reactants (T_R, V), we can determine the final state of the products (T_P, P).

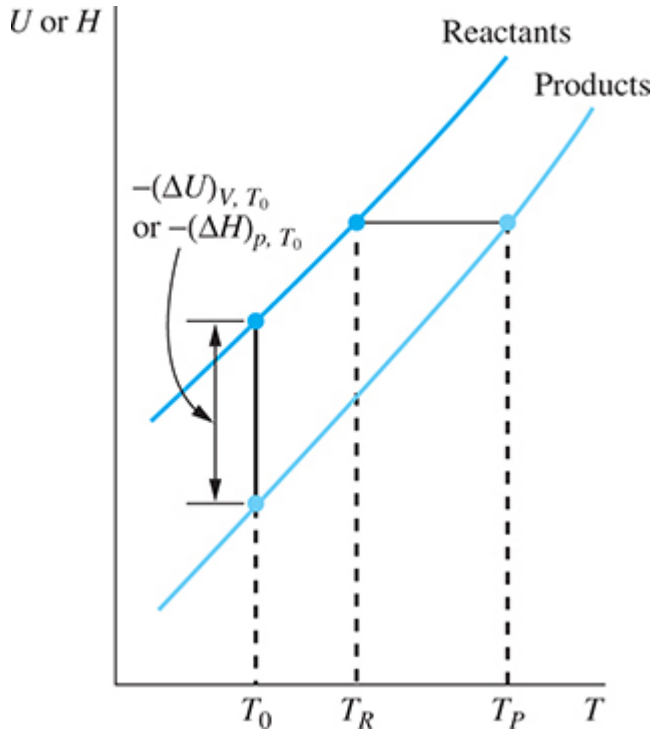


Figure 3.9 Adiabatic constant-volume combustion process on U-T diagram or adiabatic constant-pressure combustion process on H-T diagram.

For an adiabatic constant-pressure combustion process, Eq. (3.13) gives

$$H_P - H_R = 0$$

which combines with Eq. (3.16) to give

$$[H_p(T) - H_p(T_0)] - [H_R(T) - H_R(T_0)] = -(\Delta H)_{p,T_0} \tag{3.26}$$

Figure 3.9 illustrates this process also. Given the initial reactant state (T_R, p), we can determine the final product state (T_p, p).

Note that while in Figs. 3.7, 3.8, and 3.9 we have shown U and H for the reactants and products to be functions of T only, in practice for the products at high temperature, U and H will be functions of p and T. The analysis presented here is general; however, to determine the final state of the products in an adiabatic combustion process, the constant-volume or constant-pressure constraint must also be used explicitly.

The final temperature of the products in an adiabatic combustion process is called the *adiabatic flame temperature*. Examples of typical adiabatic flame temperatures are shown later in Fig. 3.13.

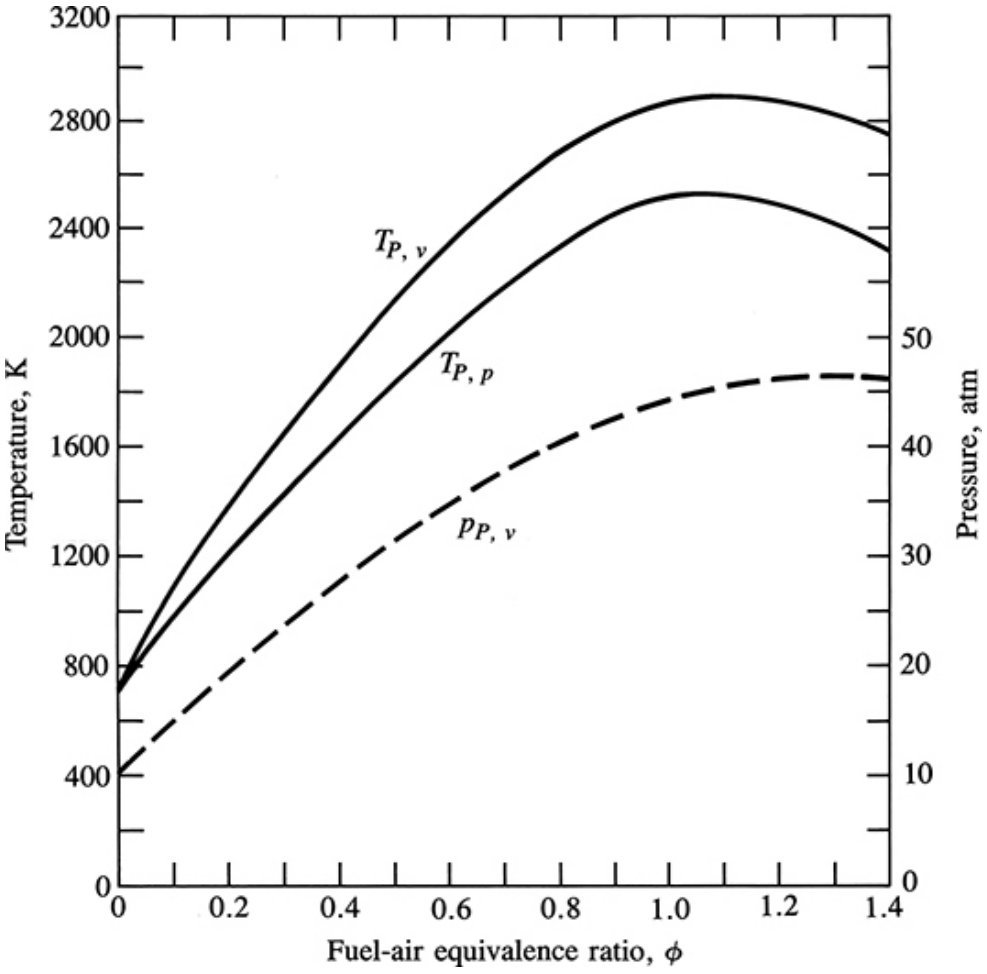


Figure 3.13 Equilibrium combustion product gas temperatures for constant-volume (T_{pv}) and constant-pressure (T_{pp}) adiabatic combustion of isooctane-air mixture initially at 700 K and 10 atm (typical end-of-compression mixture conditions in a gasoline SI engine), as a function of fuel/air equivalence ratio. Pressure ($p_{p,v}$) is equilibrium pressure for adiabatic constant-volume combustion.

3.5.5 Combustion Efficiency of an Internal Combustion Engine

In practice, the exhaust gas of an internal combustion engine contains incomplete combustion products (e.g., CO, H_2, unburned hydrocarbons, soot) as well as complete combustion products (CO_2 and H_2O); see Sec. 4.9 . Under lean operating conditions, the amounts of incomplete combustion products are small. Under fuel-rich operating conditions, these amounts become more substantial since there is insufficient oxygen to complete combustion. Because a fraction of the fuel's chemical energy is not fully released inside the engine during the combustion process, it is useful to define a *combustion efficiency*. The engine can be analyzed as an open system that exchanges heat and work with its surrounding environment (the atmosphere). Reactants (fuel and air) flow into the system; products (exhaust gases) flow out. Consider a mass m that passes through the control volume surrounding the engine shown in Fig. 3.10; the *net chemical energy release* due to combustion within the engine is given by

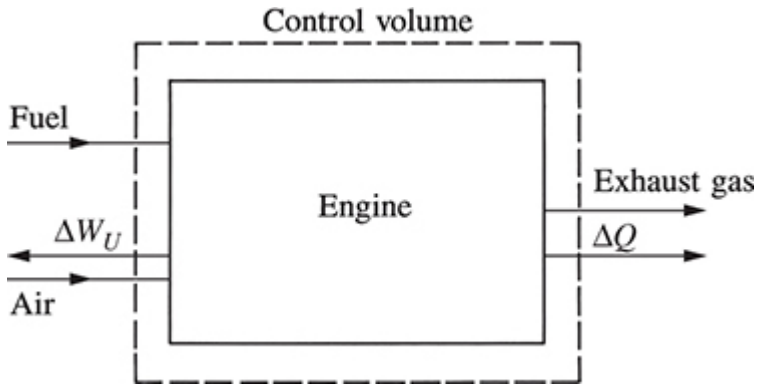


Figure 3.10 Control volume surrounding engine. ΔW_U is the useful work produced by the engine; ΔQ is the heat lost from the cylinder charge to the walls.

$$[H_R(T_A) - H_p(T_A)] = m\left(\sum_{i,\text{reactants}} n_i \Delta\tilde{h}^\circ_{f,i} - \sum_{i,\text{products}} n_i \Delta\tilde{h}^\circ_{f,i}\right)$$

Enthalpy is the appropriate property since $p_R = p_P = p_{\text{atm}}$: n_i is the number of moles of species i in the reactants or products per unit mass of working fluid, and $\Delta\tilde{h}^\circ_{f,i}$ is the standard enthalpy of formation of species i at ambient temperature T_A.

The maximum amount of fuel energy *supplied* to the control volume around the engine that can be released by combustion is $m_f Q_{\mathrm{HV}}$. Hence, the *combustion efficiency*—the fraction of the fuel energy supplied that is released in the combustion process—is given by [12]

$$\eta_c = \frac{H_R(T_A) - H_P(T_A)}{m_f Q_{\mathrm{HV}}} \tag{3.27}$$

Note that m and m_f can be replaced by the average mass flow rates $\dot{m}$ and $\dot{m}_f$.

Figure 3.11 shows how combustion efficiency varies with the fuel/air equivalence ratio for internal combustion engines. For spark-ignition engines, for lean equivalence ratios, the combustion efficiency is usually in the range of 95 to 98%. For mixtures richer than stoichiometric, lack of oxygen prevents complete combustion of the fuel carbon and hydrogen, and the combustion efficiency steadily decreases as the mixture becomes richer. Combustion efficiency is little affected by other engine operating and design variables, provided the engine combustion process remains stable. For diesel engines, which always operate lean, the combustion efficiency is slightly higher—about 98%. Details of exhaust gas composition, on which these combustion efficiencies are based, can be found in Sec. 4.9 . There, in Eq. (4.67) , Eq. (3.27) is expressed in terms of the concentrations of the incomplete combustion species (e.g., CO, H_2, and HC) and their heating values.

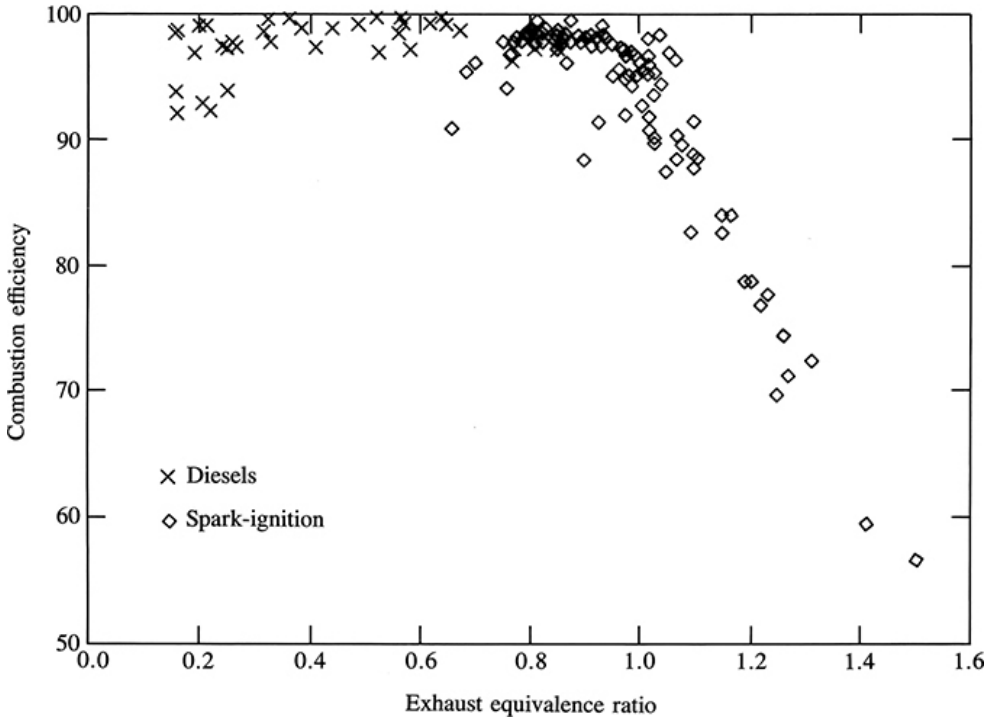


Figure 3.11 Variation of engine combustion efficiency with fuel/air equivalence ratio.

3.6 THE SECOND LAW OF THERMODYNAMICS APPLIED TO COMBUSTION

3.6.1 Entropy

In App. B, it is shown how the entropy of a mixture of ideal gases of known composition can be calculated. The discussion earlier relating the enthalpies (or internal energies) of the reactant and the product mixtures applies to entropy also. For example, the standard-state entropies of chemical species are tabulated in the JANAF [8] tables relative to zero entropy at 0 K. If the entropies of the elements at a datum-state temperature and pressure are (arbitrarily) set equal to zero, then the values of the entropy of a reactant mixture of given composition, temperature and pressure, and of the product

mixture of given composition, temperature and pressure, can both be determined.

3.6.2 Maximum Work from an Internal Combustion Engine and Efficiency

An internal combustion engine can be analyzed as an open system that exchanges heat and work with its surrounding environment (the atmosphere). Reactants (fuel and air) flow into the system; products (exhaust gases) flow out. By applying the second law of thermodynamics to a control volume surrounding the engine, as illustrated in Fig. 3.10, we can derive an expression for the maximum useful work that the engine can deliver.

Consider a mass m of fluid as it passes through the control volume surrounding the engine. The first law gives

$$\Delta Q - \Delta W_U = \Delta H$$

where ΔW_U is the useful work transfer (i.e., non- pdV work) to the environment and $\Delta H = H_P - H_R$. Since the heat transfer ΔQ occurs only with the atmosphere at temperature T_A, from the second law

$$\frac{\Delta Q}{T_A} \leq \Delta S$$

These equations combine to give

$$\Delta W_U \leq -(\Delta H - T_A \Delta S) = -\Delta B$$

where B is the steady-flow availability function, $H - T_A S$. [13] Usually, for the entering gases, $p_R = p_A$ and $T_R = T_A$, and the maximum work will be obtained when $p_P = p_A$ and $T_p = T_A$.

Under these conditions,

$$\Delta W_U \leq -\left[(H - TS)_{P_{T_A, P_A}} - (H - TS)_{R_{T_A, P_A}}\right] = -(\Delta G)_{T_A, P_A} \tag{3.28}$$

or

$$\Delta W_{U\,\max} = -(\Delta G)_{T_A, p_A}$$

G is the Gibbs free energy, $H - TS$, and $(\Delta G)_{T_A, p_A}$ is the Gibbs free energy increase in the reaction of the fuel-air mixture to products at atmospheric temperature and pressure. $-(\Delta G)_{T_A, p_A}$ will be a maximum when combustion is complete.

A fundamental measure of the effectiveness of any practical internal combustion engine is the ratio of the actual work delivered compared with this maximum work. This ratio is called the *availability conversion efficiency* η_a:

$$\eta_a = \frac{\Delta W}{\Delta W_{U\,\max}} = -\frac{\Delta W}{(\Delta G)_{T_A, p_A}} \tag{3.29}$$

The property *availability* is the maximum useful work transfer that can be obtained from a system atmosphere or control-volume atmosphere combination. This efficiency therefore defines the fraction of the availability of the unburned fuel-air mixture which, passing through the engine and interacting only with the atmosphere, is actually converted to useful work. Availability analysis of engine operation is proving valuable in identifying where the significant irreversibilities or losses in availability occur. This topic is discussed more fully in Sec. 5.7 .

$(\Delta G)_{T_A, p_A}$, or $(\Delta g)_{T_A, p_A}$, is not an easy quantity to evaluate for practical fuels; it is the heating value, $-(\Delta h)_{T_A}$, which is usually measured. Values of $(\Delta g)^\circ_{298}$ and (Δh°_{298}) for selected fuel combustion reactions are given in Table 3.3. For the pure hydrocarbons, they are closely comparable because at 298 K, $\Delta \tilde{s}^\circ << \Delta \tilde{h}^\circ / T$. For alcohols the difference is larger, 7% rather than about 1.5%. Because for practical fuels $-(\Delta h)^\circ_{298}$ is measured directly as the heating value of the fuel, it is standard practice to use the following definition of efficiency:

TABLE 3.3 Enthalpies and free energies of combustion reactions

Reaction*	$\Delta\tilde{h}^\circ_{298}$, MJ/kmol	$\Delta\tilde{g}^\circ_{298}$, MJ/kmol
$C + O_2 \rightarrow H_2O$	−393.52	−394.40
$H_2 + \frac{1}{2}O_2 \rightarrow H_2O$	−240.91	−232.78
$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$	−802.30	−800.76
$CH_4O(l) + \frac{3}{2}O_2 \rightarrow CO_2 + 2H_2O$	−638.59	−685.35
$C_3H_8(g) + 5O_2 \rightarrow 3CO_2 + 4H_2O$	−2044.0	−2074.1
$C_6H_6(l) + \frac{15}{2}O_2 \rightarrow 6CO_2 + 3H_2O$	−3135.2	−3175.1
$C_8H_{18}(l) + \frac{25}{2}O_2 \rightarrow 8CO_2 + 9H_2O$	−5074.6	−5219.9

*In the products, H_2O is a gas.

$$\eta_f = \frac{W_c}{m_f Q_{HV}} \tag{3.30}$$

which was defined as the *fuel conversion efficiency* in Sec. 2.8 . Note that sometimes the higher heating value is used in Eq. (3.30) and sometimes the lower heating value. Whichever value is used should be explicitly stated. The normal practice in internal combustion engine analysis is to use the lower heating value at constant pressure, since the engine overall is a steady flow device and the water in the exhaust is normally in vapor form. We will use $Q_{\text{LHV},p}$ in Eq. (3.30) throughout this text. The fuel conversion efficiency is the most commonly used definition of engine efficiency because it uses an easily measured quantity, the heating value, to define the usable fuel energy supplied to the engine. For hydrocarbon fuels, since $\Delta\tilde{h}^\circ \approx \Delta\tilde{g}^\circ$ the fuel conversion efficiency and the availability conversion efficiency are closely comparable in value.

In practice, not all the fuel energy supplied to the engine is released by the combustion process since combustion is incomplete: the combustion efficiency [Eq. (3.27)] is less than unity. It is sometimes useful to separate out the effects of incomplete combustion by defining an efficiency that relates the actual work per cycle to the amount of fuel chemical energy released in the combustion process. We will call this the *thermal conversion efficiency* η_t:

$$\eta_t = \frac{W_c}{H_R(T_A) - H_p(T_A)} = -\frac{W_c}{(\Delta H)_{T_A}} = \frac{W_c}{\eta_c m_f Q_{HV}} \tag{3.31}$$

Obviously the fuel conversion, thermal conversion, and combustion efficiencies are related by

$$\eta_f = \eta_c \eta_t \tag{3.32}$$

It is important to understand that there is a fundamental difference between the availability conversion efficiency of an internal combustion engine as defined by Eq. (3.29) [and the fuel conversion efficiency, Eq. (3.30) , which closely approximates it], and the efficiency of a thermodynamic heat engine. The second law limit to the availability conversion efficiency is unity. For a thermodynamic heat engine (defined as a system executing cycles experiencing heat-transfer interactions with two heat reservoirs, one at a higher and one at a lower temperature), the efficiency is limited to a value substantially less than unity by the temperatures of the heat reservoirs. [13] The maximum value of the efficiency of such engines is the Carnot cycle efficiency, $1-(T_{low}/T_{high})$. However, internal combustion engines are steady-flow devices, interacting with the atmosphere, which produce work, where fuel and air flow in and burned gases flow out. They are *not heat engines*—systems (working fluids) executing cycles, interacting with heat reservoirs. So the Carnot cycle efficiency limit has no direct relevance to the efficiency of internal combustion engines.

3.7 CHEMICALLY REACTING GAS MIXTURES

The working fluids in engines are mixtures of gases. Depending on the process under consideration and the portion of the engine cycle in which it occurs, chemical reactions may: (1) be so slow that they have a negligible effect on mixture composition (the mixture composition is essentially "frozen"); (2) be so rapid that as the mixture state changes, the composition remains in chemical equilibrium; (3) be one of the rate-controlling processes that determine how the composition of the mixture changes with time.

3.7.1 Chemical Equilibrium

It is a good approximation for performance estimates in engines to regard the burned gases produced by the combustion of fuel and air as in chemical equilibrium. [e] By this we mean that the chemical reactions by which individual species in the burned gases react together, produce and remove each species at equal rates. No net change in species composition results.

For example, if the temperature of a mass of carbon dioxide gas in a vessel is increased sufficiently, some of the CO_2 molecules *dissociate* into CO and O_2 molecules. If the mixture of CO_2, CO, and O_2 is in equilibrium, then CO_2 molecules are dissociating into CO and O_2 at the same rate as CO and O_2 molecules are *recombining* in the proportions required to satisfy the equation

$$CO + \frac{1}{2}O_2 = CO_2$$

In combustion products of hydrocarbon fuels, the major species present at low temperatures are N_2, H_2O, CO_2, and O_2 or CO and H_2. At higher temperatures (greater than about 2200 K), these major species dissociate and react to form additional species in significant amounts. For example, the adiabatic combustion of a stoichiometric mixture of a typical hydrocarbon fuel with air produces products with species mole fractions of: $N_2 \sim 0.8$; H_2O, $CO_2 \sim 0.1$; CO, OH, O_2, NO, $H_2 \sim 0.01$; H, O ~ 0.001; and other species in lesser amounts.

The second law of thermodynamics defines the criterion for chemical equilibrium as follows. Consider a system of chemically reacting substances undergoing a constant-pressure, constant-temperature process. In the absence of shear work (and electrical work, gravity, motion, capillarity), the first law gives

$$\delta Q = dH$$

The second law gives

$$\delta Q \leq T\, dS$$

Combining these gives

$$dH - T\,dS \leq 0$$

Since we are considering constant-temperature processes, this equation holds for finite changes:

$$\Delta H - T\,\Delta S = \Delta G \leq 0$$

Thus, reactions can only occur (at constant pressure and temperature) if G (= $H - TS$) for the products is less than G for the reactants. Hence at equilibrium

$$(\Delta G)_{p,T} = 0 \tag{3.33}$$

Consider a reactive mixture of ideal gases. The reactant species M_a, M_b, etc., and product species M_l, M_m, etc., are related by the general reaction whose stoichiometry is given by

$$\nu_a M_a + \nu_b M_b + \cdots = \nu_l M_l + \nu_m M_m + \cdots \tag{3.34a}$$

This can be written as

$$\sum_i \nu_i M_i = 0 \tag{3.34b}$$

where the ν_i are the stoichiometric coefficients and by convention are positive for the product species and negative for the reactant species.

Let an amount δn_a of M_a react with δn_b of M_b, etc., and produce δn_l of M_l, δn_m of M_m, etc. These amounts are in proportion:

$$\delta n_i = \nu_i\,\delta n \tag{3.35}$$

The change in Gibbs free energy of a mixture of ideal gases, at constant pressure and temperature, as the composition changes is given by

$$(\Delta G)_{p,T} = \sum \tilde{\mu}_i \delta n_i \tag{3.36}$$

where Δn_i is the change in number of moles of species i, and $\tilde{\mu}$ is the *chemical potential*. The chemical potential, an intensive property, is defined as

$$\tilde{\mu}_i = \left(\frac{\partial G}{\partial n_i}\right)_{p,T,n_j(j\neq i)} \tag{3.37}$$

It is equal in magnitude to the specific Gibbs free energy at a given temperature and pressure. For an ideal gas, it follows from Eqs. (B.13), (B.15), and (3.37) that

$$\tilde{\mu}_i = \tilde{\mu}_i^\circ(T) + \tilde{R}T\ln\frac{p_i}{p_0} \tag{3.38}$$

where $\tilde{\mu}_i^\circ$ equals $\tilde{g}_i^\circ$, the standard specific Gibbs free energy of formation. The standard state pressure p_0 is usually 1 atmosphere.

Substitution in Eq. (3.36) gives, at equilibrium,

$$\sum\left(\tilde{\mu}_i^\circ + \tilde{R}T\ln\frac{p_i}{p_0}\right)\delta n_i = 0$$

or

$$\sum\left(\tilde{\mu}_i^\circ + \tilde{R}T\ln\frac{p_i}{p_0}\right)\nu_i\delta n = 0$$

We can divide by Δn and rearrange, to obtain

$$\sum\ln\left(\frac{p_i}{p_0}\right)^{\nu_i} = -\frac{\left(\sum\tilde{\mu}_i^\circ\nu_i\right)}{RT} = -\frac{\Delta G^\circ}{RT} = \ln K_p \tag{3.39}$$

k_p is the equilibrium constant at constant pressure:

$$k_p = \prod_i\left(\frac{p_i}{p_0}\right)^{\nu_i} \tag{3.40}$$

It is obtained from the Gibbs free energy of the reaction that can be calculated from the Gibbs free energy of formation of each species in the reaction, as indicated in Eq. (3.39) above. It is a function of temperature only.

In many chemical reaction data bases (e.g., the JANAF tables [8]), to simplify the calculation of equilibrium constants, values of $\log_{10}(k_p)_i$, the equilibrium constants of formation of one mole of each species from their elements in their standard states, are tabulated against temperature. The equilibrium constant for a specific reaction is then obtained via the relation

$$\log_{10}(K_p)_{\text{reaction}} = \sum_i \nu_i \log_{10}(k_p)_i \tag{3.41}$$

With the JANAF table values of $(k_p)_i$, the pressures in Eqs. (3.40) and (3.41) must be in atmospheres.

The effect of pressure on the equilibrium composition can be deduced from Eq. (3.40) . Substitution of the mole fractions $\tilde{x}_i$ and mixture pressure p gives

$$\prod_i \left(\frac{p_i}{p_0}\right)^{\nu_i} = \prod_i \left(\tilde{x}_i \frac{p}{p_0}\right)^{\nu_i} = \left(\frac{p}{p_0}\right)^{\sum_i \nu_i} \prod_i \tilde{x}_i^{\nu_i} = K_p$$

If $\sum_i \nu_i = 0$ changes in pressure have no effect on the composition. If $\sum_i \nu_i > 0$ (dissociation reactions), then the mole fractions of the dissociation products decrease as pressure increases. If $\sum_i \nu_i > 0$ (recombination reactions), the converse is true.

An equilibrium constant, K_c, based on concentrations (usually expressed in gram moles per cubic centimeter) is also used:

$$K_c = \prod_i [M_i]^{\nu_i} \tag{3.42}$$

Equation (3.40) can be used to relate K_p and K_c:

$$K_p = K_c (\tilde{R}T)^{\sum_i \nu_i} \tag{3.43}$$

for $p_0 = 1$ atmosphere. For $\sum_i \nu_i = 0$, K_p, K_p and K_c are equal.

EXAMPLE 3.4

A stoichiometric mixture of CO and O_2 in a closed vessel, initially at 1 atm and 300 K, is exploded. Calculate the composition of the products of combustion at 2500 K and the gas pressure.

The combustion equation is

$$CO + \frac{1}{2}O_2 = CO_2$$

The JANAF tables give $\log_{10} K_p$ (equilibrium constants of formation from the elements in their standard state) at 2500 K of CO_2, CO, and O_2 as 8.280, 6.840, and 0, respectively. Thus, the equilibrium constant for the CO combustion reaction above is, from Eq. (3.41) ,

$$\log_{10} K_p = 8.280 - 6.840 = 1.440$$

which gives $K_p = 27.5$.

If the degree of dissociation in the products is α (i.e., a fraction α of the CO_2 formed by complete combustion is dissociated), the product composition is

$$CO_2(1-\alpha); CO, \alpha; O_2, \frac{\alpha}{2}$$

For this mixture, the number of moles of reactants n_R is $\frac{3}{2}$ the number of moles of products n_P is $(1 + \alpha/2)$.

The ideal gas law gives

$$p_R V = n_R \tilde{R} T_R,\ p_P V = n_P \tilde{R} T_P$$

Thus

$$\frac{p_p}{n_p} = \frac{1}{1.5} \times \frac{2500}{300} = 5.555 \text{ atm/mol}$$

The equilibrium relation [Eq. (3.40)] gives

$$\frac{1-\alpha}{\alpha(\alpha/2)^{1/2}} \left(\frac{n_p}{p_p} \right)^{1/2} = 27.5$$

which can be solved to give $\alpha = 0.074$.

The composition of the products in mole fractions is, therefore,

$$\tilde{x}_{CO_2} = \frac{1-\alpha}{n_p} = 0.893$$

$$\tilde{x}_{CO} = \frac{\alpha}{n_p} = 0.071$$

$$\tilde{x}_{O_2} = \frac{\alpha/2}{n_p} = 0.037$$

The pressure of the product mixture is

$$p = 5.555 n_p = 5.76 \text{ atm}$$

EXAMPLE 3.5

In fuel-rich combustion product mixtures, equilibrium between the species CO_2, H_2O, CO, and H_2 is often assumed to determine the burned gas composition. For $\phi = 1.2$, for C_8H_{18}-air combustion products, determine the mole fractions of the product species at 1700 K.

The reaction relating these species (often called the water-gas reaction) is

$$CO_2 + H_2 = CO + H_2O$$

From the JANAF tables, values of the $\log_{10} K_p$ of formation for these species at 1700 K are: H_2, 0; CO, 8.011; H_2O(g), 4.699. The equilibrium constant for the above reaction is, from Eq. (3.41) ,

$$\log_{10} K_p = 8.011 + 4.699 - 12.180 = 0.530$$

from which $K_p = 3.388$.

The combustion reaction for C_8H_{18}–air with $\phi = 1.2$ can be written

$$C_8H_{18} + \frac{12.5}{1.2}(O_2 + 3.773N_2) \rightarrow aCO_2 + bH_2O + cCO + dH_2 + 39.30N_2$$

A carbon balance gives:	$a + c = 8$
A hydrogen balance gives:	$2b + 2d = 18$
An oxygen balance gives:	$2a + b + c = 20.83$

The equilibrium relation gives (bc)/(ad) = 3.388 (since the equilibrated reaction has the same number of moles as there are reactants or products, the moles of each species can be substituted for the partial pressures).

These four equations can be solved to obtain

$$c^2 - 19.3c + 47.3 = 0$$

which gives $c = 2.89$, $a = 5.12$, $b = 7.72$, and $d = 1.29$. The total number of moles of products is

$$a + b + c + d + 39.3 = 56.3$$

and the mole fractions of the species in the burned gas mixture are

$$CO_2, 0.0908; \quad H_2O, 0.137; \quad CO, 0.051; \quad H_2, 0.023; \quad N_2, 0.698$$

Our development of the equilibrium relationship for one reaction has placed no restrictions on the occurrence of simultaneous equilibria. Consider a mixture of N reacting gases in equilibrium. If there are C chemical elements, conservation of elements will provide C equations that relate the concentrations of these N species. Any set of (N - C) chemical reactions, each in equilibrium, which includes each species at least once, will then provide the additional equations required to determine the concentration of each species in the mixture. Unfortunately, this complete set of equations is a coupled set of C linear and (N - C) nonlinear equations that is difficult to solve for cases where (N - C) > 2. For complex systems such as this

standardized computer methods for the calculation of complex chemical equilibrium compositions are now used, for example, Chemkin. [14] The approach taken is to minimize explicitly the Gibbs free energy of the reacting mixture (at constant temperature and pressure) subject to the constraints of element mass conservation.

A thermodynamic state of a gas mixture is commonly specified by a temperature T and pressure p. Other pairs of state variables: enthalpy and pressure (useful for constant-pressure combustion processes); temperature and volume; internal energy and volume (useful for constant-volume combustion processes); entropy and pressure, and entropy and volume (useful for isentropic compressions and expansions) can be used.

For each reactant composition and pair of thermodynamic state variables, such chemical-equilibrium computer programs calculate and print out the following:

1. *Thermodynamic mixture properties* (obtained from the equilibrium composition and the appropriate gas mixture rule; see App. B). p, T, r, h, s, M, $(\partial \ln V/\partial \ln p)_T$, $(\partial \ln V/\partial \ln T)_p$; c_p, γ, and a (sound speed)
2. *Equilibrium composition*. Mole fractions of each species (which are present in significant amounts), $\tilde{x}_i$

Figure 3.12 shows how the equilibrium composition of the products of combustion of isooctane-air mixtures at selected temperatures and 30 atm pressure varies with the fuel-air equivalence ratio. At low temperatures, the products are N_2, CO_2, H_2O and O_2 for lean mixtures and N_2, CO_2, H_2O, CO, and H_2 for rich mixtures. As temperature increases, the burned-gas mixture composition becomes much more complex with dissociation products such as OH, O, and H becoming significant. Figure 3.13 shows adiabatic flame temperatures for typical engine conditions as a function of the fuel-air equivalence ratio, obtained using the methodology reviewed in this section. The isooctane-air unburned mixture state was 700 K and 10 atm (values typical for the end of compression in a gasoline SI engine). Flame temperatures for adiabatic combustion at constant pressure (where p_R and H_R are specified) and at constant volume (where V_R and U_R are specified) are shown. Flame temperatures at constant volume are higher because the final

pressure is higher and less dissociation occurs. Maximum flame temperatures occur slightly rich of stoichiometric.

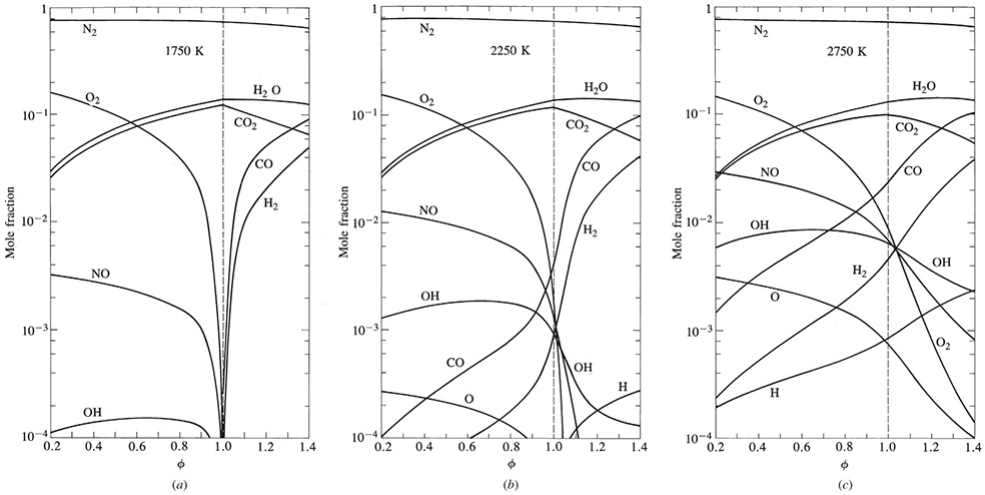


Figure 3.12 Mole fractions of equilibrium combustion products of isooctane-air mixtures as a function of fuel/air equivalence ratio at 30 atmospheres and (*a*) 1750 K; (*b*) 2250 K; (*c*) 2750 K.

3.7.2 Chemical Reaction Rates

Whether a system is in chemical equilibrium depends on whether the time constants of the controlling chemical reactions are short compared with time scales over which the system conditions (temperature and pressure) change. Chemical processes in engines are often not in equilibrium. Important examples of nonequilibrium phenomena are the flame reaction zone where the fuel is oxidized, and the air pollutant formation mechanisms. Such nonequilibrium processes are controlled by the rates at which the actual chemical reactions that convert reactants to products occur. The rates at which chemical reactions proceed depend on the concentration of the reactants, temperature, and whether any catalyst is present. This field is called chemical kinetics and some of its basic relations will now be reviewed. [2]

Most of the chemical reactions of interest in combustion are binary reactions, where two reactant molecules, M_a and M_b, with the capability of reacting together collide and form two product molecules, M_c and M_d, that is,

$$M_a + M_b = M_c + M_d \tag{3.44}$$

An important example of such a reaction is the rate-controlling step in the process by which the pollutant nitric oxide, NO, forms:

$$O + N_2 = NO + N$$

This is a second-order reaction since the stoichiometric coefficients of the reactants v_a and v_b are each unity and sum to 2. (The only relevant first-order reactions are decomposition processes.) Third-order reactions are important in combustion, also. Examples are the recombination reactions by which radical species such as H, O, and OH combine during the final stage of the fuel oxidation process, for example,

$$H + H + M = H_2 + M^* \tag{3.45}$$

M is any molecule (such as N_2) which takes part in the collision, which then carries away the excess energy (as M*).

The *law of mass action* states that the rate at which product species are produced, also the rate at which reactant species are removed, is proportional to the product of the concentrations of reacting species, with the concentration of each species raised to the power of its stoichiometric coefficient v_i. Thus, for reaction (3.44) above, the reaction rate R^+ in the forward (+) direction, reactants to products, is given by

$$R^+ = \frac{d[M_a]^+}{dt} = \frac{d[M_c]^+}{dt} = k^+[M_a][M_b] \tag{3.46}$$

If the reaction can also proceed in the reverse (–) direction, then the backward rate R^- is given by

$$R^- = -\frac{d[M_c]^-}{dt} = \frac{d[M_a]^-}{dt} = k^-[M_c][M_d] \tag{3.47}$$

k^+ and k^- are the rate constants in the forward and reverse directions for this reaction. The *net rate* of production of products or removal of reactants is, therefore,

$$R^+ - R^- = \frac{d[\mathrm{M}_c]^+}{dt} + \frac{d[\mathrm{M}_c]^-}{dt} = -\frac{d[\mathrm{M}_a]^+}{dt} + \frac{d[\mathrm{M}_a]^-}{dt}$$
$$= k^+[\mathrm{M}_a][\mathrm{M}_b] - k^-[\mathrm{M}_c][\mathrm{M}_d] \tag{3.48}$$

These results can be stated more generally as follows. Any reaction can be written as

$$\sum_{i=1}^{n} \nu_{R_i} \mathrm{M}_{R_i} = \sum_{i=1}^{m} \nu_{P_i} \mathrm{M}_{P_i} \tag{3.49}$$

where ν_i is the stoichiometric coefficient of species M_i, subscripts R and P denote reactants and products, respectively, and there are n reactant species and m product species. The forward reaction rate R^+ and the reverse reaction rate R^- are given by

$$R^+ = k^+ \prod_{i=1}^{n} [\mathrm{M}_{R_i}]^{\nu_{R_i}}$$
$$R^- = k^- \prod_{i=1}^{m} [\mathrm{M}_{P_i}]^{\nu_{P_i}} \tag{3.50}$$

The net rate of removal of reactant species M_{R_i} is

$$-\frac{d[\mathrm{M}_{R_i}]}{dt} = \nu_{R_i}(R^+ - R^-) \tag{3.51a}$$

and the net rate of production of product species M_{P_i} is

$$\frac{d[\mathrm{M}_{P_i}]}{dt} = \nu_{P_i}(R^+ - R^-) \tag{3.51b}$$

The rate constants, k, usually follow the Arrhenius form:

$$k = A \exp\left(-\frac{E_A}{RT}\right) \tag{3.52}$$

where A is called the frequency or pre-exponential factor and may be a (moderate) function of temperature; E_A is the activation energy. The Boltzmann factor exp $(-E_A/RT)$ defines the fraction of all collisions that have an energy greater than E_A, that is, sufficient energy to break the molecular bonds and cause the reaction to take place. The functional dependence of k on T and the constants in the Arrhenius form, Eq. (3.52) (if that equation is appropriate), are determined experimentally.

At equilibrium, the forward and reverse reaction rates are equal. Then, from Eq. (3.48) , with $R^+ - R^- = 0$:

$$\frac{k^+}{k^-} = \frac{[\mathrm{M}_c][\mathrm{M}_d]}{[\mathrm{M}_a][\mathrm{M}_b]} = K_c$$

where K_c is the equilibrium constant based on concentrations defined by Eq. (3.42) . It can be related to K_p, the equilibrium constant based on partial pressures, by Eq. (3.43) .

The chemical reaction mechanisms of importance in combustion are much more complex than the above illustrations of rate-controlled processes. Such mechanisms usually involve both parallel and sequential interdependent reactions. The methodology reviewed above still holds; however, one must sum algebraically the forward and reverse rates of all the reactions that produce (or remove) a species of interest. In such complex mechanisms, it is often useful to assume that (some of) the reactive intermediate species or radicals are in *steady state*. That is, these radicals react so quickly once they are formed that their concentrations are maintained in steady state—effectively, in equilibrium—with the species with which they react. The net rate at which their concentration changes with time is set equal to zero.

PROBLEMS

3.1 Gasoline is supplied to a four-cylinder spark-ignition engine at 2 g/s. Calculate the air-flow rate for stoichiometric combustion. If the engine is operating at 1500 rev/min, estimate the mass of fuel and air entering each

cylinder per cycle. The engine displaced volume is 2.4 liters. What is the volumetric efficiency?

3.2 Calculate the exhaust gas composition of a butane-fueled spark-ignition engine operating with a fuel/air equivalence ratio of 0.9. Assume the fuel is fully burned within the cylinder. Butane is C_4H_{10}.

3.3 The molar composition of dry exhaust gas of a propane-fueled SI engine is given below (water was removed before the measurement). Calculate the relative air/fuel ratio.

$$CO_2 = 10.8\%, \quad O_2 = 4.5\%, \quad CO = 0\%, \quad H_2 = 0\%$$

3.4 Evaluate and compare the lower heating values per unit mass of stoichiometric mixture and per unit volume of stoichiometric mixture (at standard atmospheric conditions) for methane, isooctane, gasoline, ethanol, methanol, and hydrogen. Assume the fuel is fully vaporized. Heating values for these fuels are given in App. D.

3.5 The measured engine fuel flow rate is 0.4 g/s, the air flow rate is 5.6 g/s, and the exhaust gas composition (measured dry) is CO_2 = 13.0%, CO = 2.8% with O_2 essentially zero. Unburned hydrocarbon emissions can be neglected. Compare the equivalence ratio calculated from the fuel and air flow with the equivalence ratio calculated from the exhaust gas composition. The fuel is gasoline with a H/C ratio of 1.87. Assume a H_2 concentration equal to one-third the CO concentration.

3.6 This problem assesses several fuels for use in an internal combustion engine. The fuels of interest are gasoline, diesel fuel, methane, ethanol, and hydrogen. Properties of these fuels can be found in Table D.4 of App. D .

(a) For each of these fuels, determine the lower heating value per unit volume of stoichiometric mixture at standard temperature and pressure. Normalize your results by the gasoline value.

(b) Also calculate the lower heating value per unit volume of each fuel at standard temperature and pressure. Again normalize your results by the gasoline value.

(c) What do these results tell you about how the engine's output (maximum torque and power) change (for a fixed engine size) when using each of these fuels? How does the amount of energy able to be stored onboard (in a nonpressurized tank of fixed size) change when using each of these fuels?

3.7 Here, we examine the effects of using reformulated gasoline in a spark-ignition engine. Reformulated gasoline (to lower the engine's emissions) contains 2% by weight of oxygen. One approach is to blend ethanol (C_2H_6O) with the gasoline (approximate as C_7H_{13}) in the appropriate ratio to provide this 2% oxygen. The heating value of gasoline is 44 MJ/kg; the heating value of ethanol is 26.9 MJ/kg. The density of the two fuels is essentially the same, 0.76 Mg/m^3.

(a) Calculate the required mass fraction of ethanol in the ethanol-gasoline mixture.

(b) The stoichiometric air/fuel ratios for ethanol and gasoline are 9.0 and 14.6, respectively. Determine the stoichiometric A/F ratio for the mixture.

(c) The fuel system injects a given volume of fuel based on the measured mass flow rate of air. So in a spark-ignition engine designed for stoichiometric operation with gasoline, the engine will run lean with the ethanol-gasoline mixture. At what lean relative air/fuel ratio will it operate?

3.8 An internal combustion engine is being operated with a pure alcohol as fuel. The fuel molecule has 7 carbon atoms. A detailed analysis of the exhaust gas constituents on a wet molar basis (i.e., with the H_2O present) in the exhaust manifold yielded the following:

$$CO_2: 13\%, CO: 5.2\%, H_2O: 9.1\%, H_2: 1.3\%, \text{and } N_2: 71.3\%$$

(a) Determine the elemental composition of the fuel molecule.

(b) Determine the relative air-fuel ratio. Is this lean, stoichiometric, or rich?

(c) The engine is a four-valve per cylinder naturally-aspirated spark-ignition engine. It has a displaced volume of 3 liters, the air flow is 0.06 kg/s, and the engine speed is 3000 rev/min. Estimate the engine load (i.e., the approximate ratio of the actual brake torque to the maximum brake torque) at these conditions. The friction mean effective pressure is 200 kPa at this speed and

is essentially independent of load at the loads considered here. The gross indicated fuel conversion efficiency is also independent of load.

Hints: You may need to make appropriate assumptions concerning the engine's breathing characteristics and normalized torque, according to the given engine technology.

3.9 An upper estimate can be made of the amount of NO formed in an engine from considering the equilibrium of the reaction $N_2 + O_2 = 2NO$. Calculate the NO concentration at equilibrium at 2500 K and 30 atm. $\text{Log}_{10} K_p = -1.2$ for this reaction at 2500 K. Assume N/O ratio in the combustion products is 15. N_2, O_2, and NO are the only reacting species present.

3.10 Carbon monoxide reacts with air at 1 atm and 1000 K in an exhaust gas reactor. The mole fractions of the exhaust gas-air mixture flowing into the reactor are CO, 3%; O_2, 7%; N_2, 74%; CO_2, 6%; H_2O, 10%.

(a) Calculate the concentration of CO and O_2 in gram moles per cm^3 in the entering mixture.

(b) The rate of reaction is given by

$$d[\text{CO}]/dt = -4.3\times10^{11}\times[\text{CO}][\text{O}_2]^{0.25}\exp[-E/(RT)]$$

[ ] denotes concentration in gram moles per cm^3, $E/R = 20{,}000$ K. Calculate the initial reaction rate of CO, $d[\text{CO}]/dt$: time is in seconds.

(c) The equilibrium constant K_p for the reaction

$\text{CO} + \frac{1}{2}\text{O}_2 = \text{CO}_2$ at 1000 K is 10^{10}. at 1000 K is 10^{10}. Find the equilibrium CO concentration.

(d) Determine the time to reach this equilibrium concentration of CO using the initial reaction rate. (The actual time will be longer but this calculation indicates approximately the time required.)

3.11 The exhaust gases of a hydrogen-fueled engine contain 22.3% H_2O, 7.44% O_2, and 70.2% N_2. At what fuel/air equivalence ratio and relative air/fuel ratio is it operating?

3.12 Gas is sampled at 1 bar pressure from the exhaust manifold of an internal combustion engine and analyzed. The mole fractions of species in the exhaust are:

$$H_2O\ 0.0468;\quad CO_2\ 0.0585;\quad O_2\ 0.123;\quad N_2\ 0.772$$

Other species such as CO and unburned hydrocarbons can be neglected.

(a) The fuel is a synthetic fuel derived from coal and biomass containing only carbon and hydrogen. What is the ratio of hydrogen atoms to carbon atoms in the fuel?

(b) Calculate the fuel/air equivalence ratio at which this engine is operating.

(c) Is the internal combustion engine a conventional spark-ignition or a diesel engine? Explain your logic.

(d) The engine has a displaced volume of 2 liters. Estimate approximately the percentage by which the fuel flow rate would be increased if this engine were operated at its maximum load at this same speed (2000 rev/min). Explain briefly what limits the equivalence ratio at maximum load.

3.13 Liquid petroleum gas (LPG) is used to fuel spark-ignition engines. A typical sample of the fuel consists of

70% by volume propane C_3H_8

5% by volume butane C_4H_{10}

25% by volume propene C_3H_6

The *higher heating values* of the fuels are: propane, 50.38 MJ/kg; butane, 49.56 MJ/kg; propylene (propene), 48.95 MJ/kg.

(a) Work out the overall combustion reaction for stoichiometric combustion of 1 mole of LPG with air, and the stoichiometric *F/A* and *A/F*.

(b) What are the higher and lower heating values for combustion of this fuel with excess air, per unit mass of LPG?

3.14 A spark-ignition engine is operated on isooctane fuel (C_8H_{18}). The exhaust gases are cooled, dried to remove water, and then analyzed for CO_2, CO, H_2, O_2. Using the overall combustion reaction for a range of

equivalence ratios from 0.5 to 1.5, calculate the mole fractions of CO_2, CO, H_2, and O_2 in the dry exhaust gas, and plot the results as a function of equivalence ratio. Assume:

(a) that all the fuel is burnt inside the engine (almost true) and that the ratio of moles CO to moles H_2 in the exhaust is 3:1, and

(b) that there is no hydrogen in the exhaust for lean mixtures.

For high power engine operation, the air/fuel ratio is 14:1. What is the exhaust gas composition, in mole fractions, *before* the water is removed?

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[a] Blends of alcohols with gasoline are now being used, for example, an E10 blend combines 10% ethanol (by volume) with 90% gasoline.

[b] Note that for fuels, which are mixtures of hydrocarbons, *a* and *b* need not be integers.

[c] The approach used here follows that developed by Spalding and Cole.[7]

[d] Standard methods for measuring heating values are defined by the American Society for Testing Materials.

[e] This assumption is not valid late in the expansion stroke and during the exhaust process (see Sec. 4.9). Nor does it take account of pollutant formation processes (see Chap. 11).

CHAPTER 4

Properties of Working Fluids

4.1 INTRODUCTION

The study of engine operation through an analysis of the processes that occur inside the engine has a long and productive history. The earliest attempts at this analysis used the constant-volume and constant-pressure ideal cycles as approximations to real engine processes (see Chap. 5). With the development of high-speed digital computers, the simulation of engine processes has become much more sophisticated and accurate (see Chap. 14). All these engine simulations (from the simplest to the most complex) require models for the composition and properties of the working fluids inside the engine, as well as models for the individual processes—induction, compression, combustion, expansion, and exhaust—that make up the engine operating cycle. This chapter deals with models for the working fluid composition, and thermodynamic and transport properties.

The composition of the working fluid, which changes during the engine operating cycle, is summarized in Table 4.1. The unburned mixture for a spark-ignition engine during intake and compression consists of air, fuel, and previously burned gases. It is, therefore, a mixture of N_2, O_2, CO_2, H_2O, with CO, and H_2 for fuel-rich mixtures, and fuel (usually vapor). The composition of the unburned mixture does not change significantly during intake and compression. It is sufficiently accurate to assume the composition is *frozen*. For the compression-ignition engine, the unburned mixture prior to injection contains no fuel; it consists of air and previously burned gas.

TABLE 4.1 Working fluid constituents

Process	Spark-ignition engine	Compression-ignition engine
Intake	Air	Air
	Fuel*	Recycled exhaust†
	Recycled exhaust†	Residual gas‡
	Residual gas‡	
Compression	Air	Air
	Fuel vapor	Recycled exhaust
	Recycled exhaust	Residual gas
	Residual gas	
Expansion	Combustion products (mixture of N_2, H_2O, CO_2, CO, H_2, O_2, NO, OH, O, H, …)	Combustion products (mixture of N_2, H_2O, CO_2, CO, H_2, O_2, NO, OH, O, H, …)
Exhaust	Combustion products [mainly N_2, CO_2, H_2O, and either O_2 ($\phi < 1$) or CO and H_2 ($\phi > 1$)]	Combustion products (mainly N_2, CO_2, H_2O, and O_2)

*With port fuel injection, mainly vapor within the cylinder: with direct injection, liquid and vapor.
†Often used to control NO_x emissions (see Secs. 11.2, 15.3.2, and 15.5.1).
‡Within the cylinder.

The combustion products or burned mixture gases, during the combustion process and much of the expansion process, are close to *thermodynamic equilibrium*. The composition of such mixtures has already been discussed (Sec. 3.7.1). As these combustion products cool, recombination occurs as indicated in Fig. 3.12. Toward the end of the expansion process, the gas composition departs from the equilibrium composition; recombination can no longer occur fast enough to maintain the reacting mixture in equilibrium. During the exhaust process, reactions in the burned gases are sufficiently slow so that for calculating thermodynamic properties the composition can be regarded as *frozen*.

The models used for predicting the thermodynamic properties of unburned and burned mixtures can be grouped into the five categories listed in Table 4.2. The first category is only useful for illustrative purposes since the specific heats of unburned and burned mixtures are significantly different. While the specific heats of the working fluids do increase with increasing temperature in the range of interest, a constant–specific–heat model can be matched to the thermodynamic data over a limited temperature range. This approach provides a simple *analytic* model that can be useful when moderate accuracy of prediction will suffice. The appropriateness of frozen and equilibrium assumptions has already been discussed above.

Approximations to thermodynamic equilibrium calculations have been useful in the past because of the savings in computational time, relative to full equilibrium calculations, which can result.

TABLE 4.2 Categories of models for thermodynamic properties

Unburned mixture	Burned mixture
1. Single ideal gas throughout operating cycle with c_v (and hence c_p) constant	
2. Ideal gas; $c_{v,u}$ constant	Ideal gas; $c_{v,b}$ constant
3. Frozen mixture of ideal gases; $c_{v,i}(T)$	Frozen mixture of ideal gases; $c_{v,i}(T)$
4. Frozen mixture of ideal gases; $c_{v,i}(T)$	Approximations fitted to equilibrium thermodynamic properties
5. Frozen mixture of ideal gases; $c_{v,i}(T)$	Mixture of reacting ideal gases in thermodynamic equilibrium

Note: Subscripts i, u, and b denote species i in the gas mixture, the unburned mixture, and burned mixture properties, respectively.

Values of thermodynamic properties of unburned and burned mixtures relevant to engine calculations are available from charts, tables, algebraic relationships developed to match tabulated data, and computer programs. A selection of this material is included in this chapter and App. D . The references indicate additional sources.

4.2 UNBURNED MIXTURE COMPOSITION

The mass of charge trapped in the cylinder (m_c) is the inducted mass per cycle (m_i), plus the residual mass (m_r) left over from the previous cycle. The residual fraction (x_r) is

$$x_r = \frac{m_r}{m_c} \tag{4.1}$$

Typical residual fractions in spark-ignition engines range from some 30% at idle to about 5% at full load. In diesels, the residual fraction is smaller (a few percent) due to the higher compression ratio, and in naturally-aspirated engines is approximately constant since the intake is unthrottled. If the

inducted mixture is fuel and air (or air only), then the *burned gas fraction* (x_b) in the unburned mixture during compression equals the residual fraction.

In some engines, a fraction of the engine exhaust gases is recycled to the intake to dilute the fresh mixture for control of NO $_x$ emissions (see Sec. 11.2). The percent of exhaust gas recycled (%EGR) is usually defined as the percent of the total intake mixture which is recycled exhaust, [a]

$$\mathrm{EGR}(\%) = \left(\frac{m_{\mathrm{EGR}}}{m_i}\right) \times 100 \tag{4.2}$$

where m_{EGR} is the mass of exhaust gas recycled, then the burned gas fraction in the fresh mixture is

$$x_b = \frac{m_{\mathrm{EGR}} + m_r}{m_c} = \left(\frac{\mathrm{EGR}}{100}\right)(1 - x_r) + x_r \tag{4.3}$$

Depending on operating conditions, in SI engines up to about 30% of the exhaust can be recycled; in diesel engines, much higher levels may be used.

The composition of the burned gas fraction in the unburned mixture can be calculated as follows. The combustion equation for a hydrocarbon fuel of average molar H/C ratio y [e.g., Eq. (3.5)] can be written per mole O_2 as

$$\begin{aligned} \varepsilon\phi\mathrm{C} + 2(1-\varepsilon)\phi\mathrm{H_2} + \mathrm{O_2} + \psi\mathrm{N_2} \rightarrow\ & n_{\mathrm{CO_2}}\mathrm{CO_2} + n_{\mathrm{H_2O}}\mathrm{H_2O} \\ & + n_{\mathrm{CO}}\mathrm{CO} + n_{\mathrm{H_2}}\mathrm{H_2} + n_{\mathrm{O_2}}\mathrm{O_2} + n_{\mathrm{N_2}}\mathrm{N_2} \end{aligned} \tag{4.4}$$

where

ψ = the molar N/O ratio (3.773 for air)

$$\varepsilon = \frac{4}{4 + y}$$

y = the molar H/C ratio of the fuel

ϕ = fuel/air equivalence ratio

n_i = moles of species i per mole O_2 reactant

The n_i are determined using the following assumptions:

1. For lean and stoichiometric mixtures ($\varphi \leq 1$) CO and H_2 can be neglected.
2. For rich and stoichiometric mixtures ($\varphi \geq 1$) O_2 can be neglected.
3. For rich mixtures, either (*a*) the water–gas reaction

$$CO_2 + H_2 = CO + H_2O$$

can be assumed to be in equilibrium with the equilibrium constant $K(T)$:

$$K(T) = \frac{n_{H_2O} n_{CO}}{n_{CO_2} n_{H_2}}$$

where $K(T)$ can be determined from a curve fit to JANAF Tables data: [1]

$$\ln K(T) = 2.743 - \frac{1.761 \times 10^3}{T} - \frac{1.611 \times 10^6}{T^2} + \frac{0.2803 \times 10^9}{T^3} \qquad (4.5)$$

where T is in K, or (*b*) K can be assumed constant over the normal engine operating range. A value of 3.5 is often assumed (see Sec. 4.9), which corresponds to evaluating the equilibrium constant at 1740 K.

The n_i obtained from an element balance and the above assumptions are shown in Table 4.3. The value of c is obtained by solving the quadratic:

TABLE 4.3 Burned gas composition under 1700 K

	n_i, moles/mole O_2 reactant	
Species	$\phi \leq 1$	$\phi > 1^*$
CO_2	$\varepsilon\phi$	$\varepsilon\phi - c$
H_2O	$2(1-\varepsilon)\phi$	$2(1-\varepsilon\phi) + c$
CO	0	c
H_2	0	$2(\phi - 1) - c$
O_2	$1 - \phi$	0
N_2	ψ	ψ
Sum: n_b	$(1-\varepsilon)\phi + 1 + \psi$	$(2-\varepsilon)\phi + \psi$

*c defined by Eq. (4.6).

$$(K-1)c^2 - c\{K[2(\phi-1)+\varepsilon\phi]+2(1-\varepsilon\phi)\}+2K\varepsilon\phi(\phi-1)=0 \tag{4.6}$$

The mole fractions are given by

$$\tilde{x}_i = \frac{n_i}{n_b}$$

where $n_b = \sum_i n_i$ is given in the bottom line of Table 4.3.

While Eq. (4.4) is for a fuel containing C and H only, it can readily be modified for alcohols or alcohol–hydrocarbon blends. For a fuel of molar composition CH_yO_z, the reactant mixture

$$CH_yO_z + \frac{1}{\phi}\left(1 + \frac{y}{4} - \frac{z}{2}\right)(O_2 + \psi N_2)$$

can be rearranged per mole of O_2 reactant as

$$\zeta\phi\varepsilon C + 2\zeta\phi(1-\varepsilon)H_2 + O_2 + \left(1 - \frac{\varepsilon z}{2}\right)\zeta\psi N_2 \tag{4.7a}$$

where

$$\zeta = \frac{2}{2 - \varepsilon z(1-\phi)} \tag{4.7b}$$

If we write

$$\phi^* = \zeta\phi \quad \text{and} \quad \psi^* = \left(1 - \frac{\varepsilon z}{2}\right)\zeta\psi \tag{4.7c}$$

the reactant expression (4.7 *a*) becomes

$$\varepsilon\phi^* C + 2(1-\varepsilon)\phi^* H_2 + O_2 + \psi^* N_2$$

which is identical in form to the reactant expression for a hydrocarbon fuel, Eq. (4.4) . Thus, Table 4.3 can still be used to give the composition of the burned gas residual fraction in the unburned mixture, except that ϕ^* replaces ϕ and ψ^* replaces ψ in the expressions for n_i.

Now consider the unburned mixture. The number of moles of fuel per mole O_2 in the mixture depends on the molecular weight of the fuel, M_f. If the average molecular formula of the fuel is $(CH_y)_\alpha$ then

$$M_f = \alpha(12 + y)$$

The fresh fuel-air mixture (not yet diluted with EGR or residual),

$$\varepsilon\phi C + 2(1-\varepsilon)\phi H_2 + O_2 + \psi N_2$$

then becomes

$$\frac{4}{M_f}(1+2\varepsilon)\phi(CH_y)_\alpha + O_2 + \psi N_2$$

The unburned mixture (fuel, air, and a burned gas fraction), per mole O_2 in the mixture, can be written:

$$(1-x_b)\left[\frac{4}{M_f}(1+2\varepsilon)\phi(CH_y)_\alpha + O_2 + \psi N_2\right] + x_b\left(n_{CO_2} + n_{H_2O} + n_{CO} + n_{H_2} + n_{O_2} + n_{N_2}\right)$$

The number of moles of each species in the unburned mixture, per mole O_2, is summarized in Table 4.4. The mole fractions of each species are then obtained by dividing by the total number of moles of unburned mixture n_u,

TABLE 4.4 Unburned mixture composition

	n_i, moles/mole O_2 reactant	
Species	$\phi \leq 1$	$\phi > 1$
Fuel	$4(1 - x_b)(1 + 2\varepsilon)\phi/M_f$	
O_2	$1 - x_b\,\phi$	$1 - x_b$
N_2	ψ	ψ
CO_2	$x_b\varepsilon\phi$	$x_b(\varepsilon\phi - c)$
H_2O	$2x_b(1 - \varepsilon)\phi$	$x_b[2(1 - \varepsilon\phi) + c]$
CO	0	$x_b c$
H_2	0	$x_b[2(\phi - 1) - c]$
Sum*	n_u	n_u

*Given by Eq. (4.8).

$$n_u = (1 - x_b)\left[\frac{4(1 + 2\varepsilon)\phi}{M_f} + 1 + \psi\right] + x_b n_b \tag{4.8}$$

where n_b is given in Table 4.3.

The molecular weights of the (low-temperature) burned and unburned mixture can now be determined. The mass of mixture (burned or unburned) per mole O_2 in the mixture, m_{RP}, is given by

$$m_{RP} = 32 + 4\phi(1 + 2\varepsilon) + 28.16\psi \tag{4.9}$$

The molecular weight of the burned mixture, M_b, is therefore

$$M_b = \frac{m_{RP}}{n_b} \tag{4.10}$$

The molecular weight of the unburned mixture, M_u, is

$$M_u = \frac{m_{RP}}{n_u} \tag{4.11}$$

Figure 4.1 gives M_u and M_b for a range of ϕ and x_b for air, isooctane, burned gas mixtures.

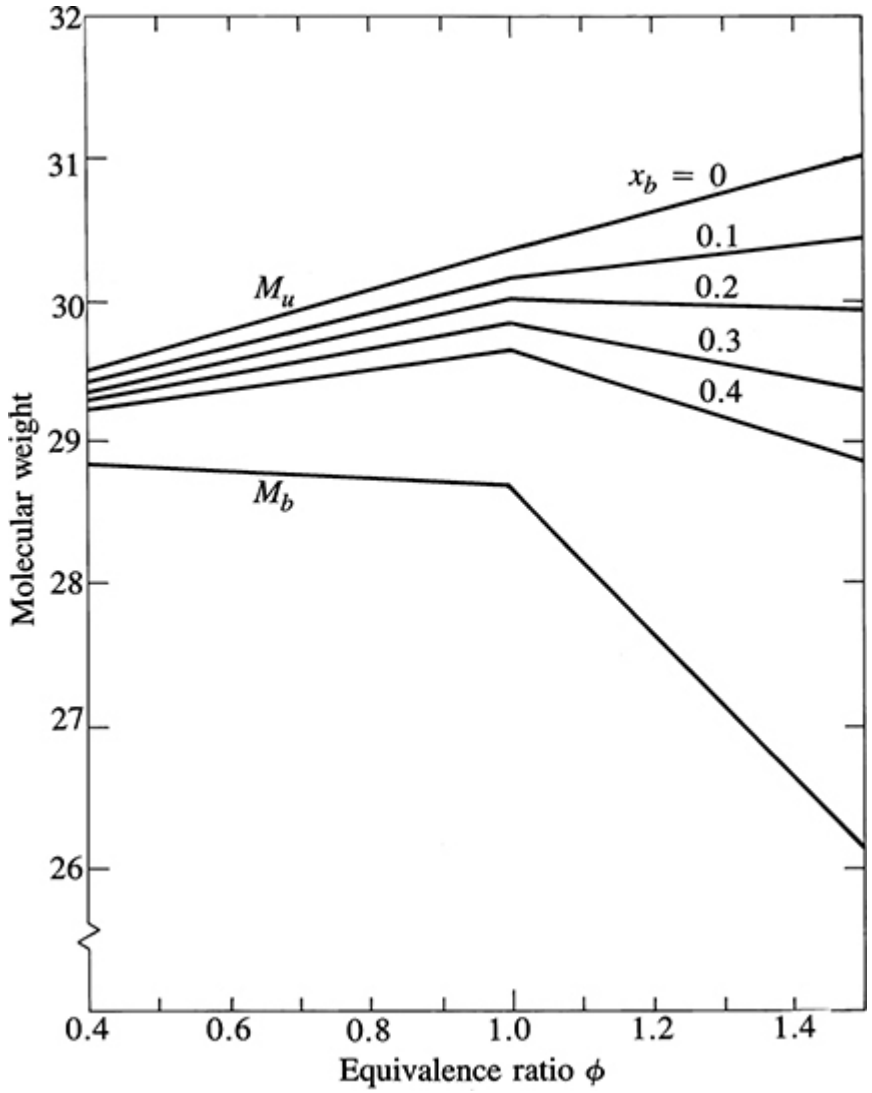


Figure 4.1 Molecular weight of unburned and low-temperature burned isooctane-air mixtures as a function of fuel/air equivalence ratio and burned gas fraction.

Frequently, thermodynamic properties of unburned and burned mixtures are expressed per unit mass of *air* in the *original* mixture (for burned mixture this is the mixture before combustion). To obtain properties in these units, we need the mass of original air, per mole O_2 in the mixture, which is

$$(32 + 28.16\psi)$$

with units of kilograms per kilomole or pound-mass per pound-mole.

Table 4.5 summarizes the factors needed to relate properties expressed on a molar and a mass basis.

TABLE 4.5 Factors for relating properties on molar and mass basis

Quantity, per mole O_2 in the mixture	General equation*		Equation for C_8H_{18}-air mixtures
Moles of burned mixture n_b	$n_b = (1-\varepsilon)\phi + 1 + \psi$,	$\phi \le 1$	$n_b = 0.36\phi + 4.773$
	$n_b = (2-\varepsilon)\phi + \psi$,	$\phi > 1$	$n_b = 1.36\phi + 3.773$
Moles of unburned mixture n_u	$(1-x_b)\left[\frac{4(1+2\varepsilon)\phi}{M_f} + 1 + \psi\right] + x_b n_b$	$\phi \le 1$	$n_u = 0.08\phi + 4.773 + 0.28x_b\phi$
		$\phi > 1$	$n_u = 0.08\phi + 4.773 + x_b(1.28\phi - 1)$
Mass of mixture† (burned or unburned)	$m_{RP} = 32 + 4\phi(1+2\varepsilon) + 28.16\psi$		$138.2 + 9.12\phi$
Mass of air†	$32 + 28.16\,\psi$		138.2

*For hydrocarbon fuels, ψ for air = 3.773; for fuels containing oxygen, ϕ' and ψ' given by Eq. (4.7c) are substituted for ϕ and ψ, respectively.
†Units: kg/kmol or lb_m/lb-mol.

4.3 GAS PROPERTY RELATIONSHIPS

The individual species in the unburned and burned gas mixtures can with sufficient accuracy be modeled as ideal gases. Ideal gas relationships are reviewed in App. B. The most important relationships for property determination for engine calculations are summarized below.

Since internal energy and enthalpy are functions of temperature only, the specific heats at constant volume and constant pressure are given by

$$c_v = \left(\frac{\partial u}{\partial T}\right)_v = \frac{du}{dT} = c_v(T) \tag{4.12a}$$

$$c_p = \left(\frac{\partial h}{\partial T}\right)_p = \frac{dh}{dT} = c_p(T) \tag{4.12b}$$

and

$$u - u_0 = \int_{T_0}^{T} c_v dT \tag{4.13a}$$

$$h - h_0 = \int_{T_0}^{T} c_p dT \tag{4.13b}$$

The entropy, *s(T, v)* or *s(T, p)*, is given by

$$s - s_0 = \int_{T_0}^{T} c_v \frac{dT}{T} + R \ln \frac{v}{v_0} \tag{4.14a}$$

$$s - s_0 = \int_{T_0}^{T} c_p \frac{dT}{T} + R \ln \frac{p}{p_0} \tag{4.14b}$$

The integrals in Eqs. (4.14 *a, b*) are functions of temperature only, and are useful in evaluating entropy changes and in following isentropic processes. If we define

$$\Psi(T) = \int_{T_0}^{T} c_v(T) \frac{dT}{T} \tag{4.15a}$$

and

$$\Phi(T) = \int_{T_0}^{T} c_p(T) \frac{dT}{T} \tag{4.15b}$$

then

$$s - s_0 = \Psi + R \ln \left(\frac{v}{v_0} \right) \tag{4.16a}$$

$$s - s_0 = \Phi + R \ln \left(\frac{p}{p_0} \right) \tag{4.16b}$$

Thus, for example, the entropy change between states (T_1, p_1) and (T_2, p_2) is

$$s_2 - s_1 = \Phi_2 - \Phi_1 - R\ln\left(\frac{p_2}{p_1}\right) \tag{4.17}$$

For an isentropic process,

$$\ln\left(\frac{p_2}{p_1}\right) = \frac{\Phi_2 - \Phi_1}{R} \tag{4.18}$$

In these equations, the units of u and h can be on a per unit mass or molar basis (i.e., joules per kilogram or joules per kilomole); similarly, s, c_v , c_p , R, ψ, and Φ can be in joules per kilogram-kelvin.

For gas mixtures, once the composition is known, mixture properties are determined either on a mass or molar basis from

$$u = \sum x_i u_i \qquad \tilde{u} = \sum \tilde{x}_i \tilde{u}_i \tag{4.19a}$$

$$h = \sum x_i h_i \qquad \tilde{h} = \sum \tilde{x}_i \tilde{h}_i \tag{4.19b}$$

$$s = \sum x_i s_i \qquad \tilde{s} = \sum \tilde{x}_i \tilde{s}_i \tag{4.19c}$$

and

$$c_v = \sum x_i c_{v,i} \qquad \tilde{c}_v = \sum \tilde{x}_i \tilde{c}_{v,i} \tag{4.20a}$$

$$c_p = \sum x_i c_{p,i} \qquad \tilde{c}_p = \sum \tilde{x}_i \tilde{c}_{p,i} \tag{4.20b}$$

4.4 A SIMPLE ANALYTIC IDEAL GAS MODEL

While the first category of model listed in Table 4.2 is too inaccurate for other than illustrative purposes, the second category—constant but different specific heats for the unburned and burned gas mixtures—can with careful

choice of specific heat values be made much more precise. The advantages of a simple analytic model may be important for certain problems.

Figure 4.2 shows an internal energy versus temperature plot for a stoichiometric mixture. It is a quantitative version of Fig. 3.5. The unburned mixture line is for a burned gas fraction of 0.1. The fuel is isooctane. Data to construct such graphs can be obtained from charts or tables or computer programs (see Secs. 4.5 to 4.7). The units for u are kilojoules per kilogram of air in the original mixture (the units of the charts in Sec. 4.5). The datum is zero enthalpy for O_2, N_2, H_2, and C (solid) at 298 K. Note that the specific heats of the unburned and burned mixtures (the slopes of the lines in Fig. 4.2) are a function of temperature; at high temperatures, the internal energy of the burned mixture is a function of temperature and pressure.

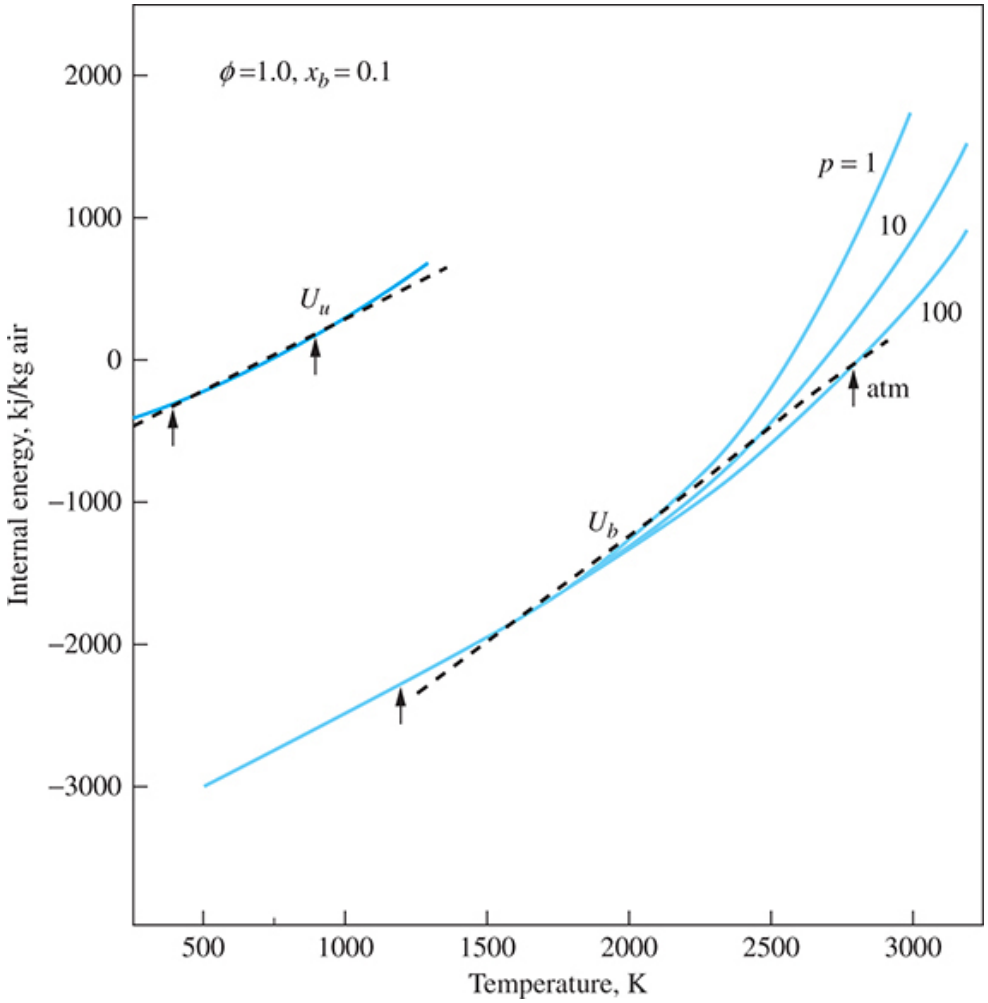


Figure 4.2 Internal energy versus temperature plot for stoichiometric unburned and burned gas mixtures: isooctane fuel; unburned residual fraction

0.1.

However, the temperature range of interest for the unburned mixture in an SI engine is 400 to 1000 K; for the burned gas mixture, the extreme end states are approximately 2800 K, 35 atm and 1200 K, 2 atm. Linear approximations to the unburned and burned mixture curves, which minimize the error in u over the temperature (and pressure) ranges of interest, are shown as dashed lines. The error in T for a given u is less than 50 K.

The basis for this ideal gas model is

$$u_u = c_{v,u} T_u + h_{f,u} \qquad h_u = c_{p,u} T_u + h_{f,u} \tag{4.21a, b}$$

$$u_b = c_{v,b} T_b + h_{f,b} \qquad h_b = c_{p,b} T_b + h_{f,b} \tag{4.22a, b}$$

where $h_{f,u}$ and $h_{f,b}$ are the enthalpies of formation of unburned and burned gas mixture, respectively, at 0 K.

Then, for a constant-volume adiabatic combustion process,

$$u_u = u_b$$

or

$$c_{v,u} T_u + h_{f,u} = c_{v,b} T_b + h_{f,b}$$

If we solve for T_b and use the relations $(R_b/R_u) = (M_u/M_b)$ and $c_v/R = 1/(\gamma - 1)$, we obtain

$$T_b = (\gamma_b - 1)\left(\frac{M_b}{M_u}\right)\left(\frac{T_u}{\gamma_u - 1} + \frac{\Delta h_f}{R_u}\right) \tag{4.23}$$

where $\Delta h_f = h_{f,u} - h_{f,b}$.

For a constant-pressure adiabatic combustion process,

$$h_u = h_b$$

and it can similarly be shown that

$$T_b = \frac{\gamma_b - 1}{\gamma_b}\left(\frac{M_b}{M_u}\right)\left(\frac{\gamma_u}{\gamma_u - 1}T_u + \frac{\Delta h_f}{R_u}\right) \tag{4.24}$$

To use the model, suitable values of γ_u, γ_b, M_u, (M_b/ M_u), and $\Delta h_f / R_u$ must be determined. Values for M_u and M_b can be obtained from Eqs. (4.10) and (4.11). [b] Values of γ_u, γ_b, and $\Delta h_f / R_u$ can be obtained from graphs such as Fig. 4.2 (see Example 4.1 below). Values for γ_u, γ_b, and $\Delta h_f / R_u$ are available in the literature (e.g. Ref. 1) and this chapter, for a range of ϕ and x_b. Values used for computations should be checked over the temperature range of interest, to ensure that the particular linear fit to $u(T)$ used is appropriate.

EXAMPLE 4.1

Determine the values of γ_u, γ_b, and $\Delta h_f / R_u$ which correspond to the straight-line fits for $u_u(T)$ and $u_b(T)$ in Fig. 4.2.

Equations for the straight lines in Fig. 4.2 are

$$u_u(\text{kJ/kg air}) = 0.96T(\text{K}) - 700$$

and

$$u_b(\text{kJ/kg air}) = 1.5T(\text{K}) - 4250$$

From Table 4.5, for isooctane fuel with $\phi = 1.0$ and $x_b = 0.1$, the number of moles of unburned mixture per mole O_2 in the mixture is

$$n_u = 0.08 \times 1 + 4.773 + 0.28 \times 0.1 \times 1 = 4.881$$

The mass of air per mole O_2 in the mixture is 138.2. Thus, the number of moles of unburned mixture per unit mass of air in the original mixture is

$$\frac{4.881}{138.2} = 0.0353$$

The molar specific heat of the unburned mixture $\tilde{c}_{v,u}$ is therefore

$$\tilde{c}_{v,u} = \frac{0.96}{0.0353} = 27.2 \text{ kJ/kmol} \cdot \text{K}$$

Since $\tilde{R} = 8.314 \text{ kJ/kmol} \cdot \text{K}$,

$$\gamma_u = \frac{27.2 + 8.314}{27.2} = 1.31$$

The number of moles of burned mixture per mole O_2 is (from Table 4.5)

$$n_b = 0.36 \times 1 + 4.773 = 5.133$$

The number of moles of burned mixture per unit mass of air in the original mixture is

$$\frac{5.133}{138.2} = 0.0371$$

The molar specific heat $\tilde{c}_{v,b}$ is therefore

$$\tilde{c}_{v,b} = \frac{1.5}{0.0371} = 40.4 \text{ kJ/kmol} \cdot \text{K}$$

and γ_b is

$$\gamma_b = \frac{40.4 + 8.314}{40.4} = 1.21$$

To find $\Delta h_f / R_u$, R_u is given by

$$R_u = 8.314 \times 0.0353 = 0.293 \text{ kJ/kg air} \cdot \text{K}$$

and so

$$\frac{\Delta h_f}{R_u} = \frac{(-700) - (-4250)}{0.293} = 1.2 \times 10^4 \text{ K}$$

4.5 THERMODYNAMIC PROPERTY CHARTS

One method of presenting thermodynamic properties of unburned and burned gas mixtures for internal combustion engine calculations is on charts. Such charts in US units were developed by Hottel et al. [2] and Newhall and Starkman. [3, 4] We developed a new set of charts in SI units, following the approach of Newhall and Starkman. While charts are no longer used extensively for engine cycle calculations—computer models for the thermodynamic properties of working fluids have replaced them—these charts are useful for illustrative purposes, and afford an easy and accurate approach where the number of calculations is limited. The charts presented below are for isooctane fuel, and the following equivalence ratios: $\phi = 0.4$, 0.6, 0.8, 1.0, 1.2.

4.5.1 Unburned Mixture Charts

The thermodynamic properties of each unburned fuel-air mixture are represented by two charts. The first chart is designed to relate the mixture temperature, pressure, and volume at the beginning and at the end of the compression process; the second gives the mixture internal energy and enthalpy as functions of temperature.

The following assumptions are made:

1. The compression process is reversible and adiabatic.
2. The fuel is in the vapor phase.
3. The mixture composition is homogeneous and frozen (no reactions between the fuel and air).
4. Each species in the mixture can be modeled as an ideal gas.
5. The burned gas fraction is zero. [c]

It proves convenient to assign zero internal energy or enthalpy to the unburned mixture at 298.15 K. Internal energy and enthalpies relative to this datum are called *sensible internal energy* u_s *or sensible enthalpy* h_s. By sensible we mean changes in u or h which result from changes in temperature alone, and we exclude changes due to chemical reaction or phase change.

Table 4.6 provides the basic composition data for the unburned mixture charts. Equations (4.13 *a, b*) provide the basis for obtaining the $u_{s,u}(T)$ and $h_{s,u}(T)$ curves shown in Fig. 4.3.

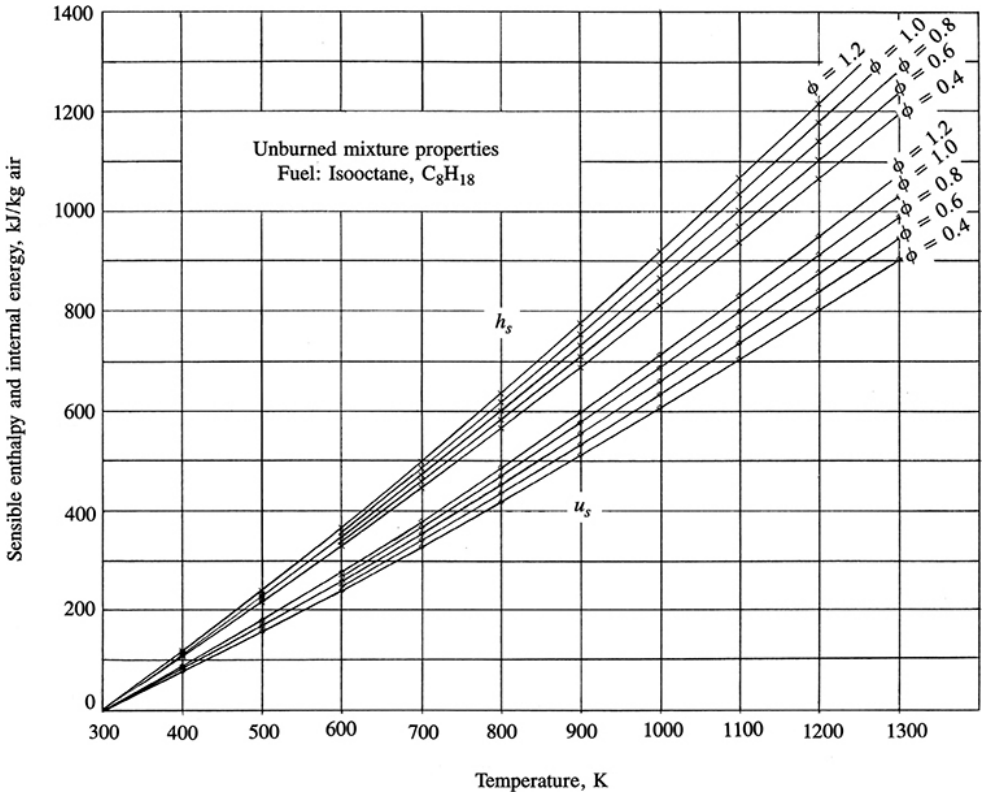


Figure 4.3 Sensible enthalpy and internal energy of unburned isooctane-air mixtures as function of temperature. Units: kJ/kg air in mixture.

TABLE 4.6 Unburned mixture composition for charts

Equivalence ratio ϕ	(F/A)	Kilograms of mixture per kilogram of air	Moles of mixture per mole of O_2	Kilomole of mixture per kilogram of air	${}^{u_u}\tilde{R}^{,*}$ J/kg air·K
0.4	0.0264	1.0264	$4.805 + 0.112x_b$	$0.0348 + 0.00081x_b$	289
0.6	0.0396	1.0396	$4.821 + 0.168x_b$	$0.0349 + 0.00122x_b$	290
0.8	0.0528	1.0528	$4.837 + 0.224x_b$	$0.0350 + 0.00162x_b$	291
1.0	0.0661	1.0661	$4.853 + 0.280x_b$	$0.0351 + 0.00203x_b$	292
1.2	0.0792	1.0792	$4.869 + 0.536x_b$	$0.0352 + 0.00388x_b$	292

*For $x_b = 0$. Error in neglecting x_b is usually small.

Equations (4.15) and (4.16) provide the basis for following a reversible adiabatic (i.e., isentropic) compression process. Between end states 1 and 2, we obtain, per kilogram of air in the mixture,

$$\Psi(T_2) = \Psi(T_1) - n_u \tilde{R} \ln\left(\frac{v_2}{v_1}\right) \tag{4.25a}$$

$$\Phi(T_2) = \Phi(T_1) + n_u \tilde{R} \ln\left(\frac{p_2}{p_1}\right) \tag{4.25b}$$

where n_u is the number of moles of unburned mixture per kilogram of air. Values of n_u and $n_u \tilde{R}$ are given in Table 4.6. $\Psi(T)$ and $\Phi(T)$ are given in Fig. 4.4. Note that v, p, and T are related by

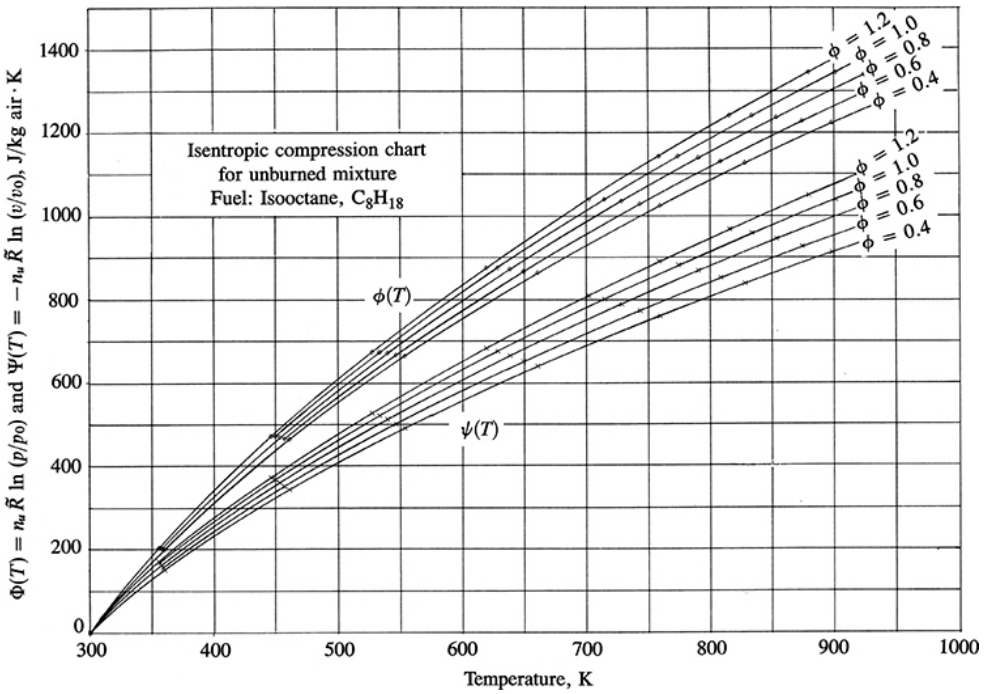


Figure 4.4 Isentropic compression functions, Φ and Ψ as function of temperature for unburned isooctane-air mixtures. Units: J/kg air · K.

$$p(\text{Pa})v(\text{m}^3/\text{kg air}) = n_u \tilde{R}(\text{J/kg air}\cdot\text{K})\, T(\text{K}) \tag{4.26}$$

EXAMPLE 4.2

The compression process in an internal combustion engine can be modeled approximately as adiabatic and reversible (i.e., as an isentropic process). A spark-ignition engine with a compression ratio of 8 operates with a

stoichiometric fuel vapor–air mixture, which is at 350 K and 1 atm at the start of the compression stroke. Find the temperature, pressure, and volume per unit mass of air at the end of the compression stroke. Calculate the compression stroke work.

Given $T_1 = 350$ K at the start of compression, find T_2 at the end of compression using the isentropic compression chart, Fig. 4.4, and Eq. (4.25 *a*) . For $T_1 = 350$ K, $\Psi_1 = 150$ J / kg air · K. From Eq. (4.25 *a*) ,

$$\Psi_2(T_2) = \Psi_1(T_1) - n_u \tilde{R} \ln\left(\frac{v_2}{v_1}\right) = 150 - 292 \ln\left(\frac{1}{8}\right) = 757 \text{ J/kg air} \cdot \text{K}$$

Figure 4.4 then gives

$$T_2 = 682 \text{ K}$$

The ideal gas law [Eq. (4.26)] gives

$$v_1 = \frac{292 \times 350}{1 \times 1.013 \times 10^5} = 1.0 \text{ m}^3/\text{kg air}$$

and

$$p_2 = p_1\left(\frac{T_2}{T_1}\right)\left(\frac{v_1}{v_2}\right) = \frac{682}{350} \times 8 = 15.5 \text{ atm}$$

$$v_2 = \frac{1.0}{8} = 0.125 \text{ m}^3/\text{kg air}$$

Note that p_2 can also be obtained from Fig. 4.4 and Eq. (4.25 *b*) :

$$\ln\left(\frac{p_2}{p_1}\right) = \frac{\Phi_2 - \Phi_1}{n_u \tilde{R}} = \frac{980 - 180}{292} = 2.74$$

$$p_2 = 15.5 \text{ atm} = 1.57 \text{ MPa}$$

The compression stroke work, assuming the process is adiabatic and using the data in Fig. 4.3, is

$$-W_{1-2} = u_s(T_2) - u_s(T_1) = 350 - 40 = 310 \text{ kJ/kg air}$$

4.5.2 Burned Mixture Charts

The primary burned mixture charts are for the products of combustion at high temperatures, that is, for the working fluid during the expansion process. The following assumptions are made:

1. Each species in the mixture can be modeled as an ideal gas.
2. The mixture is in thermodynamic and chemical equilibrium at temperatures above 1700 K; the mixture composition is frozen below 1700 K.
3. *Datum*. At the datum state of 298.15 K (25°C or 77°F) and 1 atm, the chemical elements in their naturally occurring form (N_2, O_2, H_2 as diatomic gases and C as solid graphite) are assigned zero enthalpy and entropy.

The charts were prepared with the NASA equilibrium program described in Sec. 3.7 .[5, 6] The C/H/O/N ratio of the mixture is specified for each chart. The intensive properties (internal energy, enthalpy, entropy, and specific volume) are all expressed per unit mass of air in the original mixture; that is, they correspond to the combustion of 1 kg of air with the appropriate mass of fuel. The mass basis for the unburned and burned mixture charts are the same.

Figures 4.5 to 4.9 are property charts for the high-temperature burned gas; each is a plot of internal energy versus entropy for a particular fuel and equivalence ratio. Lines of constant temperature, pressure, and specific volume are drawn on each chart. An illustration of the use of these charts follows.

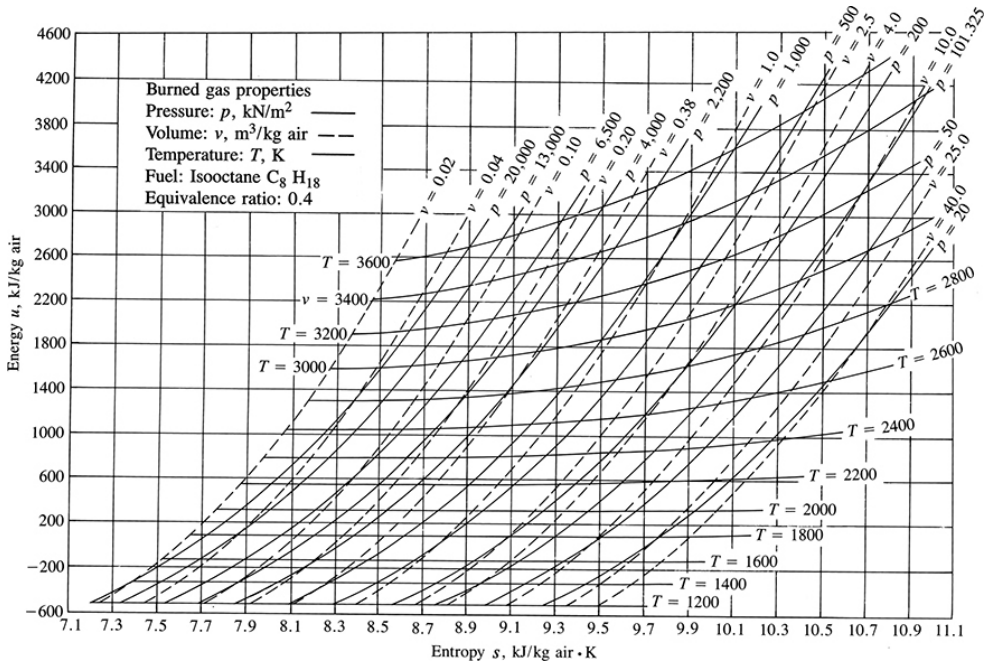


Figure 4.5 Internal energy versus entropy chart for equilibrium burned gas mixture, isooctane fuel; equivalence ratio 0.4.

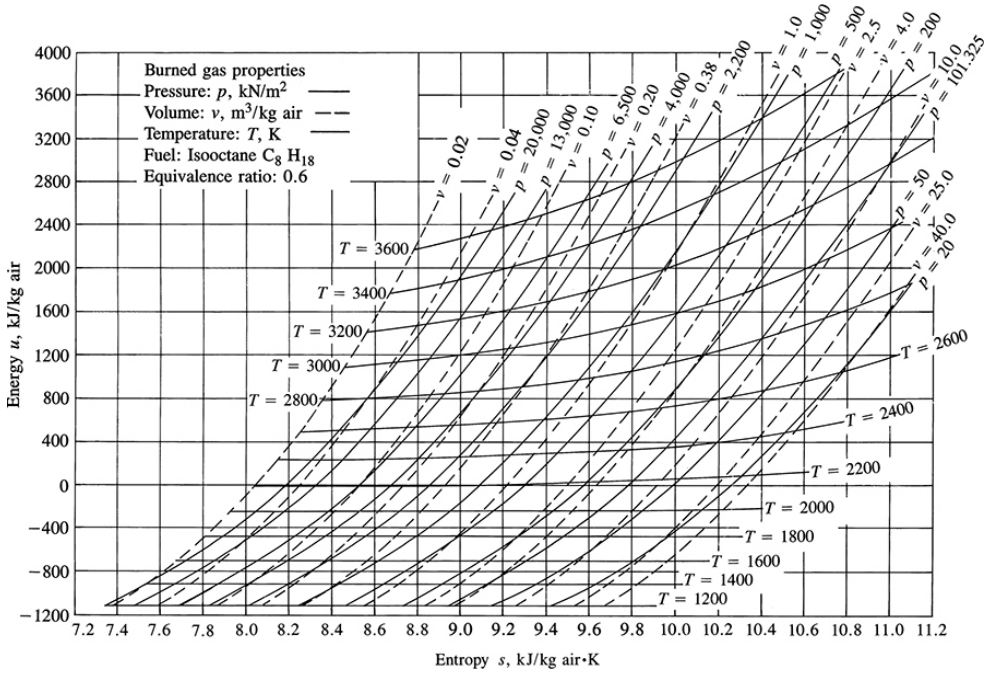


Figure 4.6 Internal energy versus entropy chart for equilibrium burned gas mixture, isooctane fuel; equivalence ratio 0.6.

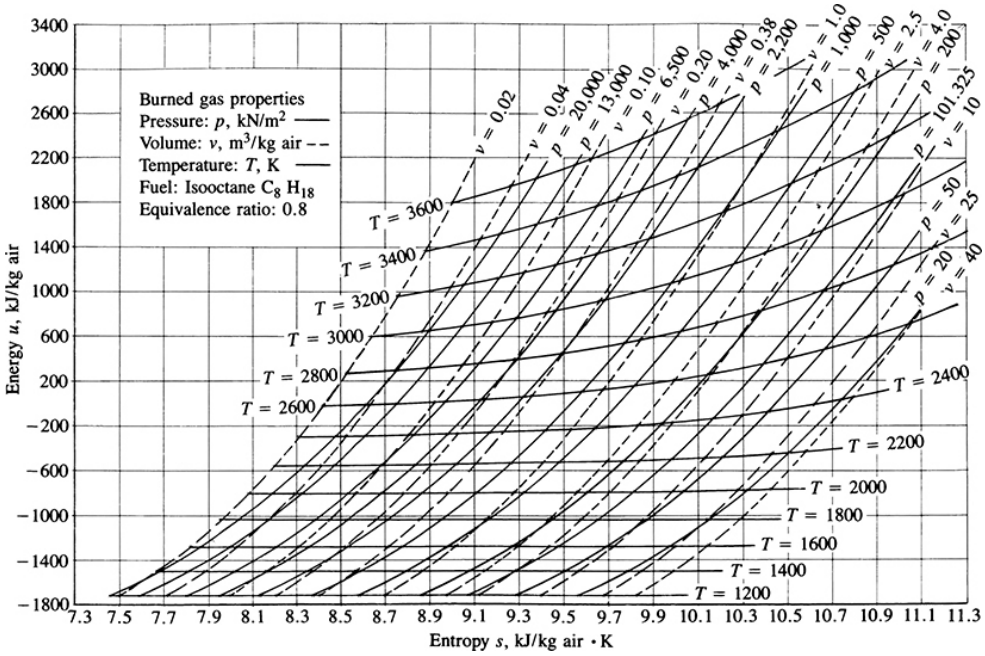


Figure 4.7 Internal energy versus entropy chart for equilibrium burned gas mixture, isooctane fuel; equivalence ratio 0.8.

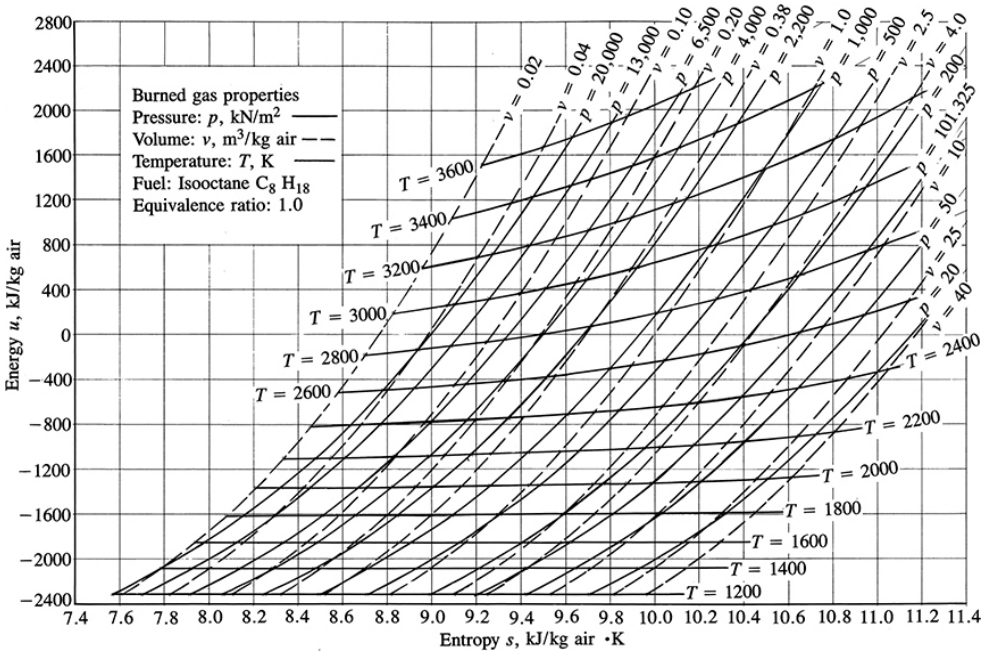


Figure 4.8 Internal energy versus entropy chart for equilibrium burned gas mixture, isooctane fuel; equivalence ratio 1.0.

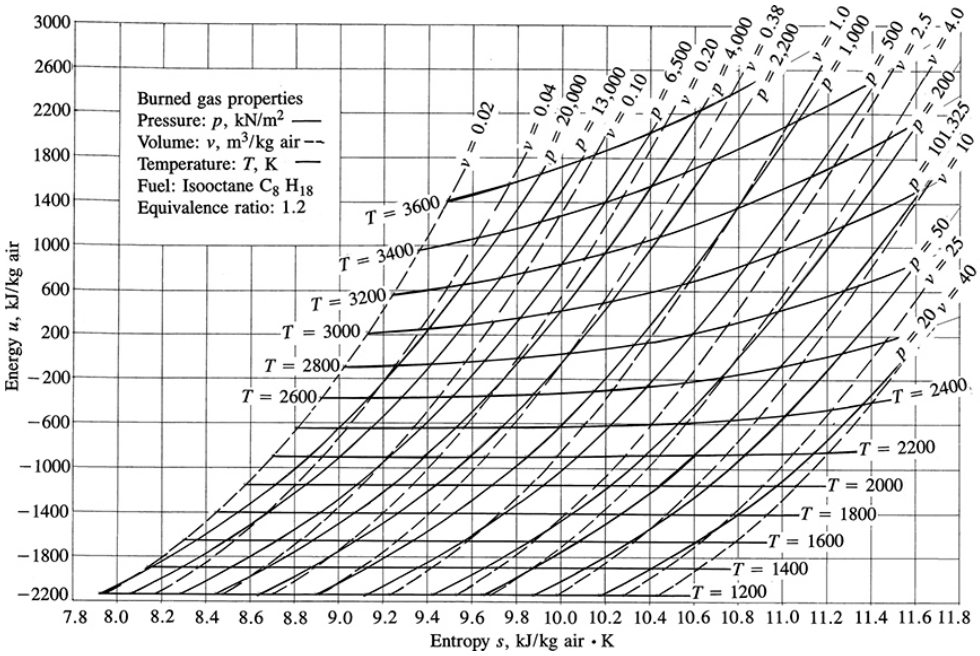


Figure 4.9 Internal energy versus entropy chart for equilibrium burned gas mixture, isooctane fuel; equivalence ratio 1.2.

EXAMPLE 4.3

The expansion process in an internal combustion engine, following completion of combustion, can be modeled approximately as an adiabatic and reversible process (i.e., isentropic). Under full-load operation, the pressure in the cylinder of a spark-ignition engine at top-center immediately following combustion is 7100 kPa. Find the gas state at the end of the expansion stroke and the expansion stroke work. The compression ratio is 8, the mixture is stoichiometric, and the volume per unit mass of air at the start of expansion is 0.125 m^3/kg air.

Locate $p_1 = 7100$ kPa and $v_1 = 0.125$ m^3/kg air on the $\phi = 1.0$ burned gas chart (Fig. 4.8). This gives $T_1 = 2825$ K, $u_1 = -5$ kJ/kg air, and $s_1 = 9.33$ kJ/kg air·K. The gas expands at constant entropy to $v_2 = 8 \times v_1 = 1$ m^3/kg air. Following a constant entropy process from state 1 on Fig. 4.8 gives

$$T_2 = 1840 \text{ K}, \qquad p_2 = 570 \text{ kPa}, \qquad \text{and} \qquad u_2 = -1540 \text{ kJ/kg air}$$

The expansion stroke work, assuming the process is adiabatic, is

$$W_{1-2} = -(u_2 - u_1) = 1540 - 5 = 1535 \text{ kJ/kg air}$$

As the burned gases in an engine cylinder cool during the expansion process, the composition eventually "freezes"—becomes fixed in composition—because the chemical reactions become extremely slow. This is usually assumed to occur at about 1700 K (see Sec. 4.9). The equilibrium composition assumption is then no longer valid. For lean and stoichiometric mixtures this distinction is not important because the mole fractions of dissociated species below this temperature are small. For rich mixtures, a frozen composition must be selected and used because the mole fractions of CO_2, CO, H_2O, and H_2 would continue to change if equilibrium is assumed as the temperature decreases. Internal energy and enthalpy, per kilogram of air in the original mixture, of the frozen burned mixture are plotted against temperature in Fig. 4.10. The assumed frozen burned mixture compositions are listed in Table 4.7. These are sensible internal energies and enthalpies, given relative to their values at 298.15 K.

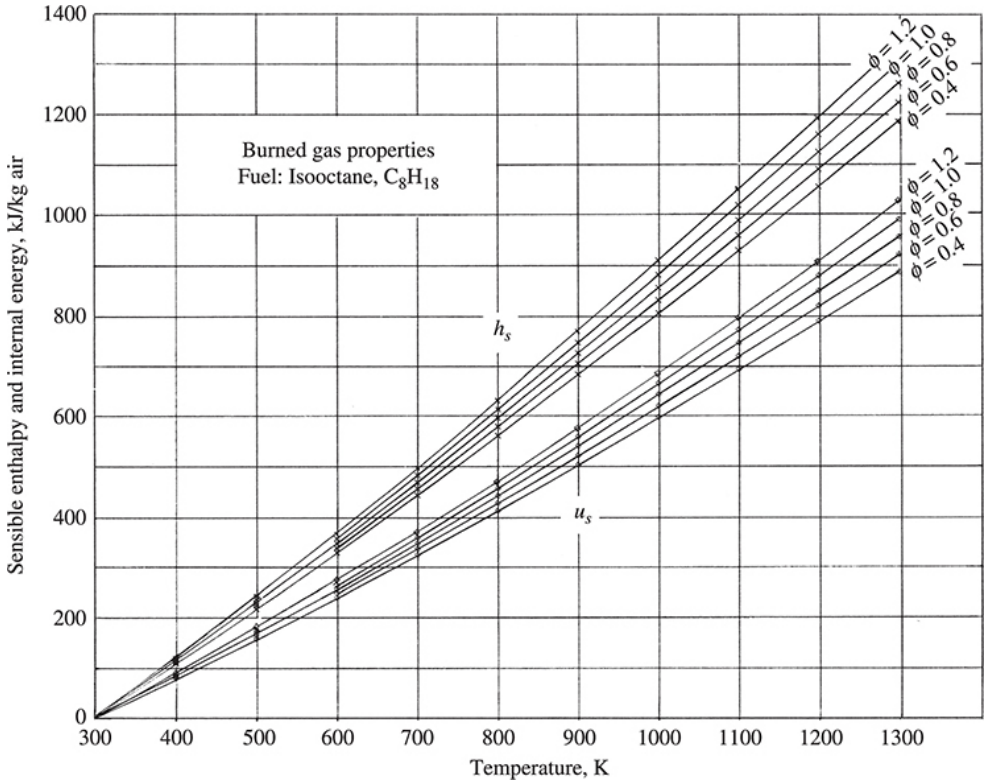


Figure 4.10 Sensible enthalpy and internal energy of low-temperature burned gases as function of temperature, isooctane fuel. Units: kJ/kg air in original mixture.

TABLE 4.7 Frozen burned gas composition: C_8H_{18}-air combustion

ϕ	CO_2	H_2O	CO	H_2	O_2	N_2	Sum	Units
0.4	0.0521	0.0586	—	—	0.122	0.767	1.000	mole fractions
	1.85	2.08	—	—	4.34	27.3	35.6	mol/kg air*
0.6	0.0770	0.0866	—	—	0.0802	0.756	1.000	mole fractions
	2.78	3.13	—	—	2.89	27.3	36.1	mol/kg air*
0.8	0.101	0.113	—	—	0.0395	0.746	1.000	mole fractions
	3.70	4.14	—	—	1.45	27.3	36.6	mol/kg air*
1.0	0.125	0.140	—	—	—	0.735	1.000	mole fractions
	4.64	5.2	—	—	—	27.3	37.1	mol/kg air*
1.2†	0.0905	0.138	0.0516	0.0224	—	0.698	1.000	mole fractions
	3.54	5.38	2.02	0.876	—	27.3	39.1	mol/kg air*

*Note mol/kg air; multiply by 10^{-3} for kmol/kg air.
Note, to obtain these burned-gas mole fractions, $K(T)$ in Eq. (4.6) was evaluated at 1740 K; $K = 3.5$.

4.5.3 Relation between Unburned and Burned Mixture Charts

We now address the questions: Given unburned mixture at T_1, p_1, v_1, what is the state of the burned mixture following (1) constant-volume adiabatic combustion or (2) constant-pressure adiabatic combustion?

The datum for internal energy and enthalpy for the unburned mixture in Fig. 4.3 is different from the datum for internal energy and enthalpy for the burned mixture. For the unburned mixture, zero internal energy and enthalpy for the *mixture* at 298.15 K was assumed. For the burned mixture, zero enthalpy for the gaseous species O_2, N_2, and H_2, and C (solid graphite) at 298.15 K was assumed. These data can be related through the enthalpies of formation, from O_2, N_2, H_2, and C, of each species in the unburned mixture.

If $\Delta\tilde{h}^{\circ}_{f,i}$ is the enthalpy of formation of species i at 298.15 K, per kilomole, and $\Delta h^{\circ}_{f,u}$ is the enthalpy of formation of the unburned mixture at 298.15 K, per kilogram of air in the original mixture, then

$$\Delta h^{\circ}_{f,u} = \sum_i n_i \Delta\tilde{h}^{\circ}_{f,i} \tag{4.27}$$

where n_i is the number of kilomoles of species i per kilogram of air. The unburned mixture enthalpy h_u, with the same datum as the burned mixture

enthalpy, is therefore given by the sum of the sensible enthalpy $h_{s,u}$ and $\Delta h^\circ_{f,u}$:

$$h_u = h_{s,u} + \Delta h^\circ_{f,u} \tag{4.28}$$

Similarly, the internal energy u_u is given by

$$u_u = u_{s,u} + \Delta u^\circ_{f,u} \tag{4.29}$$

$\Delta u^\circ_{f,u}$ can be obtained from

$$\Delta u^\circ_{f,u} = \sum_i n_i \, \Delta \tilde{u}^\circ_{f,i} \tag{4.30}$$

Alternately, Eq. (3.18) can be used to obtain $\Delta u^\circ_{f,u}$ from $\Delta h^\circ_{f,u}$:

$$\Delta u^\circ_{f,u} = \Delta h^\circ_{f,u} - (n_P - n_R)\tilde{R}T \tag{4.31}$$

Enthalpies and internal energies of formation of the relevant burned gas species and individual fuel compounds are given in Table 4.8 and App. D. Values of n_i are obtained from Tables 4.4 and 4.7. Following the procedure used in Example 4.4 below, expressions for $\Delta h^\circ_{f,u}$ and $\Delta u^\circ_{f,u}$ in kilojoules per kilogram of air can be obtained. For the charts of Figs. 4.3 and 4.5 to 4.9, these expressions are:

TABLE 4.8 Standard enthalpies and internal energies of formation*

	$\Delta h^\circ_{f,i}$, MJ/kmol	$\Delta \tilde{u}^\circ_{f,i}$, MJ/kmol
CO_2	−393.5	−393.5
H_2O (gas)	−241.8	−240.6
CO	−110.5	−111.7
C_8H_{18} (gas)	−224.1	−204.3

*At 298.15 K. $\Delta h^\circ_{f,i}$ for O_2, N_2, and H_2 is zero by definition.
Sources: JANAF tables;[1] Rossini et al.[7]

$$
\begin{aligned}
&\phi = 0.4: \quad \Delta h^{\circ}_{f,u} = -51.9 - 1181 x_b \quad \Delta u^{\circ}_{f,u} = -47.3 - 1183 x_b \\
&\phi = 0.6: \quad \Delta h^{\circ}_{f,u} = -77.8 - 1771 x_b \quad \Delta u^{\circ}_{f,u} = -70.9 - 1774 x_b \\
&\phi = 0.8: \quad \Delta h^{\circ}_{f,u} = -103.8 - 2361 x_b \quad \Delta u^{\circ}_{f,u} = -94.6 - 2365 x_b \\
&\phi = 1.0: \quad \Delta h^{\circ}_{f,u} = -129.7 - 2951 x_b \quad \Delta u^{\circ}_{f,u} = -118.2 - 2956 x_b \\
&\phi = 1.2: \quad \Delta h^{\circ}_{f,u} = -155.6 - 2759 x_b \quad \Delta u^{\circ}_{f,u} = -141.9 - 2769 x_b
\end{aligned}
\tag{4.32}
$$

EXAMPLE 4.4

Calculate $\Delta h^{\circ}_{f,u}$, the enthalpy of formation of the unburned mixture, and $\Delta u^{\circ}_{f,u}$ the internal energy of formation of the unburned mixture, for a C_8H_{18}-air mixture with $\phi = 1.0$ and burned gas fraction x_b.

Table 4.4 gives the moles of each species in the unburned mixture, per mole O_2 with $\phi = 1.0$, as

$C_8H_{18}, 0.08(1 - x_b)$	$CO_2, 0.64 x_b$
$O_2, 1 - x_b$	$H_2O, 0.72 x_b$
$N_2, 3.773$	CO and $H_2, 0$

Table 4.5 gives the mass of air per mole O_2 as 138.2 kg/kmol. Thus the number of kilomoles of each species per kilogram of air is

$C_8H_{18}, 5.787 \times 10^{-4}(1 - x_b)$	$CO_2, 4.629 \times 10^{-3} x_b$
$O_2, 7.233 \times 10^{-3}(1 - x_b)$	$H_2O, 5.208 \times 10^{-3} x_b$
$N_2, 2.729 \times 10^{-2}$	CO and $H_2, 0$

With $\Delta \tilde{h}^{\circ}_{f,i}$ from Table 4.8, Eq. (4.27) gives

$$
\begin{aligned}
\Delta h^{\circ}_{f,u} &= 5.787 \times 10^{-4} \times (-224.1 \times 10^6)(1 - x_b) \\
&\quad + x_b \left[4.629 \times 10^{-3} \times (-393.5 \times 10^6) + 5.208 \times 10^{-3} \times (-241.8 \times 10^6) \right] \\
\Delta h^{\circ}_{f,u} &= (-129.7 - 2951 x_b) \times 10^3 \text{ kJ/kg air}
\end{aligned}
$$

With $\Delta \tilde{u}^{\circ}_{f,i}$ from Table 4.8, Eq. (4.30) gives

$$\Delta u^{\circ}_{f,u} = 5.787\times10^{-4}\times(-204.3\times10^{6})(1-x_b) + x_b\left[4.629\times10^{-3}\times(-393.5\times10^{6}) + 5.208\times10^{-3}\times(-240.6\times10^{6})\right]$$

$$\Delta u^{\circ}_{f,u} = (-118.2 - 2956x_b)\times10^{3} \text{ J/kg air}$$

Alternatively, we can determine $\Delta u^{\circ}_{f,u}$ from $\Delta h^{\circ}_{f,u}$ using Eq. (4.31) . For this calculation, the "product" gas is the unburned mixture and the "reactant" gas is the mixture of elements from which the unburned mixture is formed. The number of gaseous moles in the unburned mixture n_p, per mole O_2 in the original mixture, is (from Table 4.5 for $\phi \le 1$)

$$n_p = (1-x_b)\left[4(1+2\varepsilon)\frac{\phi}{M_f} + 1 + \psi\right] + x_b\left[(1-\varepsilon)\phi + 1 + \psi\right]$$

The elemental reactant mixture from which the unburned mixture is formed is, from Eq. (4.4) ,

$$\varepsilon\phi C + 2(1-\varepsilon)\phi H_2 + O_2 + \psi N_2$$

Thus, n_R, the moles of *gaseous* elements, is

$$n_R = 2(1-\varepsilon)\phi + 1 + \psi$$

For air, $\psi = 3.773$; for C_8H_{18} fuel, $\varepsilon = 0.64$ and $M_f = 114$. For $\phi = 1$,

$$n_p - n_R = -0.64 + 0.28x_b \qquad \text{moles/mole } O_2$$

and

$$(n_p - n_R)\tilde{R}T = (-0.64 + 0.28x_b)\times8.3143\times10^{3}\times\frac{298.15}{138.2}$$

or

$$(n_p - n_R)\tilde{R}T = (-11.5 + 5.0x_b)\times10^{3} \text{ J/kg air}$$

Since

$$\Delta u^{o}_{f,u} = \Delta h^{o}_{f,u} - (n_P - n_R)\tilde{R}T$$
$$\Delta u^{o}_{f,u} = (-129.7 - 2951x_b)\times 10^3 - (-11.5 + 5.0x_b)\times 10^3$$
$$\Delta u^{o}_{f,u} = (-118.2 - 2956x_b)\times 10^3 \text{ J/kg air}$$

The combustion process links the unburned and burned mixture properties as follows: For an adiabatic constant-volume combustion process,
and

$$u_b = u_u = u_{s,u} + \Delta u^{o}_{f,u} \qquad (4.33)$$
$$v_b = v_u$$

Thus, given $u_{s,u}$ and v_u, the state of the burned mixture can be determined from the appropriate burned mixture chart.

For an adiabatic constant-pressure combustion process,

$$h_b = h_u = h_{s,u} + \Delta h^{o}_{f,u} \qquad (4.34)$$

Since

$$u_b = h_b - pv_b$$

given $h_{s,u}$ and p, u_b *and* v_b must be found by trial and error along the specified constant-pressure line on the appropriate burned mixture chart.

EXAMPLE 4.5

Calculate the temperature and pressure after constant-volume adiabatic combustion and constant-pressure adiabatic combustion of the unburned mixture (with $\phi = 1.0$ and $x_b = 0.08$) at the state corresponding to the end of the compression process examined in Example 4.2.

The state of the unburned mixture at the end of the compression process in Example 4.2 was

$$T_u = 682 \text{ K}, \quad u_{s,u} = 350 \text{ kJ/kg air}, \quad p_u = 1.57 \text{ MPa}, \quad v_u = 0.125 \text{ m}^3/\text{kg air}$$

For an adiabatic *constant-volume* combustion process [Eq. (4.33)],

$$u_b = u_u = u_{s,u} + \Delta u^\circ_{f,u}$$

For $\phi = 1.0$, $\Delta u^\circ_{f,u}$ is given by Eq. (4.32) as

$$\Delta u^\circ_{f,u} = -118.2 - 2956 x_b = -118.2 - 236.5 = -355 \text{ kJ/kg air}$$

Hence

$$u_b = 350 - 355 = -5 \text{ kJ/kg air}$$

Also

$$v_b = v_u = 0.125 \text{ m}^3/\text{kg air}$$

Locating (u_b, v_b) on the burned gas chart (Fig. 4.8) gives

$$T_b = 2825 \text{ K}, \qquad p_b = 7100 \text{ kPa}$$

For a *constant-pressure* combustion process [Eq. (4.34)],

$$h_b = h_u = h_{s,u} + \Delta h^\circ_{f,u}$$

For $\phi = 1.0$, $\Delta h^\circ_{f,u}$ is given by Eq. (4.32) as

$$\Delta h^\circ_{f,u} = -129.7 - 2951 x_b = -129.7 - 236 = -366 \text{ kJ/kg air}$$

At $T_u = 682$ K, $h_{s,u} = 465$ kJ/kg air, so

$$h_b = 465 - 366 = 99 \text{ kJ/kg air}$$

Since $p_b = p_u = 1.57$ MPa, the internal energy u_b is given by

$$u_b = h_b - p_b v_b = 99 - 1.57 \times 10^3 v_b \text{ kJ/kg air}$$

A trial–and–error solution for v_b and u_b along the p = 1570 kPa line on Fig. 4.8 gives

$$u_b = -655 \text{ kJ/kg air}, \quad T_b = 2440 \text{ K}, \quad v_b = 0.485 \text{ m}^3/\text{kg air}$$

(Use the ideal gas law to estimate p, T, or v more accurately.)

4.6 TABLES OF PROPERTIES AND COMPOSITION

Tables of thermodynamic properties of air are useful for analysis of motored engine operation, diesels, and compressors. Keenan, Chao, and Kaye's *Gas Tables* [8] are the standard reference for the thermodynamic properties of air at low pressures (i.e., at pressures substantially below the critical pressure when the ideal gas law is accurate). These gas tables are in U.S. and SI units. A set of tables for air in SI units has been prepared by Reynolds [9] following the format of the Keenan et al. tables. A condensed table of thermodynamic properties of air, derived from Reynolds, is given in App. D. It contains:

h = enthalpy, kJ/kg
u = internal energy, kJ/kg

$$\Psi = \int_0^T \left(\frac{c_v}{T}\right) dT, \text{ kJ/kg} \cdot \text{K}$$

$$\Phi = \int_0^T \left(\frac{c_p}{T}\right) dT, \text{ kJ/kg} \cdot \text{K}$$

p_r = relative pressure
v_r = relative volume
c_p = specific heat at constant pressure, kJ/kg·K
c_v = specific heat at constant volume, kJ/kg·K
γ = ratio of specific heats

all as a function of T(K).

Ψ and Φ are the standard state entropy at temperature T and 1 atm pressure relative to the entropy at 0 K and 1 atm pressure. [See Eqs. (4.14) to (4.17).] The entropy at pressures other that 1 atm is obtained using Eq. (4.14) .

The relative pressure p_r is defined by

$$\ln p_r = \frac{\Phi}{R} \tag{4.35}$$

and is a function of T only. Along a given isentropic, it follows from Eq. (4.18) that the ratio of actual pressures p_2 and p_1 corresponding to temperatures T_2 and T_1 is equal to the ratio of relative pressures, that is,

$$\left(\frac{p_2}{p_1}\right)_{s=\text{const}} = \left(\frac{p_{r_2}}{p_{r_1}}\right) \tag{4.36}$$

This affords a means of determining T_2, for an isentropic process, given T_1 and p_2/p_1 (see Example 4.6).

The relative volume v_r is defined by

$$v_r = \frac{RT}{p_r} \tag{4.37}$$

The units are selected so that v_r is in cubic meters per kilogram when T is in kelvins and p_r is in pascals. Along a given isentropic, the ratio of actual volumes V_2 and V_1 (for a fixed mass) at temperatures T_2 and T_1, from Eq. (4.37) , is equal to the ratio of relative volumes

$$\left(\frac{V_2}{V_1}\right)_{s=\text{const}} = \left(\frac{v_{r_2}}{v_{r_1}}\right) \tag{4.38}$$

This affords a means of determining T_2 for an isentropic process, given T_1 and V_2/V_1 (see Example 4.6).

Tables giving the composition and thermodynamic properties of combustion products have been compiled. They are useful sources of property and species concentrations data in burned gas mixtures for a range of equivalence ratios, temperatures, and pressures. Summary information on four generally available sets of tables is given in Table 4.9. The most extensive set of tables of combustion product composition and thermodynamic properties is the AGARD set, *Properties of Air and Combustion Products with Kerosene and Hydrogen Fuels*, by Banes et al. [10] Note, however, that their enthalpy datum differs from the usual datum (enthalpy for O_2, N_2, H_2, and C is zero at 298.15 K). The elements in their reference state at 298.15 K were assigned arbitrary positive values for enthalpy to avoid negative enthalpies for the equilibrium burned gas mixture.

TABLE 4.9 Tables of properties of air and combustion products

Source	Properties P, composition C	Mixture	Units	ϕ range	T range	p range	Enthalpy datum
Keenan et al.[8]	P	Air	U.S., SI		100–3600 K	Low	$h = 0$ at 0 K
		$(CH_2)_n$-air		0.25, 0.5, 1.0	100–2000 K	Low	$\tilde{h}_b = 0$ at 0 K
Reynolds[9]	P	Air	SI		200–1500 K	Low	
General Electric[11]	P and C	Air-$(CH_2)_n$	U.S.	0.25–4	600–5000°R	0.01–30 atm	h of C, H_2, N_2, O_2 zero at 0°R
AGARD[10]	P and C	Air	SI		100–6000 K	1–800 atm	Arbitrary, to keep $h_b > 0$
		$(CH_2)_n$-air		0.2–2			
		H_2-air		0.2–2			

EXAMPLE 4.6

In a diesel engine, the air conditions at the start of compression are $p_1 = 1$ atm and $T_1 = 325$ K. At the end of compression $p_2 = 60$ atm. Find the temperature T_2 and the compression ratio V_1/V_2.

Air tables (see App. D), at $T_1 = 325$ K, give

$$p_{r_1} = 97.13 \qquad \text{and} \qquad v_{r_1} = 960.6$$

Use Eq. (4.36) ,

$$\frac{p_{r_2}}{p_{r_1}} = \frac{p_2}{p_1} = 60$$

to give

$$p_{r_2} = 5828$$

Tables then give

$$T_2 = 992 \text{ K} \qquad \text{and} \qquad v_{r_2} = 48.92$$

The compression ratio is given by

$$\frac{V_1}{V_2} = \frac{v_{r_1}}{v_{r_2}} = \frac{960.6}{48.92} = 19.6$$

4.7 COMPUTER ROUTINES FOR PROPERTY AND COMPOSITION CALCULATIONS

When large numbers of computations are being made and high accuracy is required, engine process calculations are carried out on a computer. Relationships which model the composition and/or thermodynamic properties of unburned and burned gas mixtures have been developed for computer use.

The most complete models are based on polynomial curve fits to the thermodynamic data for each species in the mixture and the assumptions that (1) the unburned mixture is frozen in composition and (2) the burned mixture is in equilibrium. The approach used as the basis for representing JANAF table thermodynamic data [1] in the NASA equilibrium program [5, 6] (see Sec. 3.7) will be summarized here because it is consistent with the approach used throughout to calculate unburned and burned mixture properties.

For each species i in its standard state at temperature T(K), the specific heat $\tilde{c}_{p,i}$ is approximated by

$$\frac{\tilde{c}_{p,i}}{R} = a_{i1} + a_{i2}T + a_{i3}T^2 + a_{i4} + a_{i5}T^4 \tag{4.39}$$

The standard state enthalpy of species *i* is then given by

$$\frac{\tilde{h}_i}{\tilde{R}T} = a_{i1} + \frac{a_{i2}}{2}T + \frac{a_{i3}}{3}T^2 + \frac{a_{i4}}{4}T^3 + \frac{a_{i5}}{5}T^4 + \frac{a_{i6}}{T} \tag{4.40}$$

The standard state entropy of species *i* at temperature T(K) and pressure 1 atm, from Eq. (4.14) , is then

$$\frac{\tilde{s}_i}{\tilde{R}} = a_{i1}\ln T + a_{i2}T + \frac{a_{i3}}{2}T^2 + \frac{a_{i4}}{3}T^3 + \frac{a_{i5}}{4}T^4 + a_{i7} \tag{4.41}$$

Values of the coefficients *aij* for CO_2, H_2O, CO, H_2, O_2, N_2, OH, NO, O, and H from the NASA program are given in Table 4.10. Two temperature ranges are given. The 300 to 1000 K range is appropriate for unburned mixture property calculations. The 1000 to 5000 K range is appropriate for burned mixture property calculations. Figure 4.11 gives values of c_p/R for the major species, CO_2, H_2O, O_2, N_2, H_2, and CO, as a function of temperature.

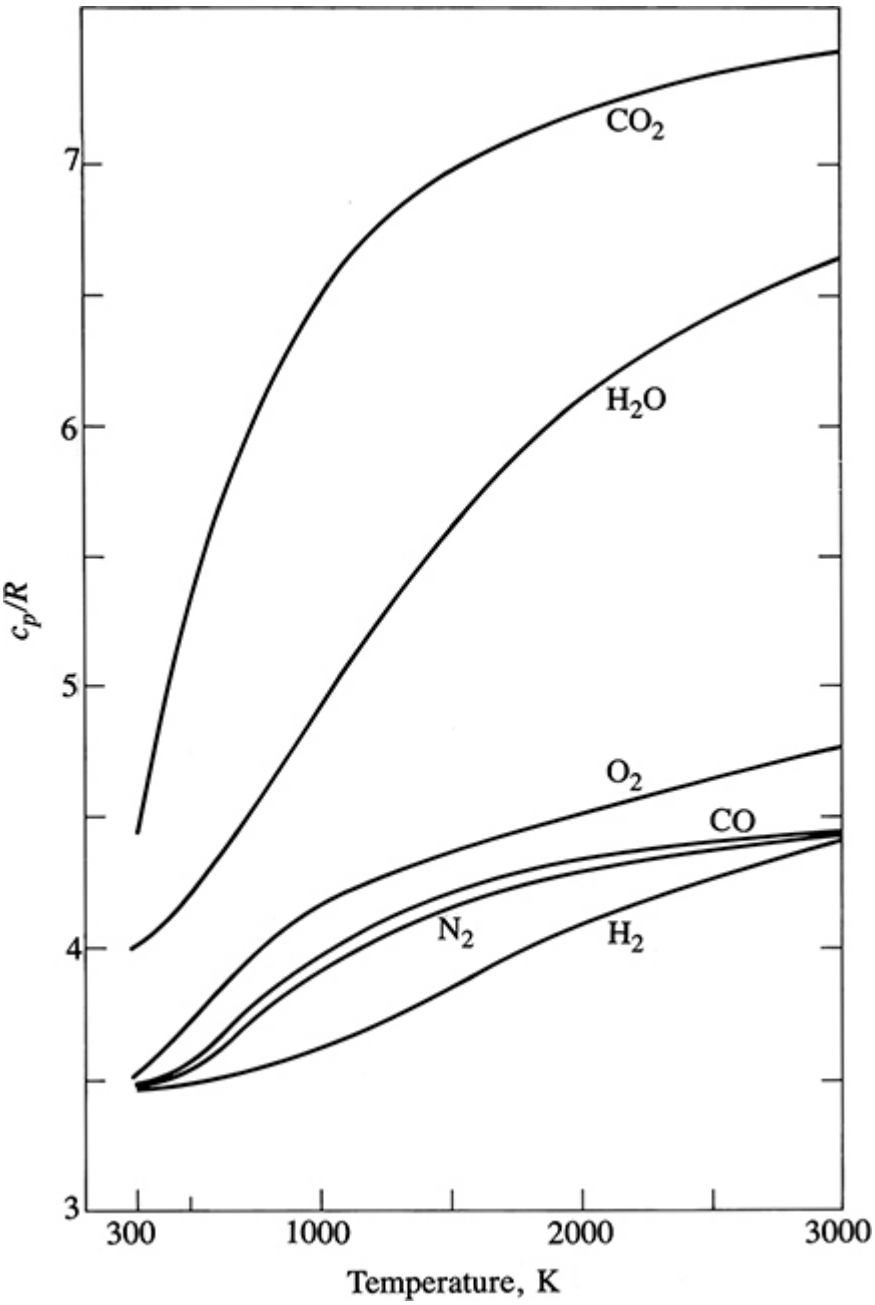


Figure 4.11 Specific heat at constant pressure, c_p/R, as function of temperature for species CO_2, H_2O, O_2, N_2, H_2, and CO. (*From JANAF tables*.[1])

TABLE 4.10 Coefficients for species thermodynamic properties

Species	T range, K	a_{i1}	a_{i2}	a_{i3}	a_{i4}	a_{i5}	a_{i6}	a_{i7}
CO_2	1000–5000	0.44608(+1)	0.30982(−2)	−0.12393(−5)	0.22741(−9)	−0.15526(−13)	−0.48961(+5)	−0.98636(0)
	300–1000	0.24008(+1)	0.87351(−2)	−0.66071(−5)	0.20022(−8)	0.63274(−15)	−0.48378(+5)	0.96951(+1)
H_2O	1000–5000	0.27168(+1)	0.29451(−2)	−0.80224(−6)	0.10227(−9)	−0.48472(−14)	−0.29906(+5)	0.66306(+1)
	300–1000	0.40701(+1)	−0.11084(−2)	0.41521(−5)	−0.29637(−8)	0.80702(−12)	−0.30280(+5)	−0.32270(0)
CO	1000–5000	0.29841(+1)	0.14891(−2)	−0.57900(−6)	0.10365(−9)	−0.69354(−14)	−0.14245(+5)	0.63479(+1)
	300–1000	0.37101(+1)	−0.16191(−2)	0.36924(−5)	−0.20320(−8)	0.23953(−12)	−0.14356(+5)	0.29555(+1)
H_2	1000–5000	0.31002(+1)	0.51119(−3)	0.52644(−7)	−0.34910(−10)	0.36945(−14)	−0.87738(+3)	−0.19629(+1)
	300–1000	0.30574(+1)	0.26765(−2)	−0.58099(−5)	0.55210(−8)	−0.18123(−11)	−0.98890(+3)	−0.22997(+1)
O_2	1000–5000	0.36220(+1)	0.73618(−3)	−0.19652(−6)	0.36202(−10)	−0.28946(−14)	−0.12020(+4)	0.36151(+1)
	300–1000	0.36256(+1)	−0.18782(−2)	0.70555(−5)	−0.67635(−8)	0.21556(−11)	−0.10475(+4)	0.43053(+1)
N_2	1000–5000	0.28963(+1)	0.15155(−2)	−0.57235(−6)	0.99807(−10)	−0.65224(−14)	−0.90586(+3)	0.61615(+1)
	300–1000	0.36748(+1)	−0.12082(−2)	0.23240(−5)	−0.63218(−9)	−0.22577(−12)	−0.10612(+4)	0.23580(+1)
OH	1000–5000	0.29106(+1)	0.95932(−3)	−0.19442(−6)	0.13757(−10)	0.14225(−15)	0.39354(+4)	0.54423(+1)
NO	1000–5000	0.31890(+1)	0.13382(−2)	−0.52899(−6)	0.95919(−10)	−0.64848(−14)	0.98283(+4)	0.67458(+1)
O	1000–5000	0.25421(+1)	−0.27551(−4)	−0.31028(−8)	0.45511(−11)	−0.43681(−15)	0.29231(+5)	0.49203(+1)
H	1000–5000	0.25(+1)	0.0	0.0	0.0	0.0	0.25472(+5)	−0.46012(0)

Source: NASA Equilibrium Code.[5]

4.7.1 Unburned Mixtures

Polynomial functions for various fuels (in the vapor phase) have been fitted to the functional form: [12–14]

$$\tilde{c}_{p,f} = A_{f1} + A_{f2}t + A_{f3}t^2 + A_{f4}t^3 + \frac{A_{f5}}{t^2} \tag{4.42}$$

$$\tilde{h}_f = A_{f1}t + A_{f2}\frac{t^2}{2} + A_{f3}\frac{t^3}{3} + A_{f4}\frac{t^4}{4} - \frac{A_{f5}}{t} + A_{f6} + A_{f8} \tag{4.43}$$

where $t = T(\mathrm{K})/1000$. A_{f6} is the constant for the datum of zero enthalpy for C, H_2, O_2, and N_2 at 298.15 K. For a 0-K datum, A_{f8} is added to A_{f6}. For pure hydrocarbon compounds, the coefficients A_{fi} were found by fitting Eqs. (4.42) and (4.43) to data from Rossini et al.[7] Values for relevant pure fuels are given in Table 4.11. The units for $\tilde{c}_{p,f}$ are cal/gmol·K, and for $\tilde{h}_f$ are kcal/gmol.

TABLE 4.11 Coefficients for polynomials [Eqs. (4.42) and (4.43)] for fuel enthalpy and specific heat

Fuel	Formula	Molecular weight	$(A/F)_s$	$(F/A)_s$	A_{f1}	A_{f2}	A_{f3}	A_{f4}	A_{f5}	A_{f6}	A_{f8}
Methane	CH_4	16.04	17.23	0.0580	−0.29149	26.327	−10.610	1.5656	0.16573	−18.331	4.3000
Propane	C_3H_8	44.10	15.67	0.0638	−1.4867	74.339	−39.065	8.0543	0.01219	−27.313	8.852
Hexane	C_6H_{14}	86.18	15.24	0.0656	−20.777	210.48	−164.125	52.832	0.56635	−39.836	15.611
Isooctane	C_8H_{18}	114.2	15.14	0.0661	−0.55313	181.62	−97.787	20.402	−0.03095	−60.751	20.232
Methanol	CH_3OH	32.04	6.47	0.1546	−2.7059	44.168	−27.501	7.2193	0.20299	−48.288	5.3375
Ethanol	C_2H_5OH	46.07	9.00	0.1111	6.990	39.741	−11.926	0	0	−60.214	7.6135
Gasoline	$C_{8.26}H_{15.5}$	114.8	14.64	0.0683	−24.078	256.63	−201.68	64.750	0.5808	−27.562	17.792
	$C_{7.76}H_{13.1}$	106.4	14.37	0.0696	−22.501	227.99	−177.26	56.048	0.4845	−17.578	15.235
Diesel	$C_{10.8}H_{18.7}$	148.6	14.4	0.0694	−9.1063	246.97	−143.74	32.329	0.0518	−50.128	23.514

Units of A_{fj} such that $\tilde{h}_f$ is in kcal/gmol and $\tilde{c}_{p,f}$ is in cal/gmol-K with $t = T(\mathrm{K})/1000$.
A_{f6} gives enthalpy datum at 298.15 K; $(A_{f6} + A_{f8})$ gives enthalpy datum at 0 K.

Multicomponent fuel coefficients were determined as follows. [13] Chemical analysis of the fuel was performed to obtain the H/C ratio, average molecular weight, heating value, and the weight percent of aromatics, olefins, and total paraffins (including cycloparaffins). The fuel was then modeled as composed of a representative aromatic, olefin, and paraffin hydrocarbon. From atomic conservation of hydrogen and carbon and the chemical analysis results, component molar fractions and average carbon numbers can be determined. Table 4.11 gives values for the coefficients A_{f1} to A_{f18} for typical petroleum-based fuels. The units of the coefficients give $\tilde{c}_{p,f}$ and $\tilde{h}_f$ in cal/gmol·K and kcal/gmol, respectively, with $t = T(\mathrm{K})/1000$.

The thermodynamic properties of the unburned mixture can now be obtained. With the moles of each species per mole O_2, n_i, determined from Table 4.4, and the mass of mixture per mole O_2, m_{RP}, determined from Table 4.5, the unburned mixture properties are given by

$$c_{p,u} = \frac{1}{m_{\mathrm{RP}}} \sum_i n_i \tilde{c}_{p,i} \tag{4.44a}$$

$$h_u = \frac{1}{m_{\mathrm{RP}}} \sum_i n_i \tilde{h}_i \tag{4.44b}$$

$$s_u = \frac{1}{m_{\mathrm{RP}}} \left\{ \sum n_i \left[\tilde{s}_i^\circ - \tilde{R} \ln\left(\frac{n_i}{n_u} \right) \right] - n_u \tilde{R} \ln p \right\} \tag{4.44c}$$

where p is in atmospheres.

Figures 4.12 and 4.13, obtained with the above relations, show how $c_{p,u}$ and $\gamma_u (= c_{p,u}/c_{v,u})$ vary with temperature, equivalence ratio, and burned gas

fraction, for a gasoline-air mixture.

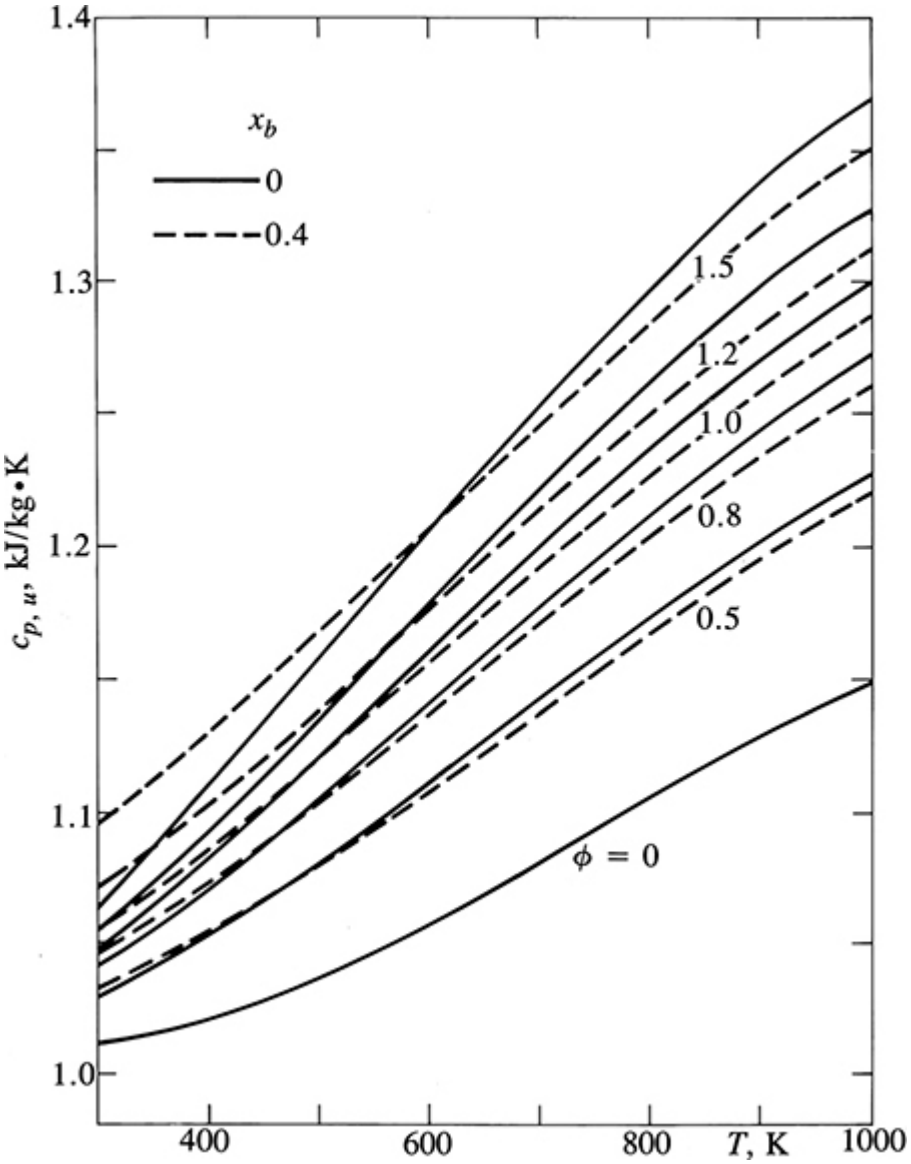


Figure 4.12 Specific heat at constant pressure of unburned gasoline, air, burned gas mixtures as function of temperature, equivalence ratio, and burned gas fraction. Units: kJ/kg mixture·K.

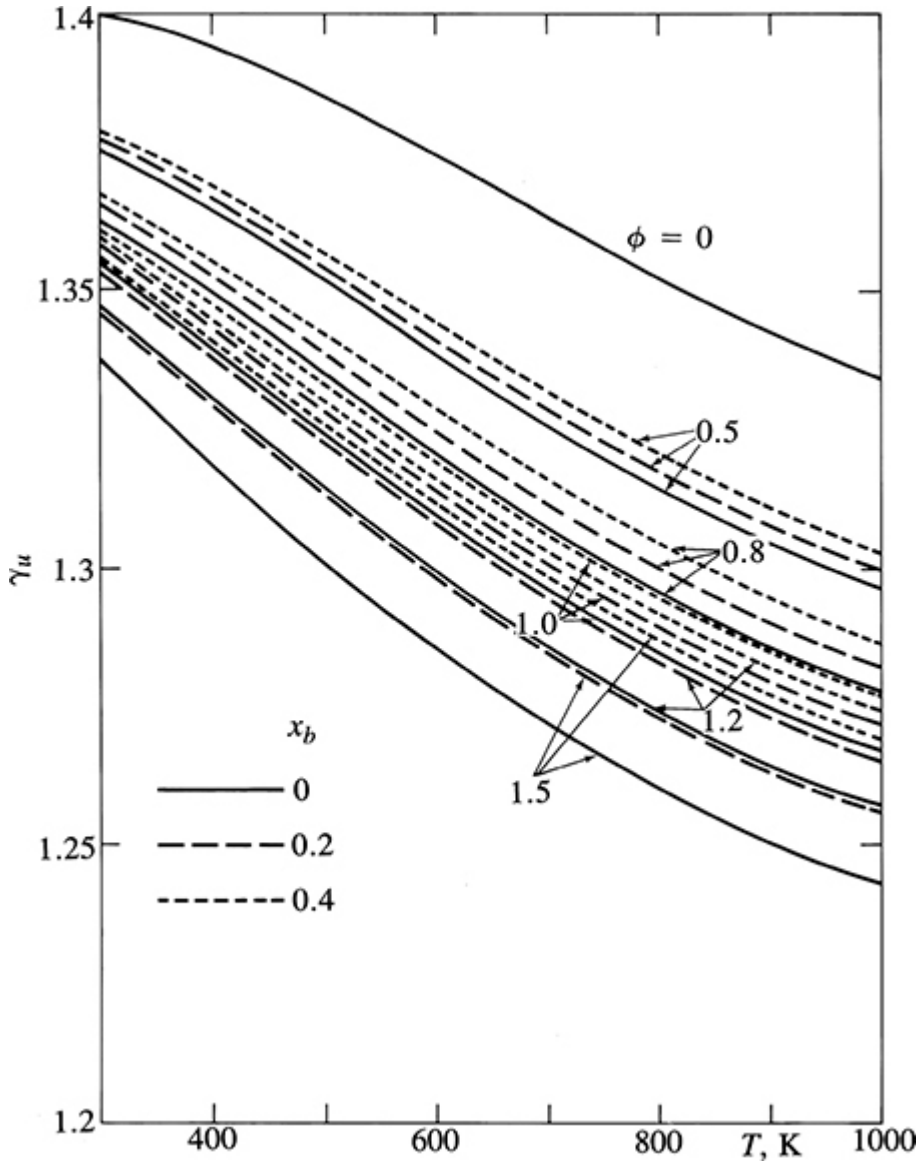


Figure 4.13 Ratio of specific heats, $\gamma_u = c_{p,u}/c_{v,u}$, of unburned gasoline, air, burned gas mixtures as function of temperature, equivalence ratio, and burned gas fraction.

4.7.2 Burned Mixtures

The most accurate approach for burned mixture property and composition calculations is to use a thermodynamic equilibrium program at temperatures above about 1700 K and a frozen composition below 1700 K. The properties of each species at high and low temperatures are given by polynomial functions such as Eqs. (4.39) to (4.41) and their coefficients in Table 4.10.

The NASA equilibrium program (see Sec. 3.7) is readily available for this purpose and is well documented.[5, 6] The following are examples of its output.

Figure 3.10 showed species concentration data for burned gases as a function of equivalence ratio at 1750, 2250, and 2750 K, at 30 atm. Figure 4.14 shows the burned gas molecular weight M_b, and Figs. 4.15 and 4.16 give cp,b and γ_b as functions of equivalence ratio at 1750, 2250, and 2750 K at 30 atm. Figures 4.17 and 4.18 show cp,b and γ_b as a function of temperature and pressure for selected equivalence ratios for mixtures lean and rich of stoichiometric.[15] For rich mixtures ($\phi > 1$), for $T > 2000$ K, cp,b and γ_b are equilibrium values. For 1200 K $\leq T \leq$ 2000 K, "frozen" composition data are shown where the gas composition is in equilibrium at the given T and p but is frozen as c_p and c_v are computed. Below about 1500 K, fixed composition data are shown corresponding to a value of 3.5 for the water–gas equilibrium constant which adequately describes exhaust gases [see Eq. (4.60) , Sec. 4.9].

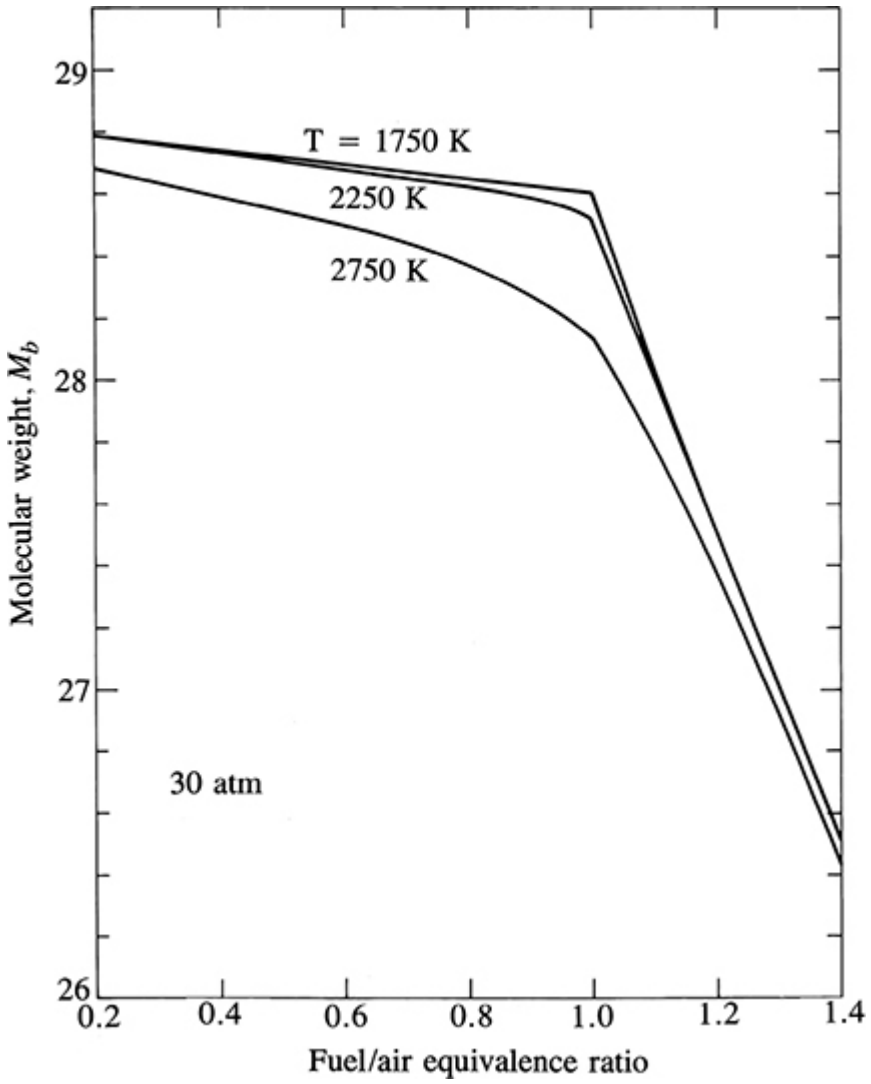


Figure 4.14 Molecular weight of equilibrium burned gases as a function of equivalence ratio at $T = 1750$, 2250, and 2750 K, and 30 atm. Fuel: isooctane.

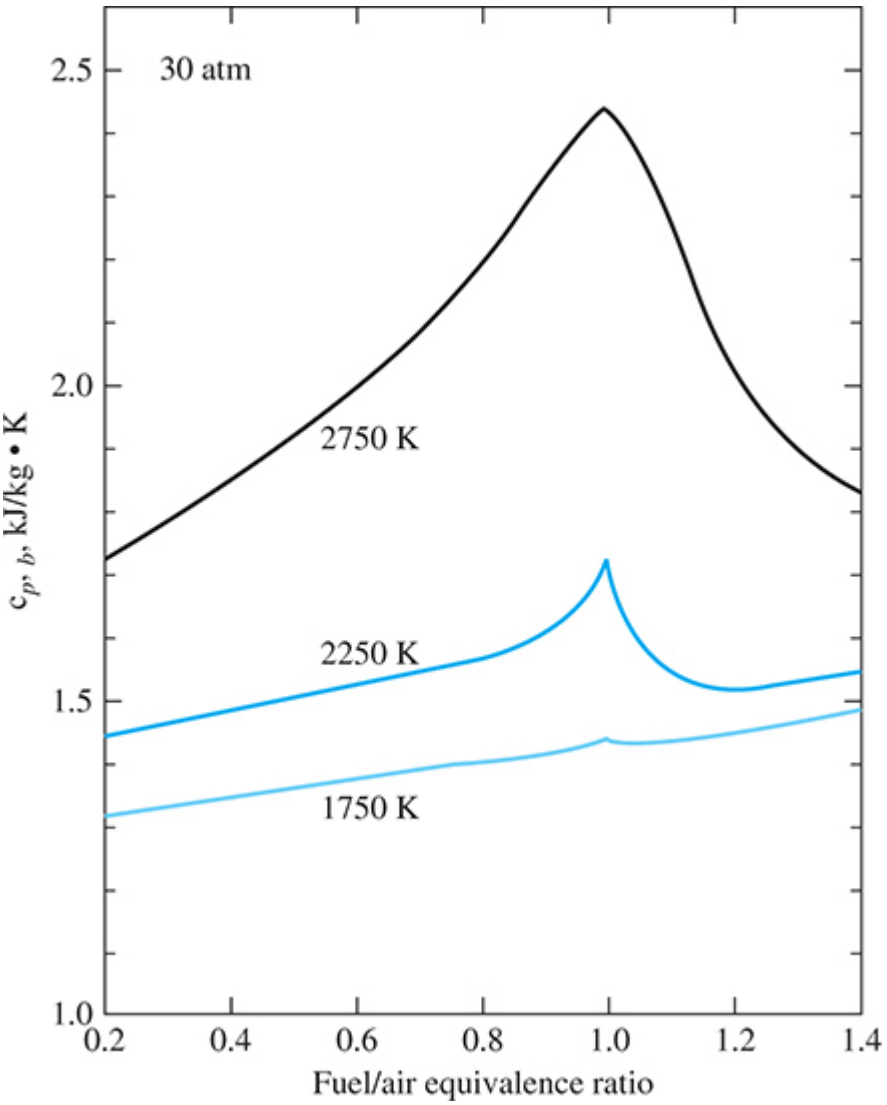


Figure 4.15 Specific heat at constant pressure of equilibrium burned gases as a function of equivalence ratio at $T = 1750$, 2250, and 2750 K, and 30 atm. Fuel: isooctane. Units: kJ/kg mixture·K.

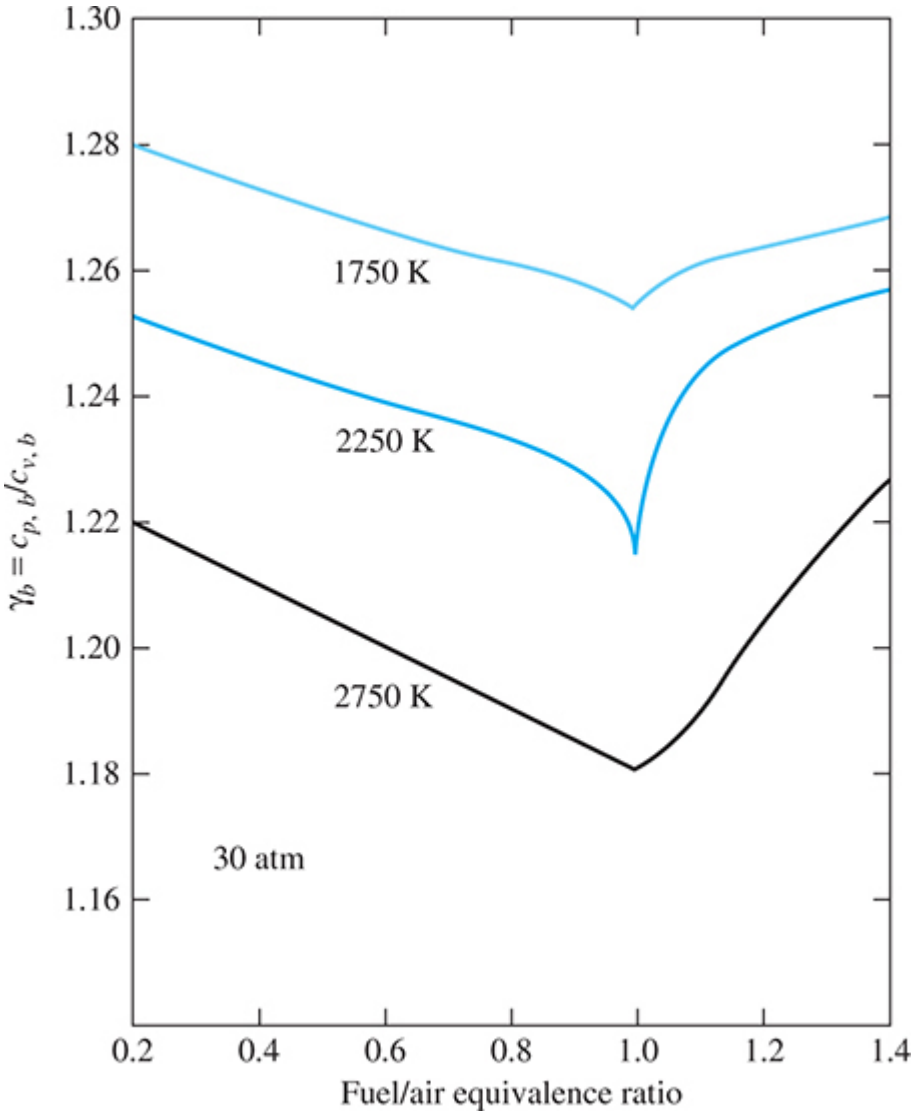
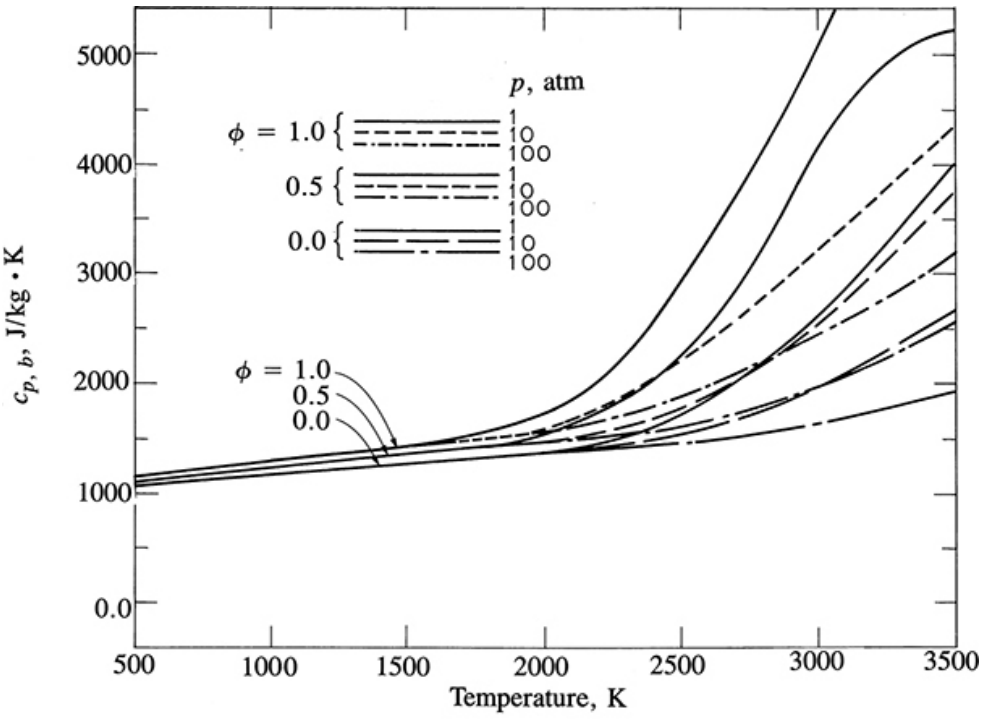
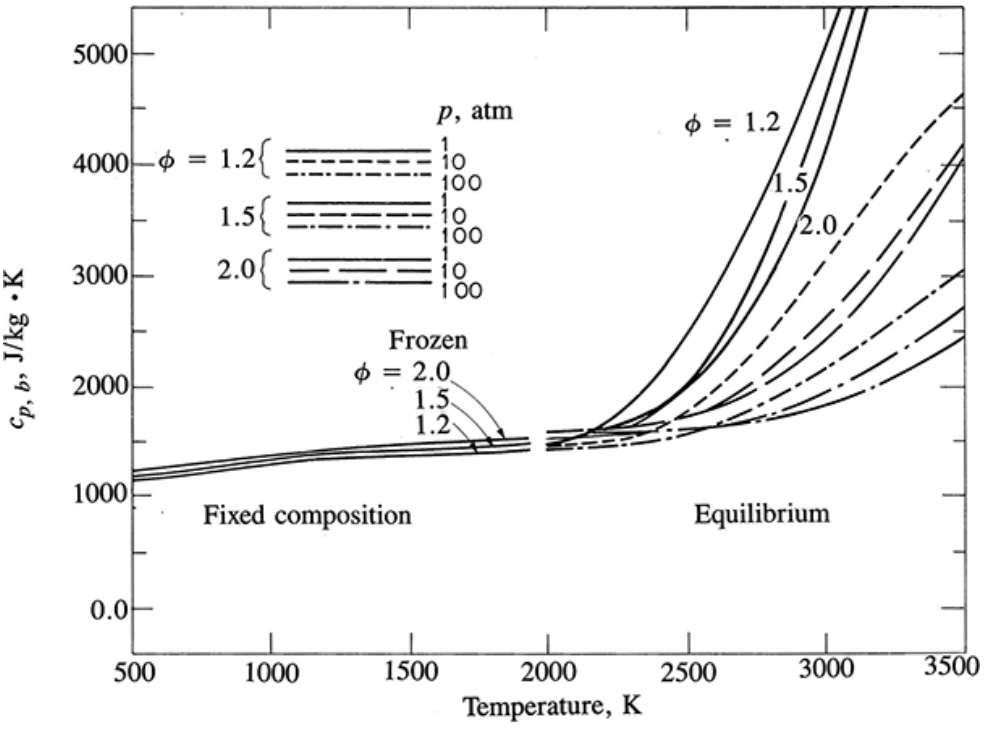


Figure 4.16 Ratio of specific heats, $g_b = cp,b/cv,b$, for equilibrium burned gases as a function of equivalence ratio at $T = 1750$, 2250, and 2750 K, and 30 atm. Fuel: isooctane.

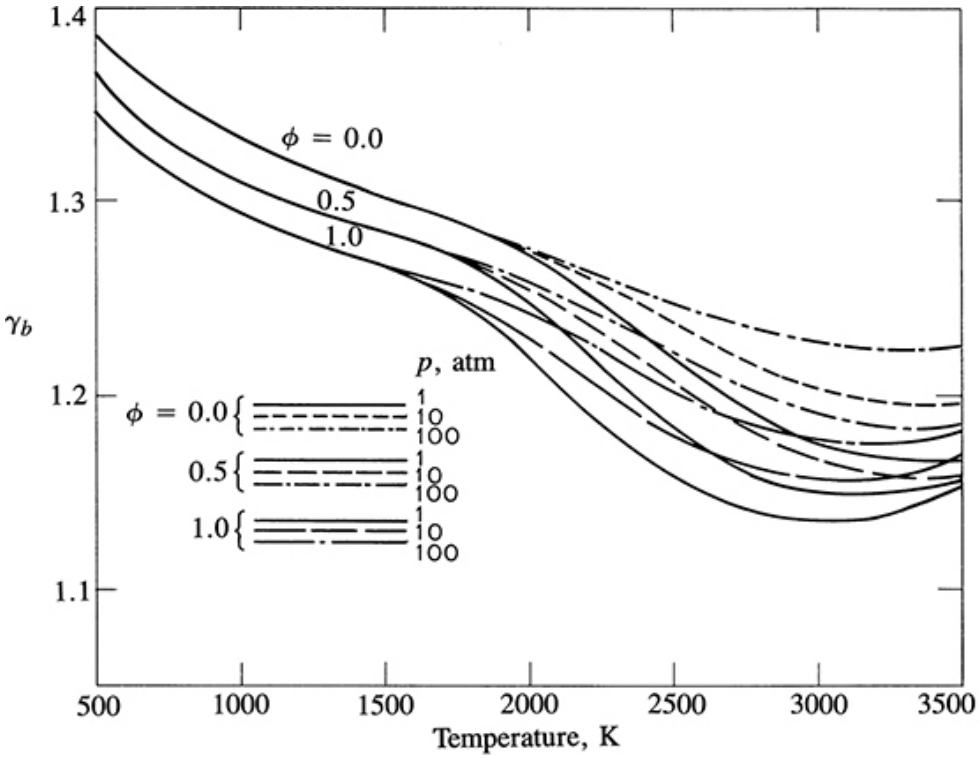

p, atm
ϕ = 1.0
0.5
0.0
1
10
100
ϕ = 1.0
0.5
0.0
$c_{p,b}$, J/kg • K
5000
4000
3000
2000
1000
0.0
500
1000
1500
2000
2500
3000
3500
Temperature, K

(*a*)

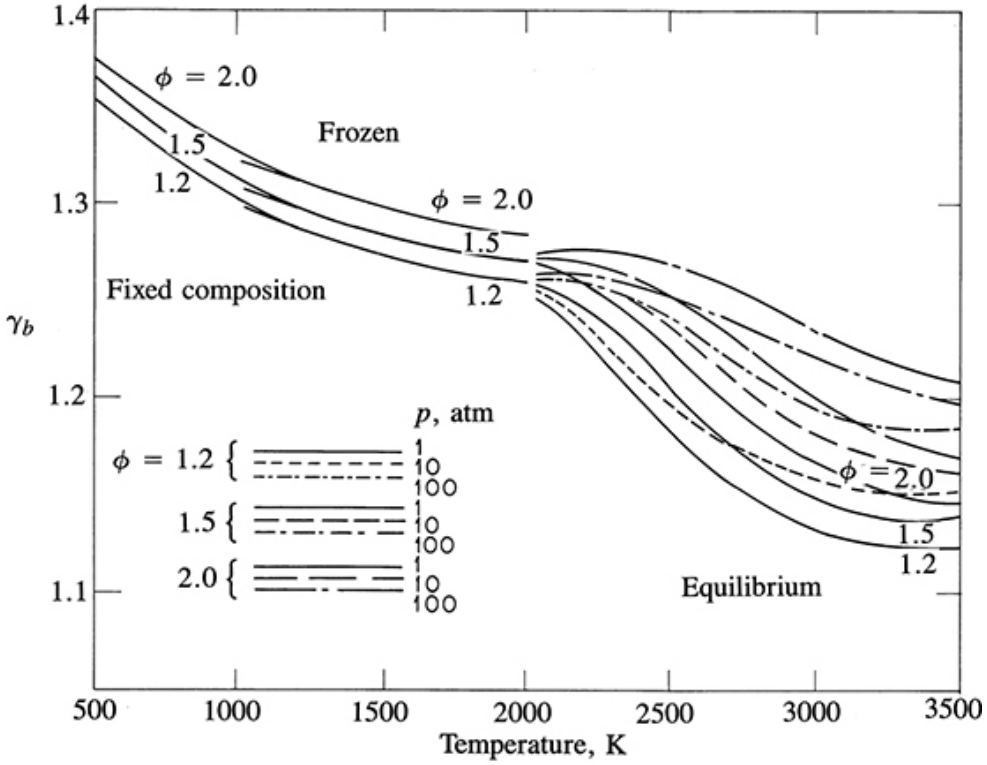

p, atm
ϕ = 1.2
1.5
2.0
1
10
100
Frozen
ϕ = 2.0
1.5
1.2
ϕ = 1.2
1.5
2.0
Fixed composition
Equilibrium
$c_{p,b}$, J/kg • K
5000
4000
3000
2000
1000
0.0
500
1000
1500
2000
2500
3000
3500
Temperature, K

(*b*)

Figure 4.17 Specific heat at constant pressure for equilibrium, frozen, and fixed composition burned gases as a function of temperature and pressure: (*a*) equivalence ratio $\phi \leq 1.0$; (*b*) equivalence ratio $\phi > 1$. Units: J/kg mixture·K. Fuel: C nH $_2$ n.


1.4
1.3
1.2
1.1
γ_b
ϕ = 0.0
0.5
1.0
p, atm
ϕ = 0.0
0.5
1.0
1
10
100
500
1000
1500
2000
2500
3000
3500
Temperature, K

(*a*)


1.4
1.3
1.2
1.1
γ_b
ϕ = 2.0
1.5
1.2
Frozen
Fixed composition
ϕ = 2.0
1.5
1.2
p, atm
ϕ = 1.2
1.5
2.0
1
10
100
ϕ = 2.0
1.5
1.2
Equilibrium
500
1000
1500
2000
2500
3000
3500
Temperature, K

(*b*)

Figure 4.18 Ratio of specific heats, $g_b = cp,b/cv,b$, for equilibrium, frozen, and fixed composition burned gases as a function of temperature.

4.8 TRANSPORT PROPERTIES

The processes by which mass, momentum, and energy are transferred from one point in a system to another are called rate processes. In internal combustion engines, examples of such processes are evaporation of liquid fuel, fuel-air mixing, friction at a gas/solid interface, and heat transfer between gas and the walls of the engine combustion chamber. In engines, most of these processes are turbulent and are therefore strongly influenced by the properties of the fluid flow. However, turbulent rate processes are usually characterized by correlations between dimensionless numbers (e.g., Reynolds, Prandtl, Nusselt numbers, etc.), which contain the fluid's transport properties of viscosity, thermal conductivity, and diffusion coefficient as well as the flow properties.

The simplest approach for computing the transport properties is based on the application of kinetic theory to a gas composed of hard-sphere molecules. By analyzing the momentum flux in a plane Couette flow, [d] it can be shown (Chapman and Cowling, [16] p. 218) that the viscosity μ of a monatomic hard-sphere gas [where $\mu = \tau/(du/dx)$, τ being the shear stress and (du/dx) the velocity gradient] is given by

$$\mu = \frac{\left[5/(16\sqrt{\pi})\right](m\tilde{k}T)^{1/2}}{d^2} \tag{4.45}$$

where m is the mass of the gas molecule, d is the molecular diameter, and $\tilde{k}$ is Boltzmann's constant, 1.381×10^{-23} J/K.

For such a gas, the viscosity varies as $T^{1/2}$, but will not vary with gas pressure or density. Measurements of viscosity show it does only vary with temperature, but generally not proportionally to $T^{1/2}$. The measured temperature dependence can only be explained with more sophisticated models for the intermolecular potential energy than that of a hard sphere. Effectively, at higher temperatures, the higher average kinetic energy of a

pair of colliding molecules requires that they approach closer to each other and experience a greater repulsive force to be deflected in the collision. As a result, the molecules appear to be smaller spheres as the temperature increases.

An expression for the thermal conductivity k of a monatomic hard-sphere gas [$k = \dot{q}/T/dx$), where $\dot{q}$ is the heat flux per unit area and dT/dx is the temperature gradient] can be derived from an analysis of the thermal equivalent of plane Couette flow (Ref. 16, p. 235):

$$k = \frac{\left[75/(64\sqrt{\pi})\right](\tilde{k}^3 T/m)^{1/2}}{d^2} \tag{4.46}$$

which has the same temperature dependence as μ. Equations (4.45) and (4.46) can be combined to give

$$k = \frac{5}{2}\mu c_v$$

since, for a monatomic gas, the specific heat at constant volume is $3\tilde{k}/(2m)$. This simple equality is in good agreement with measurements of μ and k for monatomic gases.

The above model does not take into account the vibrational and rotational energy exchange in collisions between polyatomic molecules which contribute to energy transport in gases of interest in engines. Experimental measurements of k and μ show that k is less than (5/2) μc_v for such polyatomic gases, where c_v is the sum of the translational specific heat and the specific heat due to internal degrees of freedom. It was suggested by Eucken that transport of vibrational and rotational energy was slower than that of translational energy. He proposed an empirical expression

$$k = \frac{9\gamma - 5}{4}\mu c_v$$

or

$$\mathrm{Pr} = \frac{\mu c_p}{k} = \frac{4\gamma}{9\gamma - 5} \tag{4.47}$$

where Pr is the Prandtl number, which is in good agreement with experimental data.

A similar analysis of a binary diffusion process, where one gas diffuses through another, leads to an expression for the binary diffusion coefficient D_{ij}. D_{ij} is a transport property of the gas mixture composed of species i and j, defined by Fick's law of molecular diffusion which relates the fluxes of species i and j, Γ_{xi} and Γ_{xj}, in the x direction to the concentration gradients, dn_i/dx and dn_j/dx (n is the molecular number density):

$$\Gamma_{xi} = -D_{ij}\left(\frac{dn_i}{dx}\right) \qquad \Gamma_{xj} = -D_{ij}\left(\frac{dn_j}{dx}\right)$$

The binary diffusion coefficient for a mixture of hard-sphere molecules is (Ref. 16, p. 245)

$$D_{ij} = \frac{3}{16nd^2}\left(\frac{2\tilde{k}T}{\pi m_{ij}}\right)^{1/2} \tag{4.48}$$

where m_{ij} is the reduced mass $m_i m_j/(m_i + m_j)$.

A more rigorous treatment of gas transport properties, based on more realistic intermolecular potential energy models, can be found in Hirschfelder et al., [17] who also present methods for computing the transport properties of mixtures of gases. The NASA computer program "Thermodynamic and Transport Properties of Complex Chemical Systems" [6] computes the viscosity, thermal conductivity, and Prandtl number in addition to the thermodynamic calculations described in Secs. 3.7 and 4.7 for high-temperature equilibrium and frozen gas composition mixtures. The procedures used in the NASA program to compute these transport properties are based on the techniques described in Hirschfelder et al. [17] The NASA program has been used to compute the transport properties of hydrocarbon–air combustion products. [15] These quantities are functions of temperature T, equivalence ratio ϕ, and (except for viscosity) pressure p. Approximate correlations were then fitted to the calculated data of viscosity and Prandtl number. The principal advantage of these correlations is computational speed. For Prandtl number ($\mu c_p/k$), it was found convenient to use γ, the

specific heat ratio (c_p/c_v), as an independent variable. Values of γ and c_p then permit determination of the thermal conductivity.

The viscosity of hydrocarbon–air combustion products over the temperature range 500 up to 4000 K, for pressures from 1 up to 100 atm, for $\phi = 0$ up to $\phi = 4$ is shown in Fig. 4.19. The viscosity as a function of temperature of hydrocarbon–air combustion products differs little from that of air. Therefore, a power law based on air viscosity data was used to fit the data:

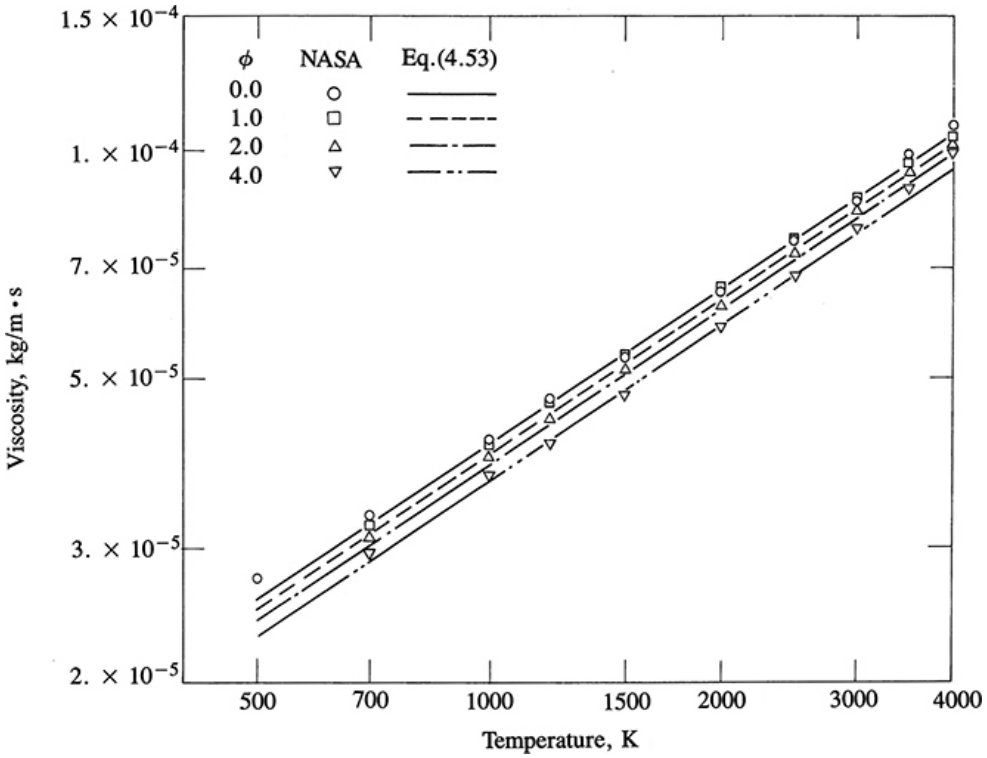


Figure 4.19 Viscosity, kg/m·s, of combustion products as a function of temperature and equivalence ratio. Equations shown are (4.49) and (4.50).

$$\mu_{air}(\text{kg/m}\cdot\text{s}) = 3.3\times10^{-7}\times T^{0.7} \tag{4.49}$$

where T is in kelvins. The viscosity of combustion products is almost independent of pressure. This correlation was corrected to include the effect of the equivalence ratio ϕ on the viscosity of hydrocarbon–air combustion products:

$$\mu_{\text{prod}} = \frac{\mu_{\text{air}}}{1 + 0.027\phi} \tag{4.50}$$

Figure 4.19 shows that the viscosity predicted using Eqs. (4.49) and (4.50) is very close to the viscosity values calculated with the NASA program. There is less than 4% error.

The Prandtl number of hydrocarbon–air combustion products has also been correlated over the above ranges of temperatures, pressures, and equivalence ratios. Since the expression for Prandtl number of a monatomic hard-sphere molecule gas is a function of γ, a second-order polynomial of γ was used to curve-fit the calculated Prandtl number data. A good fit to the data for lean combustion product mixtures was the following:

$$\text{Pr} = 0.05 + 4.2(\gamma - 1) - 6.7(\gamma - 1)^2 \qquad \phi \leq 1 \tag{4.51}$$

The values of Pr predicted with Eq. (4.51) are within 5% of the equilibrium Pr values calculated with the NASA program. For rich mixtures the following equation is a good fit to the equilibrium values of Pr using equilibrium values of γ, for temperatures greater than 2000 K:

$$\text{Pr} = \frac{0.05 + 4.2(\gamma - 1) - 6.7(\gamma - 1)^2}{1 + 0.015 \times 10^{-6}(\phi T)^2} \qquad 1 < \phi \leq 4 \tag{4.52}$$

The predicted values of Pr in this case are also close to the calculated values of Pr, with less than 10% error. Equation (4.52) is also a reasonable fit to the frozen values[e] of Pr for rich mixtures, using frozen values of γ, for the temperature range 1200 to 2000 K. As there are no data for Pr of rich mixtures at low temperatures, we suggest that where a fixed composition for the mixture is appropriate (e.g., during the exhaust process in an internal combustion engine), Eq. (4.52) can also be used with fixed composition values of γ.

The Prandtl number can be obtained from the above relations if γ is known. The thermal conductivity can be obtained from the Prandtl number if values of γ and c_p are known. Values of γ_b and $c_{p,b}$ as functions of temperature, pressure, and equivalence ratio are given in Figs. 4.15 to 4.18.

Since the fundamental relations for viscosity and thermal conductivity are complicated, various approximate methods have been proposed for

evaluating these transport properties for gas mixtures. A good approximation for the viscosity of a multicomponent gas mixture is

$$\mu_{\text{mixt}} = \sum_{i=1}^{\nu} \frac{\tilde{x}_i^2}{\tilde{x}_i^2/\mu_i + 1.385 \sum_{j=1,\, j\neq 1}^{\nu} \tilde{x}_i \tilde{x}_j (\tilde{R}T/pM_i D_{ij})} \tag{4.53}$$

where $\tilde{x}_i$ and M_i are the mole fraction and molecular weight of the ith species, μ_i is the viscosity of the ith species, μ is the number of species in the mixture, and D_{ij} is the binary diffusion coefficient for species i and j. [17]

4.9 EXHAUST GAS COMPOSITION

While the formulas for the products of combustion used in Sec. 3.4 are useful for determining unburned mixture stoichiometry, they do not correspond closely to the actual burned gas composition. At high temperatures (e.g., during combustion and the early part of the expansion stroke) the burned gas composition is close to the equilibrium composition at the local temperature, pressure, and equivalence ratio. During the expansion process, recombination reactions simplify the burned gas composition. However, late in the expansion stroke and during exhaust blowdown, the recombination reactions are unable to maintain the gases in chemical equilibrium and, in the exhaust process, the composition becomes frozen. In addition, not all the fuel which enters the engine is fully burned inside the cylinder; the combustion inefficiency even when excess air is present is a few percent (see Fig. 3.11). Also, the contents of each cylinder are not necessarily uniform in composition, and the amounts of fuel and air fed to each cylinder of a multicylinder engine are not exactly the same. For all these reasons, the composition of the engine exhaust gases cannot easily be calculated.

The composition of engine exhaust gases can readily be measured. This is done to determine engine emissions (e.g., CO, NO_x, unburned hydrocarbons, and particulates). It is also done to determine the relative proportions of fuel and air which enter the engine so that its operating equivalence ratio can be computed. In this section, typical engine exhaust gas composition will be

reviewed, and techniques for calculating the fuel/air equivalence ratio from exhaust gas composition will be given.

4.9.1 Species Concentration Data

Standard instrumentation for measuring the concentrations of the major exhaust gas species has been developed. [18] Normally a small fraction of the engine exhaust gas stream is drawn off into a sample line. Part of this sample is fed directly to the instrument used for unburned hydrocarbon analysis, a flame ionization detector (FID). The hydrocarbons present in the exhaust gas sample are burned in a small hydrogen–air flame, producing ions in an amount proportional to the number of carbon atoms burned. The FID is effectively a carbon atom counter. It is calibrated with sample gases containing known amounts of hydrocarbons. Unburned hydrocarbon concentrations are normally expressed as a mole fraction or volume fraction in parts per million (ppm) as C_1. Sometimes results are expressed as ppm propane (C_3H_8) or ppm hexane (C_6H_{14}); to convert these to ppm C_1 multiply by 3 or 6, respectively. Measurements of unburned hydrocarbons are sometimes made with a nondispersive infrared (NDIR) analyzer, where the infrared absorption by the hydrocarbons in a sample cell was used to determine their concentration. Values of HC concentrations in engine exhaust gases measured by an FID are about two times the equivalent values measured by an NDIR analyzer (on the same carbon number basis, e.g., C_1). NDIR-obtained concentrations are usually multiplied by 2 to obtain an estimate of actual HC concentrations. Substantial concentrations of oxygen in the exhaust gas can affect the FID measurements. Analysis of *unburned* fuel-air mixtures should be done with special care. To prevent condensation of hydrocarbons in the sample line (especially important in diesel exhaust gas), the sample line is often heated.

NDIR analyzers are used for CO_2 and CO concentration measurements. Infrared absorption in a sample cell containing exhaust gas is compared to absorption in a reference cell. The detector contains the gas being measured in two compartments separated by a diaphragm. Radiation not absorbed in the sample cell is absorbed by the gas in the detector on one side of the diaphragm. Radiation not absorbed in the reference cell is absorbed by the gas in the other half of the detector. Different amounts of absorption in the

two halves of the detector result in a pressure difference being built up which is measured in terms of diaphragm distention. NDIR detectors are calibrated with sample gases of known composition. Since water vapor IR absorption overlaps CO_2 and CO absorption bands, the exhaust gas sample is dried with an ice bath and chemical dryer before it enters the NDIR instrument.

Oxygen concentrations are usually measured with paramagnetic analyzers. Oxides of nitrogen, either the amount of nitric oxide (NO) or total oxides of nitrogen ($NO + NO_2$, NO_x), are measured with a chemiluminescent analyzer. The NO in the exhaust gas sample stream is reacted with ozone in a flow reactor. The reaction produces electronically excited NO_2 molecules which emit radiation as they decay to the ground state. The amount of radiation is measured with a photomultiplier and is proportional to the amount of NO. The instrument can also convert any NO_2 in the sample stream to NO by decomposition in a heated stainless steel tube so that the total NO_x ($NO + NO_2$) concentration can be determined. Gas chromatography can be used to determine all the inorganic species (N_2, CO_2, O_2, CO, H_2) or can be used to measure the individual hydrocarbon compounds in the total unburned hydrocarbon mixture. Particulate emissions are measured by filtering the particles from the exhaust gas stream onto a previously weighed filter, drying the filter plus particulate, and reweighing.

Spark-Ignition Engine Data

Dry exhaust gas composition data, as a function of the fuel/air equivalence ratio, for several different multi- and single-cylinder automotive spark-ignition engines over a range of engine speeds and loads are shown in Fig. 4.20. The fuel compositions (gasolines and isooctane) had H/C ratios ranging from 2.0 to 2.25. Exhaust gas composition is substantially different on the lean and the rich side of the stoichiometric air/fuel or fuel/air ratios; thus, the fuel/air equivalence ratio ϕ (or its inverse, the relative air/fuel ratio λ) is the appropriate correlating parameter. On the lean side of stoichiometric, as ϕ decreases, CO_2 concentrations fall, oxygen concentrations increase, and CO levels are low but not zero ($\sim$ 0.2%). On the rich side of stoichiometric, CO and H_2 concentrations rise steadily as ϕ increases and CO_2 concentrations fall. O_2 levels are low ($\sim$ 0.2 to 0.3%) but are not zero. At stoichiometric

operation, there is typically half a percent O_2 and three-quarters of a percent CO.

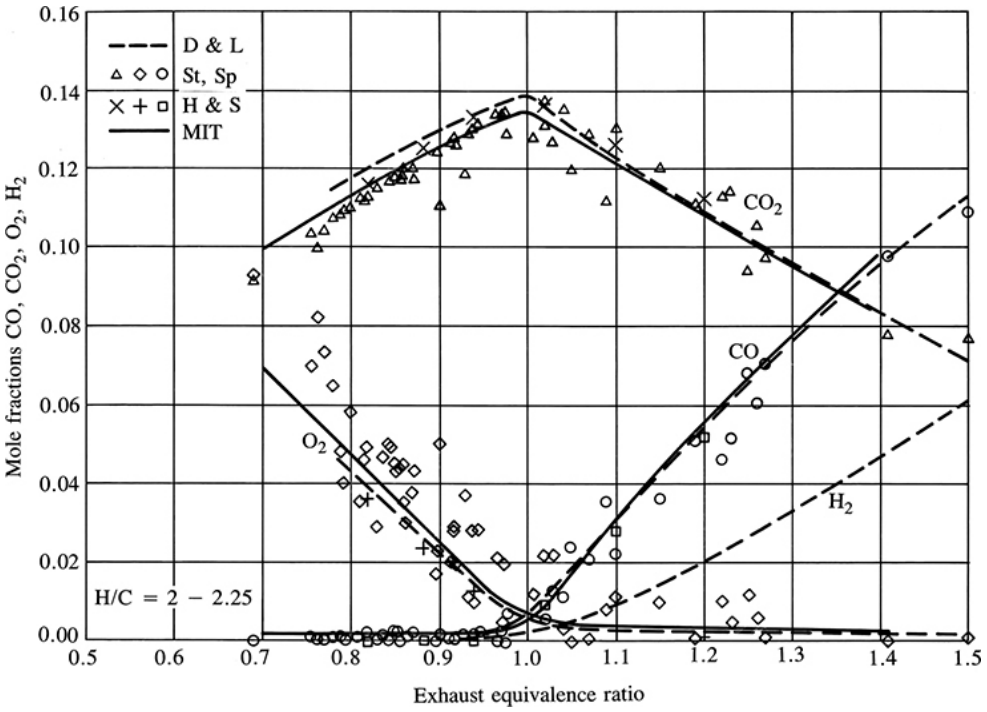


Figure 4.20 Spark-ignition engine exhaust gas composition data in mole fractions as a function of fuel/air equivalence ratio. Fuels: gasoline and isooctane, H/C 2 to 2.25. (*From D'Alleva and Lovell*, [19] *Stivender*, [20] *Harrington and Shishu*, [21] *Spindt*, [22] *and data from the author's laboratory at MIT*.)

The carbon:hydrogen ratio of the fuel has only a modest effect on the magnitude of the species concentrations shown. Measurements with a wide range of liquid fuels show that CO concentrations depend only on the equivalence ratio or relative fuel/air ratio (see Fig. 11.20). [21] A comparison of exhaust CO concentrations with gasoline, propane (C_3H_8), and natural gas (predominantly methane, CH_4) shows that only with the high H/C ratio of methane, and then only for CO ≥ 4%, is fuel composition significant. [23] The values of CO_2 concentration at a given ϕ are slightly affected by the fuel H/C ratio. For example, for stoichiometric mixtures with 0.5% O_2 and 0.75% CO, as the H/C ratio decreases CO_2 concentrations increase from 13.7% for

isooctane (H/C = 2.25) to 14.2 to 14.5% for typical gasolines (H/C in range 2 to 1.8), to 16 for toluene (H/C = 1.14). [24]

Unburned hydrocarbon exhaust concentrations vary significantly with engine design and operating conditions. Spark-ignition engine exhaust levels in a modern low-emission engine are typically of the order of 2000 ppm C_1 with liquid hydrocarbon fuels, and about half that level with natural gas and propane fuels.

Hydrogen concentrations in engine exhaust are not routinely measured. However, when the mixture is oxygen deficient (i.e., fuel rich), hydrogen is present with CO as an incomplete combustion product. Figure 4.21 summarizes much of the available data on H_2 concentrations plotted as a function of CO.[25] Approximations for estimating hydrogen concentrations in spark-ignition engine exhaust gases have been proposed: for example, the molar concentration of H_2 is half the molar concentration of CO. This is a reasonable approximation to the data in Fig. 4.21 for $4\% \lesssim \tilde{x}_{CO} \lesssim 10\%$. The curve shown in the figure corresponds to

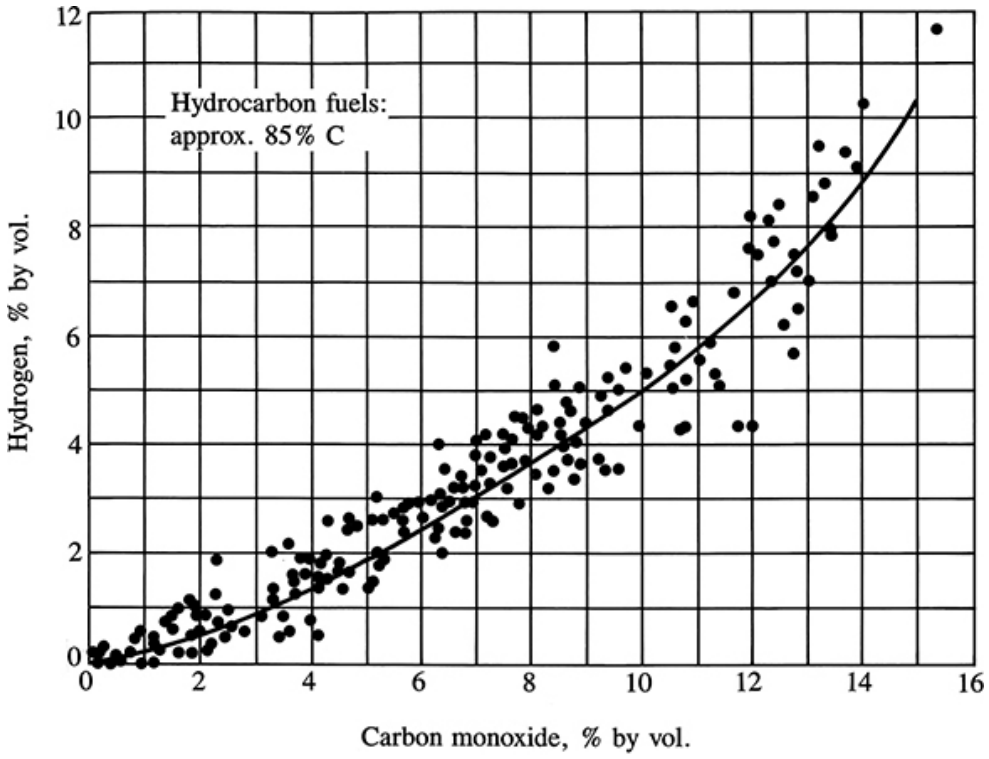


Figure 4.21 Hydrogen concentration in spark-ignition engine exhaust as a function of carbon monoxide concentration. Units: percent by volume. [25]

$$\tilde{x}_{H_2} = 0.3\,\tilde{x}_{CO} + 0.02\,\tilde{x}_{CO}^{\,2} + 3\times10^{-5}\,\tilde{x}_{CO}^{\,3} \tag{4.54}$$

Diesel Exhaust Data

Since diesels normally operate significantly lean of stoichiometric ($\phi \lesssim 0.8$) and the diesel combustion process is essentially complete (combustion inefficiency is less than 2%), their exhaust gas composition is straight forward. Figure 4.22 shows that O_2 and CO_2 concentrations vary linearly with the fuel/air equivalence ratio over the normal operating range. Diesel emissions of CO and unburned HC are low.

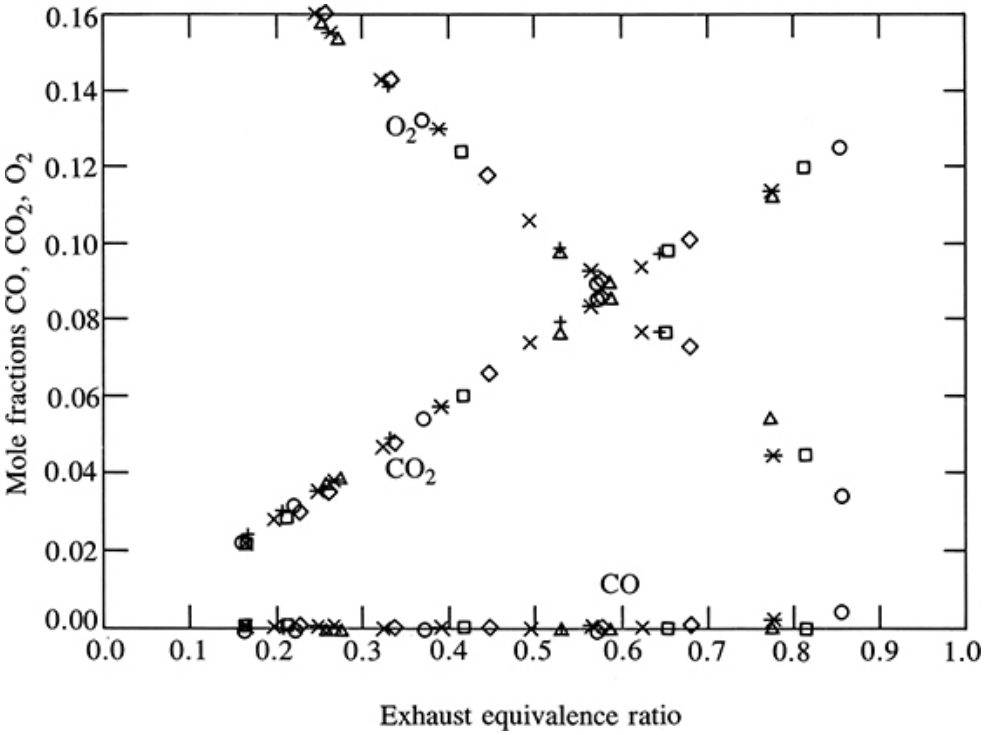


Figure 4.22 Exhaust gas composition from several diesel engines in mole fractions on a dry basis as a function of fuel/air equivalence ratio.[26]

4.9.2 Equivalence Ratio Determination from Exhaust Gas Constituents

Exhaust gas composition depends on the relative proportions of fuel and air fed to the engine, fuel composition, and completeness of combustion. These relationships can be used to determine the operating fuel/air equivalence

ratio of an engine from a knowledge of its exhaust gas composition. A general formula for the composition of fuel can be represented as $C_nH_mO_r$. For conventional petroleum-based fuels, oxygen will be absent; for fuels containing alcohols, oxygen will be present. The overall combustion reaction can be written as

$$\text{Fuel} + \text{oxidizer} \rightarrow \text{products}$$

The fuel is $C_nH_mO_r$; the oxidizer is air ($O_2 + 3.773N_2$). The products are CO_2, H_2O, CO, H_2, O_2, NO_x, N_2, unburned hydrocarbons (unburned fuel and products of partial fuel reaction), and soot particles (which are mainly solid carbon). The amount of solid carbon present is usually sufficiently small (≤0.5% of the fuel mass) for it to be omitted from the analysis. The overall combustion reaction can be written explicitly as

$$\begin{aligned} C_nH_mO_r + \frac{n_{O_2}}{\phi}(O_2 + 3.773N_2) = n_P\big(&\tilde{x}_{C_aH_b}C_aH_b + \tilde{x}_{CO}CO + \tilde{x}_{CO_2}CO_2 \\ &+ \tilde{x}_{O_2}O_2 + \tilde{x}_{N_2}N_2 + \tilde{x}_{NO}NO \\ &+ \tilde{x}_{NO_2}NO_2 + \tilde{x}_{H_2O}H_2O + \tilde{x}_{H_2}H_2\big) \end{aligned} \tag{4.55}$$

where ϕ is the measured fuel/air equivalence ratio [$(F/A)_{\text{actual}}/(F/A)_{\text{stoichiometric}}$], n_{O_2} is the number of O_2 molecules required for complete combustion ($n + m/4 - r/2$), n_P is the total number of moles of exhaust products, and $\tilde{x}_i$ is the mole fraction of the ith component.

There are several methods for using Eq. (4.55) to determine ϕ, the equivalence ratio, depending on the amount of information available. Normally CO_2, CO, O_2, NO_x concentrations as mole fractions and unburned hydrocarbon (as mole fraction or ppm C_1, i.e., $\tilde{x}_{CH_{b/a}}$) are measured. The concentrations of the inorganic gases are usually measured *dry* (i.e., with H_2O removed) or *partially dry*. Unburned hydrocarbons may be measured *wet* or *dry* or *partially dry*. NO_x is mainly nitric oxide (NO); its concentration is usually sufficiently low (<0.5%) for its effect on equivalence ratio determination to be negligibly small. Thus, in Eq. (4.55) there are seven unknowns, which are: $\phi, \tilde{x}_{H_2}, \tilde{x}_{H_2O}, \tilde{x}_{N_2}, n_p, a, b$. (There will be additional unknowns if the measurements listed above are incomplete.)

To solve for these unknowns we need seven additional equations. We can obtain five equations using an atomic balance for each element and the definition of mole fraction, as follows:
Carbon balance:

$$n = n_p(a\tilde{x}_{C_aH_b} + \tilde{x}_{CO} + \tilde{x}_{CO_2}) \tag{4.56}$$

Hydrogen balance:

$$m = n_p(b\tilde{x}_{C_aH_b} + 2\tilde{x}_{H_2O} + 2\tilde{x}_{H_2}) \tag{4.57}$$

Oxygen balance:

$$r + \frac{2n_{O_2}}{\phi} = n_p(\tilde{x}_{CO} + 2\tilde{x}_{CO_2} + \tilde{x}_{NO} + 2\tilde{x}_{O_2} + \tilde{x}_{H_2O}) \tag{4.58}$$

Nitrogen balance:

$$\frac{7.546n_{O_2}}{\phi} = n_p(2\tilde{x}_{N_2} + \tilde{x}_{NO}) \tag{4.59}$$

The mole fractions add up to 1:

$$\tilde{x}_{C_aH_b} + \tilde{x}_{CO} + \tilde{x}_{H_2} + \tilde{x}_{H_2O} + \tilde{x}_{N_2} + \tilde{x}_{NO} + \tilde{x}_{CO_2} + \tilde{x}_{O_2} = 1 \tag{4.60}$$

An additional assumption is made, based on available exhaust gas composition data, that CO_2, CO, H_2O, and H_2 concentrations are related by

$$\frac{\tilde{x}_{CO}\tilde{x}_{H_2O}}{\tilde{x}_{CO_2}\tilde{x}_{H_2}} = K \tag{4.61}$$

where K is a constant.[f] Values of 3.8 [22, 23] and 3.5 [25] are commonly used for K. The difference between these values has little effect on the computed magnitude of ϕ. To complete the analysis, various assumptions are made concerning the composition and relative importance of the unburned hydrocarbons. The most common approaches are summarized below.

Oxygen Balance Air/Fuel and Equivalence Ratios

For fuels composed of carbon and hydrogen only, when all species are measured with the *same background moisture* (wet, dry, or partially dry), the following expression based on the ratio of measured and computed oxygen-containing species to measured carbon-containing species gives the air/fuel ratio. It has been assumed that the unburned hydrocarbons have the same C/H ratio as the fuel: [27]

$$\left(\frac{A}{F}\right) = 4.773\left(\frac{M_{\text{air}}}{M_f}\right)\frac{(CO_2)+(CO)/2+(H_2O)/2+(NO)/2+(NO_2)+(O_2)}{(HC)+(CO)+(CO_2)} \tag{4.62}$$

where ( ) denotes molar concentration (all with the same background moisture) in percent; $M_{\text{air}} = 28.96$, $M_f = 12.01 + 1.008\,y$ where y is the H/C ratio of the fuel, (HC) is molar percent unburned hydrocarbons as C_1, and

$$(H_2O) = 0.5y\frac{(CO_2)+(CO)}{(CO)/[K(CO_2)]+1} \tag{4.63}$$

Since nitrogen oxides collectively comprise less than 0.5% of the exhaust mixture, their concentrations can be omitted with negligible error.

The fuel/air equivalence ratio ϕ is obtained from the ratio of the stoichiometric air/fuel ratio [Eq. (3.6)] and Eq. (4.62) above.

Carbon Balance Air/Fuel and Equivalence Ratios

When oxygen analysis is not available, for fuels composed of carbon and hydrogen only, a carbon balance air/fuel ratio may be employed: [20, 27]

$$\frac{A}{F} = \frac{M_{\text{air}}}{M_f}\left[\frac{100+(HC)-(CO)/2+3(H_2O)/2-(H_2O)_a}{(HC)+(CO)+(CO_2)} - \frac{y}{2}\right] \tag{4.64}$$

The symbols are as defined above. (H_2O) is the molar percent water in the combustion products defined by Eq. (4.63) and $(H_2O)_a$ is the molar percent water vapor at the analyzers.

This carbon balance (*A/F*) is sensitive to moisture concentration at the analyzers. The use of ice bath exhaust sample chillers generally reduces the $(H_2O)_a$ term to less than 1% and little accuracy is then lost by neglecting it.

For completely "wet" analysis (uncondensed), $(H_2O)_a = (H_2O)$, and Eq. (4.64) is accurate. For partially dry exhaust gas analysis, knowledge of the dew point of the mixture will provide the $(H_2O)_a$ term by reference to steam tables.

The fuel/air equivalence ratio ϕ is obtained from the ratio of the stoichiometric air/fuel ratio [Eq. (3.6)] and Eq. (4.64) above.

Equivalence Ratio Based on Wet HC and Dry Inorganic Gas Analysis

Engine exhaust gas composition is often determined by analyzing a fully dried sample stream for CO_2, CO, O_2, and NO_x, and a fully wet (uncondensed) stream with an FID for unburned hydrocarbons. Equations (4.62) and (4.64) are not applicable under these circumstances. The following equations define the exhaust gas composition and equivalence ratio under these conditions. The notation $\tilde{x}_i$ denotes the wet mole fraction of species i and $\tilde{x}_i^*$ denotes the dry mole fraction of species i. Equations (4.55) to (4.60), with Eq. (4.61) to relate CO_2, CO, H_2O, and H_2 concentrations and the assumption that $b/a = m/n$, were used to derive these results. The equations apply for a fuel of composition $C_mH_nO_r$.

The fuel/air equivalence ratio is given by

$$\phi = \frac{2n_{O_2}}{n_P \tilde{x}_{H_2O} + n_P\left(1-\tilde{x}_{H_2O}\right)\left(\tilde{x}^*_{CO} + 2\tilde{x}^*_{CO_2} + 2\tilde{x}^*_{O_2} + \tilde{x}^*_{NO} + 2\tilde{x}^*_{NO_2}\right) - r} \tag{4.65}$$

where the wet and dry mole fractions are related by

$$\tilde{x}_i = \left(1-\tilde{x}_{H_2O}\right)\tilde{x}_i^*$$

and

$$n_P = \frac{n}{\tilde{x}_{CH_{b/a}} + \left(1-\tilde{x}_{H_2O}\right)\left(\tilde{x}^*_{CO} + \tilde{x}^*_{CO_2}\right)}$$

$$\tilde{x}_{H_2O} = \frac{m}{2n}\frac{\tilde{x}^{*}_{CO} + \tilde{x}^{*}_{CO_2}}{\left[1 + \tilde{x}^{*}_{CO}/\left(K\tilde{x}^{*}_{CO_2}\right) + (m/2n)\left(\tilde{x}^{*}_{CO} + \tilde{x}^{*}_{CO_2}\right)\right]}$$

$$\tilde{x}_{H_2} = \frac{\tilde{x}_{H_2O}\,\tilde{x}^{*}_{CO}}{K\tilde{x}^{*}_{CO_2}} \qquad (4.66)$$

$$\tilde{x}_{N_2} = \frac{3.773 n_{O_2}}{\phi n_P} - \left(1 - \tilde{x}_{H_2O}\right)\frac{\left(\tilde{x}^{*}_{NO} + \tilde{x}^{*}_{NO_2}\right)}{2}$$

Note that $\tilde{x}_{CH_{b/a}}$ is the measured (wet) HC concentration as a mole fraction C_1 (ppm $C_1 \times 10^{-6}$): $\tilde{x}_{CH_{b/a}} = a\tilde{x}_{C_aH_b}$. Figure 4.23 shows wet exhaust gas concentrations, based on the MIT measured dry concentrations of CO_2, CO, O_2 shown in Fig. 4.20, and wet HC concentration, as well as Eqs. (4.66) .

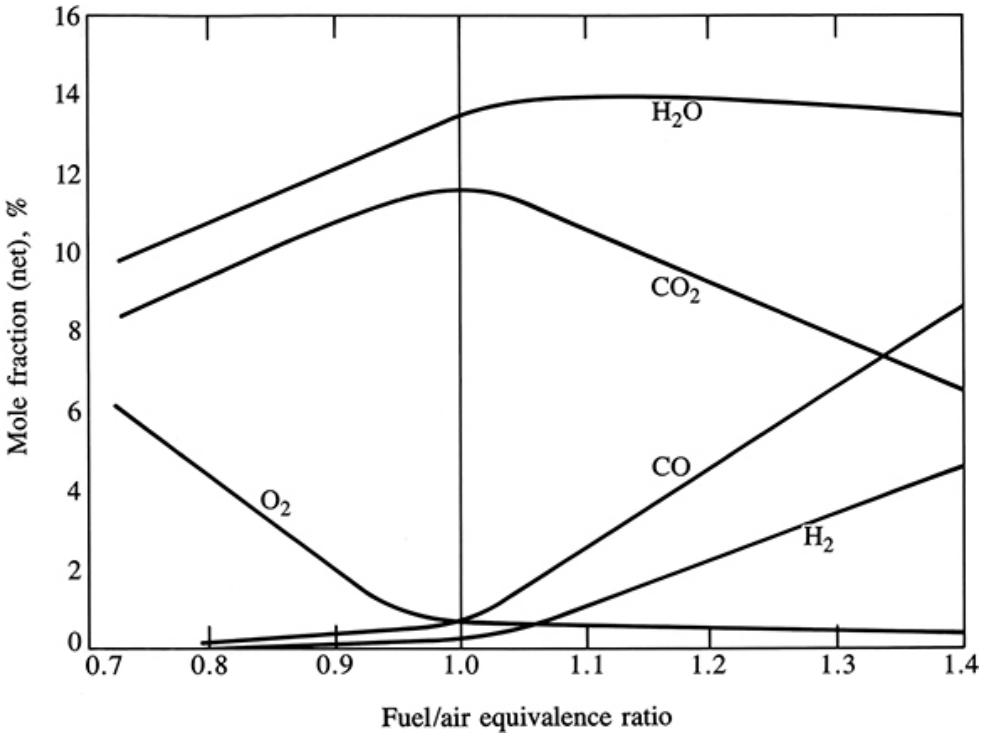


Figure 4.23 Wet exhaust gas species concentrations as a function of fuel/air equivalence ratio based on the dry exhaust gas composition data in Fig. 4.20 and Eqs. (4.66) .

For lean mixtures, varying the value of K between 1.5 and 5.5 has a negligible effect on the value of ϕ computed from Eq. (4.65) . For stoichiometric mixtures, varying K from 2.5 to 4.5 varied the computed ϕ by 2 to 3%. For $\phi \approx 1.2$, varying K from 2.5 to 4.5 varied the computed ϕ by 3 to 4%. The error in ϕ involved in omitting NO_x is 0.2% for an NO_x level of 1000 ppm, increasing to 1% for an NO_x level of 5000 ppm. The sensitivity of the computed ϕ to errors in the measurements of CO_2, CO, and O_2 is modest within the normal range of ϕ used. A 2% error in CO_2 or CO or O_2 at $\phi \approx 1$ gives about a 0.1% error in computed ϕ. For leaner and richer mixtures, the error in ϕ increases for errors in measured CO_2 concentration, and CO ($\phi > 1$) and O_2 ($\phi < 1$) concentrations, but is still significantly less than the measurement error in fuel and air flow.

4.9.3 Effects of Fuel/Air Ratio Nonuniformity

Neither the masses of air inducted into the different cylinders of a multicylinder engine per cycle nor the masses of fuel which enter the different cylinders per cycle are exactly equal. In addition, mixing of fuel and air within each cylinder is not necessarily completely uniform. Thus the exhaust gas composition may correspond to a distribution in the fuel/air ratio in the unburned mixture about the mean value. For example, if the mean fuel/air ratio is stoichiometric, extra oxygen will be contributed by any cylinders operating lean of the average and extra carbon monoxide by any cylinders rich of the average, so that the exhaust gas will have higher levels of O_2 and CO (and a lower level of CO_2) relative to an engine operating with identical fuel/air ratios in each cylinder.

Eltinge has developed a method for defining this nonuniformity in the fuel/air ratio distribution for spark-ignition engines which operate close to stoichiometric. [24] A function $f(x)$ for the fuel/air ratio distribution ($x = F/A$) was assumed, with standard deviation S_x. For each value of x (i.e., F/A) complete utilization of the available oxygen was assumed (i.e., no exhaust HC) and the CO_2, H_2O, CO, H_2 concentrations were related by Eq. (4.61) (with $K = 3.5$). The concentrations of each species for each (F/A) were weighted by the distribution function $f(x)$ and summed to produce the average exhaust concentration. (A correction to allow for the presence of unburned

HC in the exhaust was also developed.) Figure 4.24 shows one set of results, for a fuel H/C ratio of 1.8 (typical of gasoline), for a normal distribution in the fuel/air ratio. For each mean (*F/A*) and maldistribution parameter S_x (the standard deviation of the *F/A* distribution) the corresponding dry concentrations of CO_2, CO, and O_2 are shown. This type of information can be used to define the fuel-air mixture nonuniformities in spark-ignition engines operating relatively close to stoichiometric. For diesel engines the variations of major exhaust gas species concentrations with fuel/air equivalence ratio are linear, so the effects of any nonuniformities are not apparent in this manner (see Fig. 4.22).

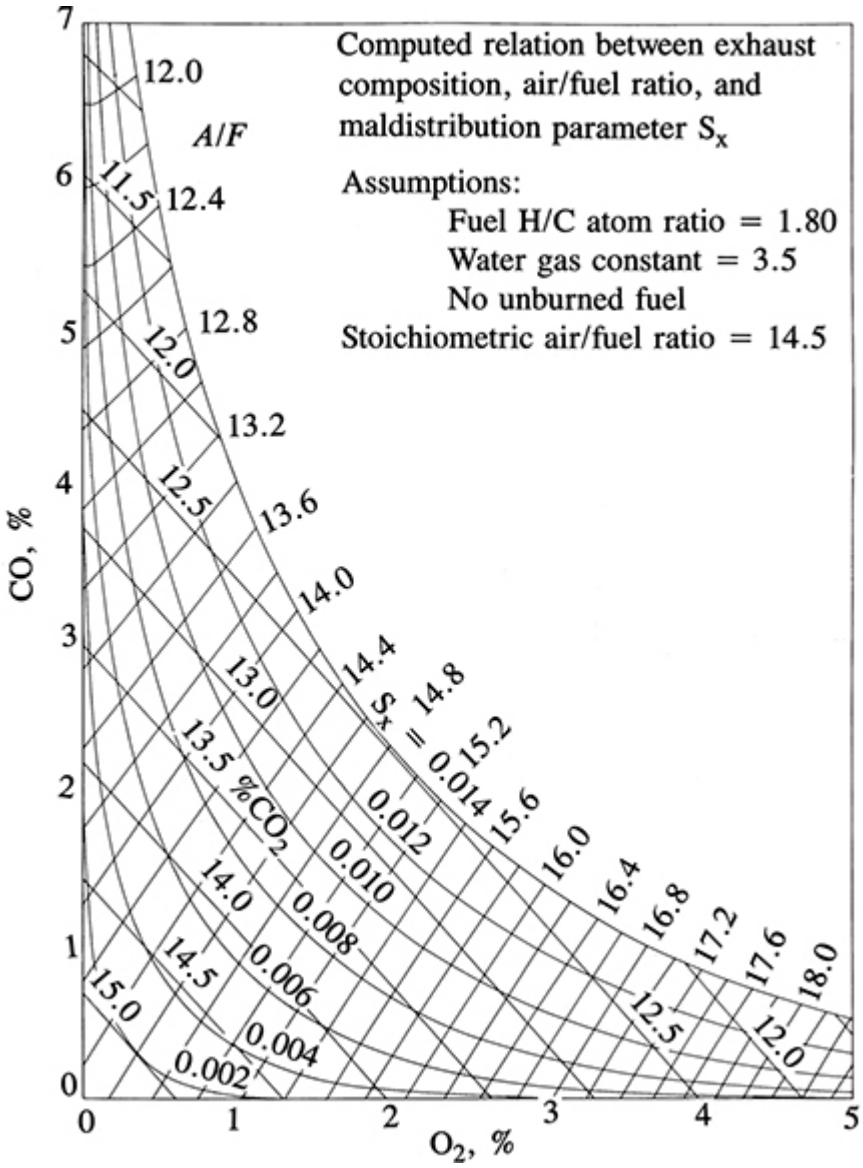


Figure 4.24 Computed relationship between dry exhaust gas composition (CO_2, CO, and O_2), air/fuel ratio, and maldistribution parameter S_x. Fuel: ($CH_{1.8}$) n. The correction to be added to the burned gas (F/A) which allows for the measured unburned hydrocarbon concentration is $4.7 \times 10^{-7} \times$ HC (ppm C_1). (*From Eltinge.* [24])

4.9.4 Combustion Inefficiency

Internal combustion engine exhaust contains combustible species: CO, H_2, unburned hydrocarbons, and particulates. When their concentrations are known, the combustion efficiency h_c given by Eq. (3.27) can be calculated. The chemical energy carried out of the engine in these combustibles represents the combustion inefficiency, $1 - \eta_c$:

$$1-\eta_c=\frac{\sum_i x_i Q_{\mathrm{HV}}}{\left[\dot{m}_f/\left(\dot{m}_a+\dot{m}_f\right)\right]Q_{\mathrm{HV}_f}} \tag{4.67}$$

where the x_i are the mass fractions of CO, H_2, HC, and particulates, respectively; the Q_{HV_i} are the lower heating values of these species; and the subscripts f and a denote fuel and air. The heating values for CO (10.1 MJ/kg) and H_2 (120 MJ/kg) are well defined. The composition of the unburned HC is not usually known. However, the heating values of hydrocarbons are closely comparable, so the fuel heating value (typically 42 to 44 MJ/kg) is used. The particulates (only present in diesels) are soot with some adsorbed hydrocarbons; usually the mass fraction is low enough for their contribution to be small, and a heating value for solid carbon (32.8 MJ/kg) can be used. Combustion efficiency data as a function of equivalence ratio have already been presented in Fig. 3.9.

PROBLEMS

4.1 (a) Calculate the low-temperature burned gas composition resulting from the combustion of 7 g/s air with 0.48 g/s ethane (C_2H_6). Assume $K = 3.5$ in

Eq. (4.6) .

(b) Calculate the low-temperature burned gas composition for the combustion of 7 g/s air with 0.48 g/s ethanol (C_2H_6O). Assume $K = 3.5$ in Eq. (4.6) . (Note the difference in relative air/fuel ratio and burned gas composition due to this difference in fuel.)

4.2 To evaluate the accuracy of the simple analytic ideal gas model, use the results of Example 4.1 and Eqs. (4.23) (constant-volume adiabatic combustion) and (4.24) (constant-pressure adiabatic combustion) to calculate T_b for a stoichiometric isooctane-air mixture. Compare this result with that obtained using Figs. 4.3 and 4.8. Assume the following unburned mixture conditions: $T = 700$ K, $v = 0.125$ m^3/kg air, $p = 15$ bar, and $x_b = 0.1$.

4.3 Show how, for an ideal gas with fixed composition, the molecular weight M is related to the specific heats c_p and c_v, and $\tilde{R}$. Use Figs. 4.15 and 4.16 to calculate M_b for $T_b = 1750$, 2250, and 2750 K, and $\lambda = 1.0$. Compare these results with values of M_b obtained from Fig. 4.14 and explain any differences.

4.4 EGR, exhaust gas recirculation, is often used to reduce NO_x by acting as a diluent in the intake mixture.

(a) For low-temperature isooctane–air combustion products at $\phi = 1.0$, determine the percentage of the burned gases average specific heat at constant pressure which comes from each component in the burned gas mixture.

(b) Compare the specific heat at constant pressure of isooctane–air combustion products at $\phi = 1.0$ to that of air, both at 1750 K. This difference is one reason why EGR instead of leaner fuel-air mixture is used to control NO_x emissions.

4.5 Explain qualitatively the causes of the trends in the curves in Fig. 4.23 as ϕ is increased from 1.0, and decreased from 1.0.

4.6 Compare the O_2, CO_2, and CO data in Fig. 4.22 with the predictions of the elemental balance summarized in Table 4.3. Explain any differences.

Assume the fuel is a hydrocarbon with H/C ratio of 2.

4.7 The following exhaust data were obtained from a four-stroke cycle spark-ignition engine. CO, CO_2, and NO_x molar concentrations are all measured fully dry; HC is measured fully wet as ppm C_1. Determine the fuel/air equivalence ratio ϕ for the following three sets of data. Make the following assumptions: (1) the constant K in Eq. (4.61) equals 3.5; (2) the fuel composition is C_8H_{15}; (3) the unburned hydrocarbon H/C ratio is the same as that of the fuel; and (4) NO_x is entirely NO.

(a) CO_2 14.0%, CO 0.64%, O_2 0.7%, NO_x 3600 ppm, HC 3200 ppm C_1.

(b) CO_2 13.8%, CO 3.05%, O_2 0%, NO_x 1600 ppm, HC 3450 ppm C_1.

(c) CO_2 12.5%, CO 0.16%, O_2 4.0%, NO_x 4600 ppm, HC 2100 ppm C_1.

4.8 The exhaust from a spark-ignition engine has the following composition (in mole fractions):

$$CO_2 = 0.12; \quad H_2O = 0.14; \quad CO = 0.01; \quad H_2 = 0.005;$$
$$N_2 = 0.7247; \quad C_8H_{18} = 0.0003$$

The fuel is isooctane, C_8H_{18}; as shown above, a small fraction of the fuel escapes from the cylinder unburned. The lower heating value of isooctane is 44.4 MJ/kg, of carbon monoxide is 10.1 MJ/kg, and of hydrogen is 120 MJ/kg. The atomic weights of the elements are: C = 12, O = 16, H = 1, N = 14.

(a) Calculate the combustion *in*efficiency of the engine; that is, the percentage of the entering fuel enthalpy which is not fully released in the combustion process and leaves the engine in the exhaust gases (for this problem, the exhaust can be assumed to be at room temperature).

(b) What fraction of this inefficiency is due to the unburned fuel (HC) emissions?

4.9 A 2-liter displacement four-cylinder engine, operating at 2000 rev/min and 30% of its maximum power at that speed, has the following exhaust composition (in percent by volume or mole percent):

CO_2, 11%; H_2O, 11.5%; CO, 0.5%; H_2, 0%; O_2, 2%; unburned hydrocarbons (expressed as CH_2, i.e., with a molecular weight of 14), 0.5%;

N_2, 74.5%.

The fuel is $(CH_2)n$ with a heating value of 44 MJ/kg. The atomic weights of the elements are C = 12, H = 1, O = 16, N = 14. The heating values of CO and HC are 10 and 44 MJ/kg, respectively.

(a) Is the engine a diesel or spark-ignition engine? Is there enough oxygen in the exhaust to burn the fuel completely? Briefly explain.

(b) Calculate the fraction of the input fuel energy ($m_f Q_{HV}$) which exits the engine unburned as (1) CO and (2) unburned HC.

(c) An inventor claims a combustion efficiency of 100% can be achieved. What percentage improvement in engine specific fuel consumption would result?

4.10 A gasoline engine operates steadily on a mixture of isooctane and air. The air and fuel enter the engine at 25°C. The fuel consumption is 3.0 g/s. The output of the engine is 50 kW. The temperature of the combustion products in the exhaust manifold is 660 K. At this temperature, an analysis of the combustion products yields the following values (on a dry volumetric basis): CO_2, 11.4%; O_2, 1.6%; CO, 2.9%; N_2, 84.1%.

(a) Find the composition in moles (number of moles per mole of isooctane) of the reactants and the reaction products.

(b) Determine the heat-transfer rate from the working fluid as the working fluid passes through the engine.

Constants for the calculations:

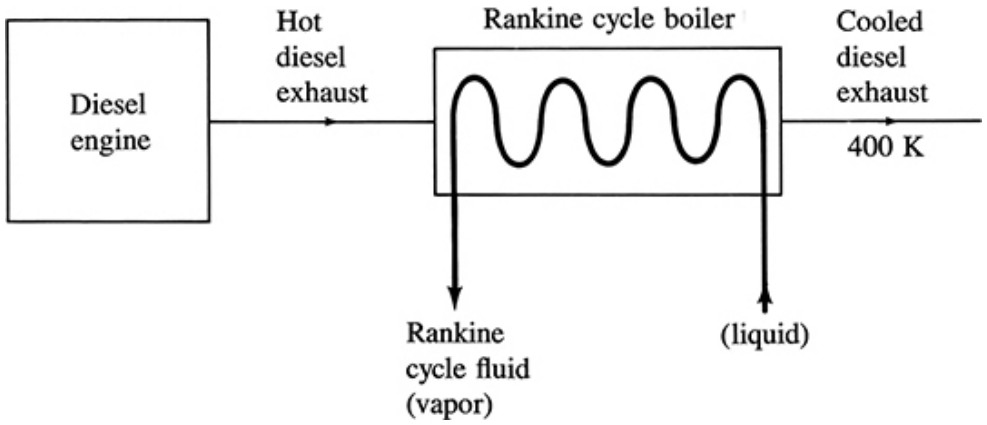
	Enthalpy of formation, kJ/kmol	Sensible enthalpy at 660 K, kJ/kmol
C_8H_{18}	−259,280	—
CO_2	−393,522	15,823
CO	−110,529	10,789
H_2O	−241,827	12,710
O_2	—	11,200
N_2	—	10,749

4.11 A direct-injection four-stroke cycle diesel engine is used to provide power for pumping water. The engine operates at its maximum rated power

at 2000 rev/min, with an equivalence ratio of 0.8 and an air flow of 0.5 kg/s. The gross indicated fuel conversion efficiency is 45%, and the heat losses from the working fluid to the engine coolant and elsewhere within the engine are 280 kW. Diesel fuel has a heating value of 42 MJ/kg and stoichiometric fuel/air ratio of 0.067. Fuel and air enter the engine at ambient conditions. The mechanical efficiency of the diesel engine is 85%.

(a) Calculate the rated brake power of the engine, the average sensible enthalpy of the exhaust gases as they leave the engine, and the average exhaust gas temperature.

(b) Since the exhaust gas temperature is significantly above ambient, the advantages of using the diesel exhaust gas stream to heat the boiler of a Rankine cycle (see sketch) and generate additional power are to be explored. If the exhaust gases *leave* the Rankine-cycle system boiler at 400 K and 30% of the heat transferred from the exhaust gas stream in the boiler is converted to power at the Rankine-cycle power plant drive shaft, calculate the additional power obtained and the brake fuel conversion efficiency of the combined cycle system (diesel plus Rankine cycle).



4.12 A diesel engine has a compression ratio of 22:1. The conditions in the cylinder at the start of compression are $p = 101.3$ kPa and $T = 325$ K. Calculate the pressure and temperature at the end of compression, assuming the compression process is isentropic:

(a) Assume the cylinder contains an ideal gas with $\gamma = 1.4$ and $R = 287$ J/kg·K.

(b) Assume the cylinder contains air which may be regarded as a semiperfect gas (use the gas tables).

(c) Compare the work of compression in (*a*) and (*b*) above.

In practice, heat losses reduce the final compression temperature by 100 K. For a diesel engine operating at an equivalence ratio of 0.75 (full load).
(d) Calculate the ratio of heat loss during compression to the fuel energy added per cycle.

4.13 While the geometric compression ratio of an engine is V_{max}/V_{min}, the actual compression process starts somewhere between bottom-center and when the intake valve closes, and it is conditions at time of spark (for an SI engine) or fuel injection (for a CI engine) that determine ignition. At low engine speed, compression starts about the time when the intake valve closes. With this assumption, for the diesel engine of Prob. 4.12, calculate the air pressure and temperature at the start of injection. The intake valve closes at 30° after BC; injection commences 15° before TC. Use the gas tables. Compare your answers with those of Prob. 4.12(*b*).

4.14 Use an equilibrium computer code (which calculates the composition and properties of chemically reacting gas mixtures in equilibrium) to calculate the data you need for the following graphs:
(a) Values of c_p, γ, molecular weight, and gas composition (mole fractions of N_2, CO_2, H_2O, CO, H_2, O_2, OH, O, H, and NO) as a function of the equivalence ratio ($\phi = 0.2$ to 1.4) for products of combustion of isooctane (C_8H_{18}) with air at $p = 40$ atm and $T = 2500$ K. Put all species concentrations on the same graph. Use a log scale for the composition axis.
(b) Unburned mixture consisting of isooctane vapor and air at 700 K and 20 atm is burned first at constant pressure and then at constant volume.

1. Calculate the enthalpy and internal energy of isooctane vapor at 700 K in cal/gmol; also calculate the volume per unit mass of mixture (cm^3/g) for $\phi = 0.2, 0.4, 0.6, 0.8, 1.0, 1.2, 1.4$.
2. Use these data and the equilibrium program to calculate the temperature attained after combustion at constant pressure, and temperature and pressure attained after combustion at constant volume. Plot these temperatures and pressures against the equivalence ratio ϕ.

T, K	$\tilde{c}_p$, cal/mol·K	$\tilde{h} - \tilde{h}_{298}$, kcal/mol	$\Delta\tilde{h}_f^\circ$, kcal/mol
298	45.14	0.00	−53.57
700	85.66	27.02	−62.79

4.15 A strong walled vessel with a volume of 1000 cm^3 contains a mixture of isooctane with the stoichiometric air requirement at p = 101.3 kPa and T = 25°C. The mixture is then ignited with a spark. Find the pressure and temperature of the equilibrium combustion products just after combustion is complete (i.e., before heat losses to the wall are significant). Assume the burned gases are uniform.

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[a] An alternative definition of percent EGR (EGR*) is also used based on the ratio of EGR to fresh mixture (fuel and air):

$$\varepsilon = \frac{4}{4+y}$$

The two definitions are related by

$$\text{EGR}^*(\%) = \left(\frac{m_{\text{EGR}}}{m_a + m_f} \right) \times 100$$

[b] The error in ignoring the effect of dissociation on M_b is small.

[c] This assumption introduces negligible error into calculations of the compression process for mixtures with normal burned gas fractions, since the major constituent of the residual is N_2. The burned gas fraction must, however, be included when the unburned mixture properties are related to burned mixture properties in a combustion process.

[d] In Couette flow, the fluid is contained between two infinite plane parallel surfaces, one at rest and one moving with constant velocity. In the absence of pressure gradients, the fluid velocity varies linearly across the distance between the surfaces.

[e] In the NASA program, "frozen" means the gas composition is in equilibrium at the given T and p, but is frozen as c_p, c_v, and k are computed.

[f] Equation (4.61) is often described as assuming a specific value for the water–gas reaction equilibrium constant. In fact, K is an empirical constant determined from exhaust gas composition data.

CHAPTER 5

Ideal Models of Engine Cycles

5.1 INTRODUCTION

The operating cycle of an internal combustion engine can be broken down into a sequence of separate processes: intake, compression, combustion, expansion, and exhaust. With models for each of these processes, a simulation of a complete engine cycle can be built up which can be analyzed to provide information on engine performance. Models of individual engine processes at various levels of approximation have been developed. In this chapter we consider the simplest set of models which provide useful insights into the performance and efficiency of engines. The cycles analyzed are commonly called the constant-volume, constant-pressure, and limited-pressure cycles; each title describes the approximation made for the engine combustion process. [a] The description of more accurate simulations of engine processes is deferred until Chap. 14 .

For each engine cycle, a choice of models for working fluid thermodynamic properties must be made. These models have been reviewed in Chap. 4 . Ideal engine cycle models combined with a simple ideal gas model (specific heats constant throughout the engine cycle, model 1 in Table 4.2) provide analytic results and are useful for illustrative purposes; we will call these cycles *ideal gas standard cycles*. Ideal engine cycles combined with more realistic models of working fluid properties (a frozen mixture of ideal gases for the unburned mixture and an equilibrium mixture for the burned mixture, model 5 in Table 4.2) are called *fuel-air cycles* and provide more quantitative information on engine operation.

An internal combustion engine is not a heat engine in the thermodynamic definition of the term. The engine's boundaries do not define a closed system. The working fluid does not execute a thermodynamic cycle. The temperature changes which occur around minimum and maximum cylinder volumes are not primarily a result of heat-transfer interactions. An internal combustion engine can best be analyzed as an open system which exchanges heat and work with its surrounding environment (the atmosphere). Reactants (fuel and air) flow into the engine; products (exhaust gases) flow out. (An overall second law analysis of the engine from this point of view has already been presented in Sec. 3.6 .) Thus, the cycles discussed in this chapter are not thermodynamic cycles. Rather, each is a sequence of processes through which we can follow the state of the working fluid as the engine executes a complete *operating* cycle.

5.2 IDEAL MODELS OF ENGINE PROCESSES

The sequence of processes which make up a typical SI and CI engine operating cycle has been described in Sec. 1.3 . To illustrate these processes, cylinder pressure (p) and volume (V) data from a throttled four-stroke cycle SI engine are plotted as a p-V diagram in Fig. 5.1. The cycle can be divided into compression, combustion, expansion, exhaust, and intake processes. Sets of assumptions which simplify each of these processes to a form convenient for analysis are given in Table 5.1.

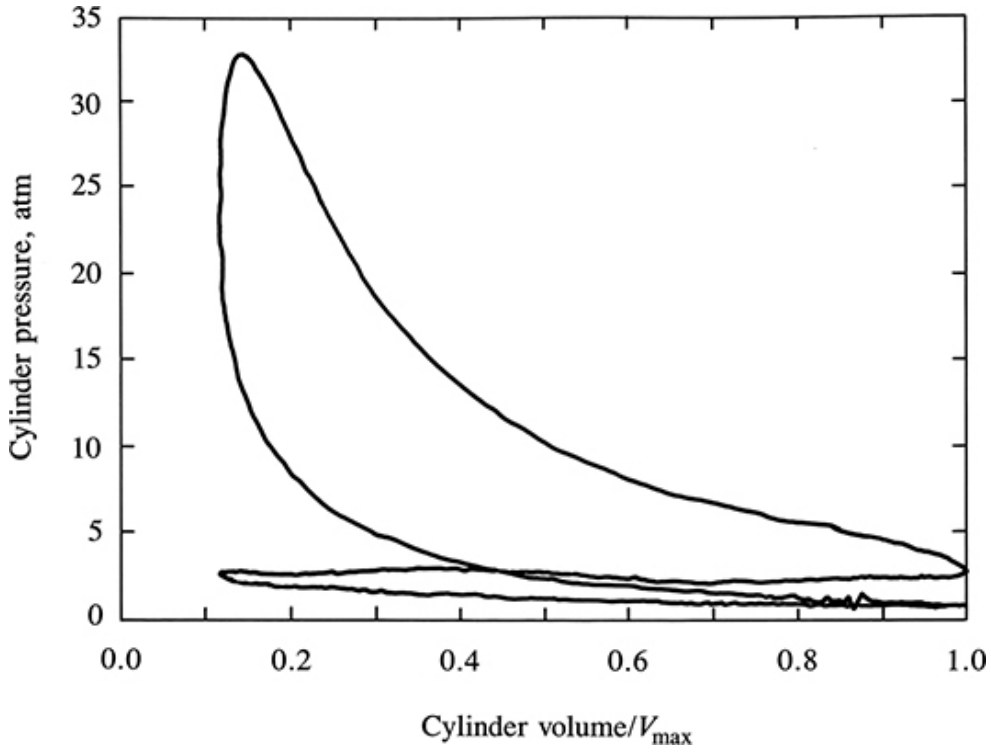


Figure 5.1 Pressure-volume diagram of firing spark-ignition engine. r_c = 8.4, 3500 rev/min, p_i = 0.4 atm, p_e = 1 atm, imep$n$ = 2.9 atm.

TABLE 5.1 Ideal models of engine processes

Process	Assumptions
Compression (1-2)	1. Adiabatic and reversible (hence isentropic)
Combustion (2-3)	1. Adiabatic 2. Combustion occurs at (*a*) Constant volume (*b*) Constant pressure (*c*) Part at constant volume and part at constant pressure (called limited pressure) 3. Combustion is complete ($\eta_c = 1$)
Expansion (3-4)	1. Adiabatic and reversible (hence isentropic)
Exhaust (4-5-6) and intake (6-7-1)	1. Adiabatic 2. Valve events occur at top- and bottom-center 3. No change in cylinder volume as pressure differences across open valves drop to zero 4. Inlet and exhaust pressures constant 5. Velocity effects negligible

Pressure-volume diagrams for the constant-volume, constant-pressure, and limited-pressure cycles for unthrottled engine operation are illustrated in Figs. 5.2*a* to *c*. Throttled engine operation ($p_i < p_e$) and supercharged engine operation ($p_i > p_e$) are shown in Figs. 5.2*d* and *e*. In each cycle, 1-2 is the compression process, 2-3 is the combustion process, 3-4 (or 2-4 in the constant-pressure cycle) is the expansion process, 4-5-6 is the exhaust process, and 6-7-1 is the intake process.

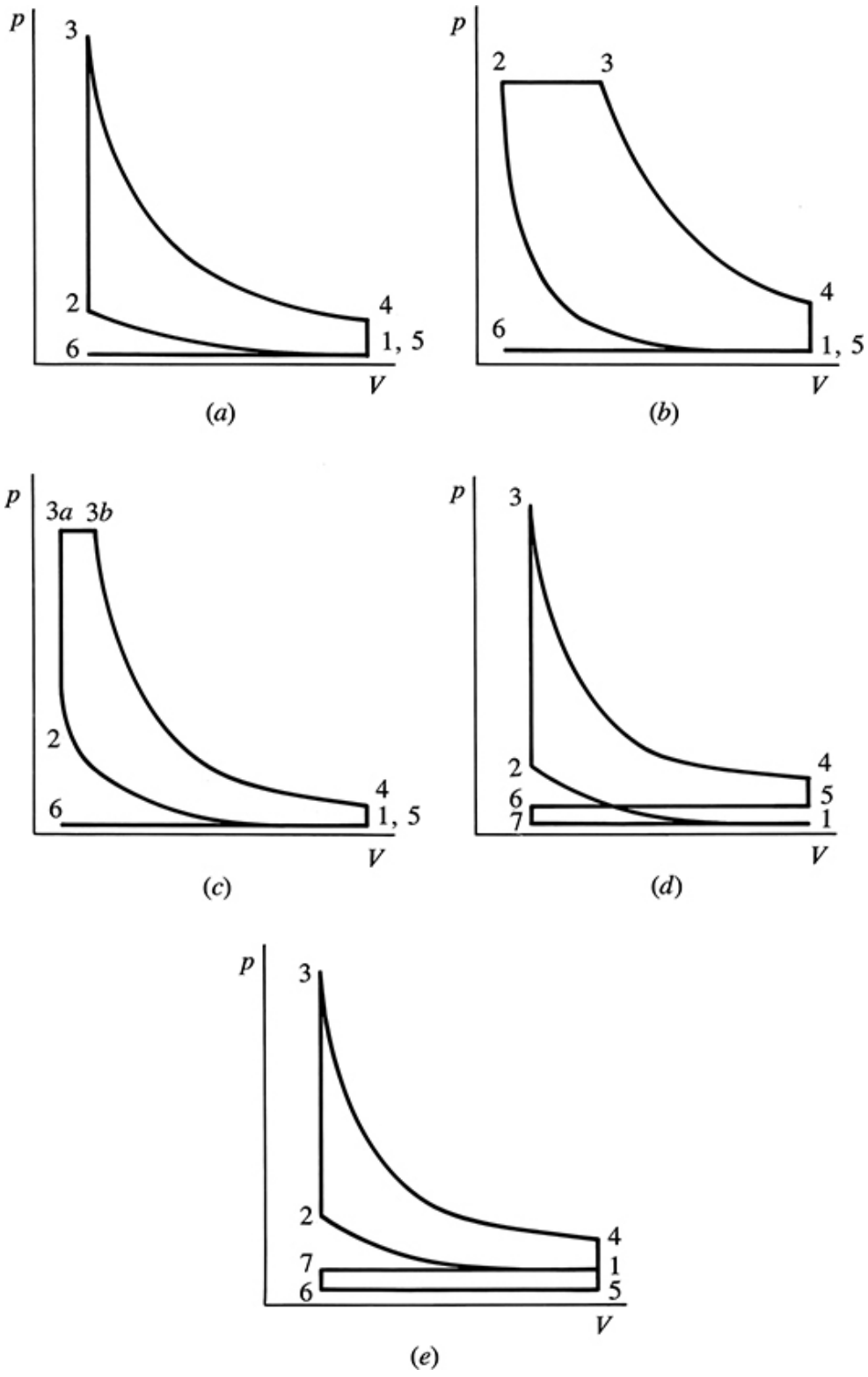


Figure 5.2 Pressure-volume diagrams of ideal cycles. Unthrottled operation: (*a*) constant-volume combustion; (*b*) constant-pressure

combustion; (*c*) limited-pressure combustion. (*d*) Throttled constant-volume cycle; (*e*) supercharged constant-volume cycle.

The most critical assumption in determining how useful these ideal cycles are as indicators of engine performance is the form assumed for the combustion process. The real engine combustion process occupies a finite crank angle period (between about 20 and 60 crank angle degrees), and the spark or fuel-injection timing may be retarded from its optimum advance to closer to top center, TC. The constant-volume cycle is the limiting case of infinitely fast combustion at TC; the constant-pressure cycle would correspond to slow and late combustion; the limited-pressure cycle lies in between.

5.3 THERMODYNAMIC RELATIONS FOR ENGINE PROCESSES

The overall engine operating parameters of greatest interest which can be determined from a thermodynamic analysis of the engine operating cycle are:

The indicated fuel conversion efficiency $\eta_{f,i}$:

$$\eta_{f,i} = \frac{W_{c,i}}{m_f Q_{\mathrm{LHV}}} \tag{5.1}$$

(which, since the combustion efficiency is unity, is equal to the indicated thermal conversion efficiency $\eta_{t,i}$; see Sec. 3.6.2 .)

The indicated mean effective pressure (imep):

$$\mathrm{imep} = \frac{W_{c,i}}{V_d} = \frac{m_f Q_{\mathrm{LHV}} \eta_{f,i}}{V_d} \tag{5.2}$$

$W_{c,i}$, the indicated work per cycle, is the algebraic sum of the expansion stroke work and the compression stroke work (a negative quantity):

$$W_{c,i} = W_E + W_C \tag{5.3}$$

Using the notation of Fig. 5.2 to define the endpoints of each engine process, the following relationships are obtained by applying the first and second laws of thermodynamics to the cylinder contents:

Compression stroke:

$$\frac{v_1}{v_2} = r_c \tag{5.4}$$

Since the process is adiabatic and reversible,

$$s_2 = s_1 \tag{5.5}$$

The compression work is

$$W_C = U_1 - U_2 = m(u_1 - u_2) \tag{5.6}$$

Combustion process:

For the constant-volume cycle,

$$v_3 = v_2 \qquad u_3 - u_2 = 0 \tag{5.7a, b}$$

For the constant-pressure cycle,

$$p_3 = p_2 \qquad h_3 - h_2 = 0 \tag{5.7c, d}$$

For the limited-pressure cycle,

$$v_{3a} = v_2 \qquad p_{3b} = p_{3a} \tag{5.7e, f}$$

and

$$u_{3a} - u_2 = 0 \qquad h_{3b} - h_{3a} = 0 \tag{5.7g, h}$$

Expansion stroke:

For the constant-volume cycle,

$$\frac{v_4}{v_3} = r_c \qquad s_4 = s_3 \tag{5.8a, b}$$

and the expansion work is

$$W_E = U_3 - U_4 = m(u_3 - u_4) \tag{5.9}$$

For the constant-pressure cycle,

$$p_3 = p_2 \qquad \frac{v_4}{v_2} = r_c \qquad s_4 = s_3 \tag{5.10a, b, c}$$

and the expansion work is

$$\begin{aligned} W_E &= U_3 - U_4 + p_2(V_3 - V_2) \\ &= m\left[(u_3 - u_4) + p_2(v_3 - v_2)\right] \\ &= m\left[(h_3 - h_4) + p_4 v_4 - p_2 v_2\right] \end{aligned} \tag{5.11}$$

For the limited-pressure cycle,

$$v_4 / v_{3a} = r_c \qquad p_{3b} = p_{3a} \qquad s_4 = s_{3b} \tag{5.12a, b, c}$$

and the expansion work is

$$\begin{aligned} W_E &= U_{3b} - U_4 + p_3(V_{3b} - V_{3a}) \\ &= m\left[(u_{3b} - u_4) + p_3(v_{3b} - v_{3a})\right] \\ &= m\left[(h_{3b} - h_4) + p_4 v_4 - p_3 v_{3a}\right] \end{aligned} \tag{5.13}$$

The indicated fuel conversion efficiency is found by substitution into Eqs. (5.3) and (5.1):
For the constant-volume cycle:

$$\eta_{f,i} = \frac{m\left[(u_3 - u_4) - (u_2 - u_1)\right]}{m_f Q_{\text{LHV}}} \tag{5.14}$$

For the constant-pressure cycle:

$$\eta_{f,i} = \frac{m\left[(h_3 - h_4) - (u_2 - u_1) + p_4 v_4 - p_2 v_2\right]}{m_f Q_{\text{LHV}}} \tag{5.15}$$

For the limited-pressure cycle:

$$\eta_{f,i} = \frac{m\left[(h_{3b} - h_4) - (u_2 - u_1) + p_4 v_4 - p_3 v_{3a}\right]}{m_f Q_{\text{LHV}}} \tag{5.16}$$

The state of the mixture at point 1 in the cycle depends on the intake mixture properties and the residual gas properties at the end of the exhaust stroke. When the exhaust valve opens at point 4, the cylinder pressure is above the exhaust manifold pressure and a *blowdown* process occurs. In the ideal exhaust process model, this blowdown occurs with the piston stationary at BC. During this blowdown process, the gas which remains inside the cylinder expands isentropically. The gases escaping from the cylinder undergo an unrestrained expansion or throttling process which is irreversible. It is assumed that the kinetic energy acquired by each gas element as it is accelerated through the exhaust valve is dissipated in a turbulent mixing process in the exhaust port into internal energy and flow work. Since it is also assumed that no heat transfer occurs, the enthalpy of each element of gas after it leaves the cylinder remains constant.

These processes are illustrated on an *h-s* diagram in Fig. 5.3. The gas remaining in the cylinder expands isentropically along the line 4-5. The first element of gas which leaves the cylinder at point 4 enters the exhaust manifold at state *a* on the pressure = p_e line. An element that leaves the cylinder at an intermediate state *b* on the expansion line 4-5 would enter the exhaust manifold at state *c*. At the end of the blowdown process the gas in the cylinder and the last gas to leave have the same state, 5. There is, therefore, a gradient in temperature within the exhausted gas. The temperature of the first element exhausted, T_a, is slightly less than T_4; the temperature of the last element exhausted is T_5.

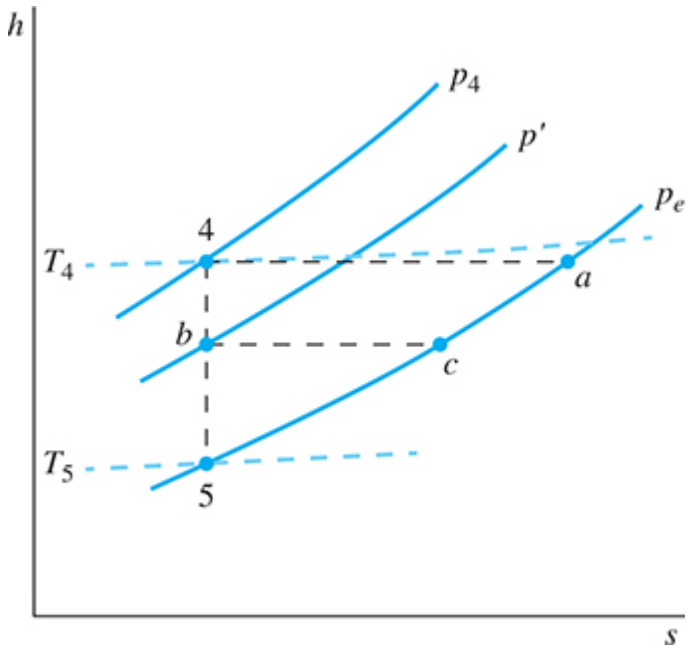


Figure 5.3 Enthalpy-entropy diagram of gas state during exhaust process. See text for explanation.

A displacement of gas out of the cylinder follows the blowdown process as the piston moves from BC to TC. If heat-transfer and kinetic energy dissipation effects are neglected, no change in thermodynamic state of the gas occurs. In this *displacement* process, the mass of gas within the cylinder at the end of the blowdown process is further decreased by the ratio V_5/V_6.

The mass of residual gas m_r in the cylinder at point 6 in the cycle is obtained by first determining the state of the gas (T_5, v_5) at the end of the blowdown process following an isentropic expansion from p_4 to p_e and then by reducing the cylinder volume to the clearance volume V_6. The residual mass fraction is thus given by

$$\frac{m_r}{m} = x_r = \frac{v_4/v_5}{r_c} = \frac{v_2}{v_5} \tag{5.17}$$

The average state of the exhausted gas can be determined by considering the open system defined by the piston face, cylinder walls, and cylinder head, shown in Fig. 5.4. Applying the first law of thermodynamics for an open system gives

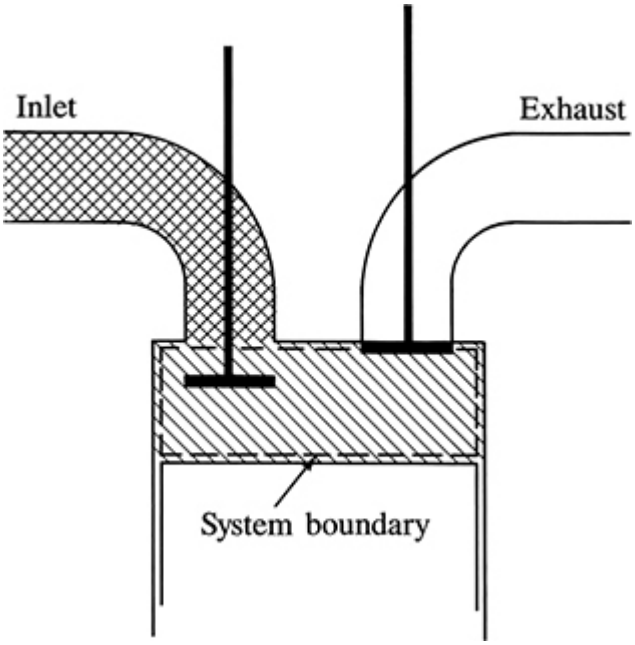


Figure 5.4 Definition of system boundary for thermodynamic analysis of ideal cycle processes.

$$U_6 - U_4 = p_e(V_4 - V_6) - H_e \tag{5.18a}$$

where H_e is the enthalpy of the mass of gas exhausted from the cylinder. The average specific exhaust enthalpy is, therefore,

$$\bar{h}_e = \frac{m_4 u_4 - m_6 u_6 + p_e V_d}{m_4 - m_6} \tag{5.18b}$$

which, with $p = p_e$, defines the average exhausted-gas state.

The mixture temperature at the end of the intake stroke and at the start of the compression stroke (point 1 in Fig. 5.2) can now be determined, again using the open system in Fig. 5.4. Application of the first law between points 6 and 1 gives

$$U_1 - U_6 = -p_i(V_1 - V_6) + (m_1 - m_6)h_i \tag{5.19a}$$

or

$$m_1 u_1 - m_6 u_6 = -p_i(V_1 - V_6) + (m_1 - m_6)h_i \tag{5.19b}$$

or

$$m_1 h_1 = m_6 h_6 + (m_1 - m_6)h_i + V_2(p_i - p_e) \tag{5.19c}$$

where h_i is the specific enthalpy of the inlet mixture and $p_1 = p_i$.

Note that when $p_i < p_e$, part of the residual gas in the cylinder at the end of the exhaust stroke will flow into the intake system when the intake valve opens. This flow will cease when the cylinder pressure equals p_i. However, provided no heat transfer occurs, this backflow will not affect Eqs. (5.19) above, since the flow of residual through the intake valve is then a constant enthalpy process.

In many engines, the closing of the exhaust valve and the opening of the intake valve overlap. Flow of exhausted gases from the exhaust system through the cylinder into the intake system can then occur. Equations (5.18) and (5.19) would have to be modified to account for valve overlap.

In the four-stroke engine cycle, work is done on the piston during the intake and the exhaust processes. The work done by the cylinder gases on the piston during exhaust is

$$W_e = p_e(V_2 - V_1) \tag{5.20}$$

The work done by the cylinder gases on the piston during intake is

$$W_i = p_i(V_1 - V_2) \tag{5.21}$$

The net work to the piston over the exhaust and intake strokes, the *pumping work*, is

$$W_p = (p_i - p_e)(V_1 - V_2) \tag{5.22}$$

which, for the cylinder gas system, is negative for $p_i < p_e$ and positive for $p_i > p_e$. [b] The pumping mean effective pressure (pmep) is usually defined as a positive quantity. Thus:

For

$$p_i < p_e: \quad \text{pmep} = p_e - p_i \tag{5.23a}$$

For

$$p_i > p_e: \quad \text{pmep} = p_i - p_e \tag{5.23b}$$

The net and gross indicated mean effective pressures are related by

$$\text{imep}_n = \text{imep}_g - (p_e - p_i) \tag{5.24}$$

The net indicated fuel conversion efficiency is related to the gross indicated fuel conversion efficiency by

$$\eta_{f,\text{in}} = \eta_{f,\text{ig}} \left(1 - \frac{p_e - p_i}{\text{imep}_g} \right) \tag{5.25}$$

5.4 CYCLE ANALYSIS WITH IDEAL GAS WORKING FLUID WITH c_v AND c_p CONSTANT

If the working fluid in these ideal cycles is assumed to be an ideal gas, with c_v and c_p constant throughout the engine operating cycle, the equations developed in the previous section which describe engine performance and efficiency can be further simplified. We will use the notation of Fig. 5.2.

5.4.1 Constant-Volume Cycle

The compression work [Eq. (5.6)] becomes

$$W_C = mc_v(T_1 - T_2) \tag{5.26}$$

The expansion work [Eq. (5.9)] becomes

$$W_E = mc_v(T_3 - T_4) \tag{5.27}$$

The denominator in Eq. (5.14) , $m_f Q_{\mathrm{LHV}}$, can be related to the temperature rise during combustion. For the working fluid model under consideration, the *U(T)* lines for the reactants and products on a *U-T* diagram such as Fig. 3.5 are parallel and have equal slopes, of magnitude c_v. Hence, for a constant-volume adiabatic combustion process

$$mc_v(T_3 - T_2) = m_f Q_{\mathrm{LHV}} \tag{5.28}^c$$

Note that the heating values at constant volume and constant pressure are the same for this working fluid. For convenience we will define

$$Q^* = \frac{m_f Q_{\mathrm{LHV}}}{m} \tag{5.29}$$

Q * is the specific internal energy (and enthalpy) decrease during isothermal combustion, per unit mass of working fluid.

The relation for indicated fuel conversion efficiency [Eq. (5.14)] becomes

$$\eta_{f,i} = \frac{(T_3 - T_4) - (T_2 - T_1)}{T_3 - T_2} = 1 - \frac{T_4 - T_1}{T_3 - T_2} \tag{5.30}$$

Since 1-2 and 3-4 are isentropic processes between the same volumes, V_1 and V_2,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = r_c^{\gamma-1} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = \frac{T_3}{T_4}$$

where $\gamma = c_p / c_v$ Hence:

$$\frac{T_4}{T_1} = \frac{T_3}{T_2}$$

and Eq. (5.30) can be rearranged as

$$\eta_{f,i} = 1 - \frac{1}{r_c^{\gamma-1}} \tag{5.31}$$

Values of $\eta_{f,i}$ for different values of γ (a variable that depends on mixture composition) are shown in Fig. 5.5. The indicated fuel conversion efficiency increases with increasing compression ratio, and as γ increases.

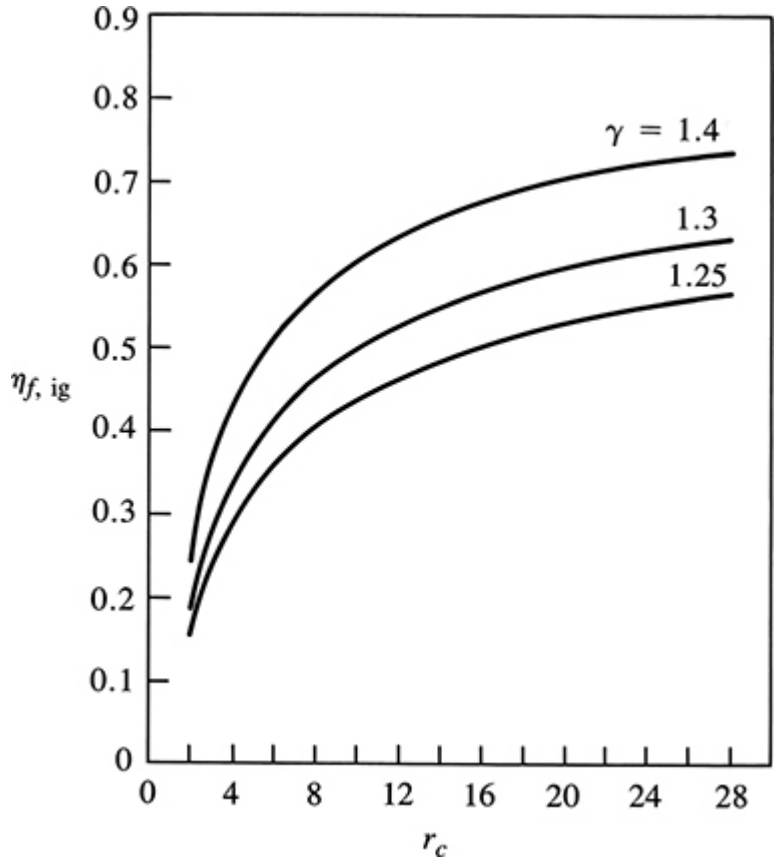


Figure 5.5 Ideal gas constant-volume cycle fuel conversion efficiency as a function of compression ratio; $\gamma = c_p/c_v$.

These trends can be explained by examining how the p-V diagrams for the constant-volume cycle change as r_c and γ are increased, at constant cylinder displaced volume V_d. Increasing r_c at fixed γ increases the pressure at the start of expansion, and the volume ratio through which the charge expands during the expansion stroke. Both of these effects add to the indicated work per cycle. They add positive area to the p-V diagram which more than offsets a slight increase in the (negative) work required during compression. The

effect of increasing γ at constant r_c has an analogous impact. The peak pressure (and temperature and energy) at the start of expansion is increased and the charge expands through a larger temperature ratio during expansion due to its higher γ, thus transferring a larger fraction of its energy at start of expansion to the piston as work.

The indicated mean effective pressure, using Eqs. (5.2) and (5.31), becomes

$$\frac{\text{imep}}{p_1} = \left(\frac{Q^*}{c_v T_1}\right)\left(\frac{1}{\gamma - 1}\right)\left(\frac{r_c}{r_c - 1}\right)\left(1 - \frac{1}{r_c^{\gamma-1}}\right) \tag{5.32}$$

The dimensionless numbers r_c, γ, and $Q^*/(c_v T_1)$ are sufficient to describe the characteristics of the constant-volume ideal gas standard cycle, relative to its initial conditions p_1, T_1.

It is useful to compare the imep—a measure of the effectiveness with which the displaced volume of the engine is used to produce work—and the maximum pressure in the cycle, p_3. The ratio p_3/p_1 can be determined from the ideal gas law applied at points 2 and 3, and the relation

$$\frac{T_3}{T_2} = 1 + \frac{Q^*}{c_v T_1 r_c^{\gamma-1}} \tag{5.33}$$

obtained from Eq. (5.28) . Equations (5.32) and (5.33) then give

$$\frac{\text{imep}}{p_3} = \frac{1}{(\gamma-1)r_c^{\gamma}}\left(\frac{r_c}{r_c - 1}\right)\frac{1 - 1/r_c^{\gamma-1}}{c_v T_1/Q^* + 1/r_c^{\gamma-1}} \tag{5.34}$$

A high value of imep/p_3 is desirable. Engine weight will increase with increasing p_3 to withstand the increasing stresses in components.

The indicated fuel conversion efficiency and the ratios imep/p_1 and imep/p_3 for this ideal cycle model do not depend on whether the cycle is throttled or supercharged. However, the relationships between the working fluid properties at points 1 and 6 do depend on the degree of throttling or supercharging. For throttled engine operation, the residual gas mass fraction

x_r can be determined as follows. From Eq. (5.17) , since state 5 corresponds to an isentropic expansion from state 4 to $p = p_e$, x_r is given by

$$x_r = \frac{(p_e/p_4)^{1/\gamma}}{r_c} = \frac{(p_e/p_i)^{1/\gamma}(p_1/p_4)^{1/\gamma}}{r_c}$$

Since

$$\frac{p_1}{p_4} = \frac{p_1}{p_2}\frac{p_2}{p_3}\frac{p_3}{p_4} = \frac{1}{r_c^\gamma}\frac{T_2}{T_3}r_c^\gamma = \left(1 + \frac{Q^*}{c_v T_1 r_c^{\gamma-1}}\right)^{-1}$$

it follows that

$$x_r = \frac{1}{r_c}\frac{(p_e/p_i)^{1/\gamma}}{\left[1 + Q^*/(c_v T_1 r_c^{\gamma-1})\right]^{1/\gamma}} \tag{5.35}$$

The residual mass fraction increases as p_i decreases below p_e, decreases as r_c increases, and decreases as $Q^*/(c_v T_1)$ increases.

Through a similar analysis, the temperature of the residual gas T_6 can be determined:

$$\frac{T_6}{T_1} = \left(\frac{p_e}{p_i}\right)^{(\gamma-1)/\gamma}\left(1 + \frac{Q^*}{c_v T_1 r_c^{\gamma-1}}\right)^{1/\gamma} \tag{5.36}$$

The mixture temperature at point 1 in the cycle can be related to the inlet mixture temperature, T_i, with Eq. (5.19) . For a working fluid with c_v and c_p constant, this equation becomes

$$c_p T_1 = x_r c_p T_6 + (1 - x_r)c_p T_i - \frac{RT_1}{r_c}\left(\frac{p_e}{p_i} - 1\right) \tag{5.37}$$

Use of Eqs. (5.36) and (5.37) leads to the relation

$$\frac{T_1}{T_i} = \frac{1 - x_r}{1 - 1/(\gamma r_c)\left[p_e/p_i + (\gamma - 1)\right]} \tag{5.38}$$

Extensive results for the constant-volume cycle with $\gamma = 1.4$ can be found in Taylor. [1]

5.4.2 Limited- and Constant-Pressure Cycles

The constant-pressure cycle is a limited-pressure cycle with $p_3 = p_2$. For the limited-pressure cycle, the compression work remains

$$W_C = mc_v(T_1 + T_2) \tag{5.39}$$

The expansion work, from Eq. (5.13) , becomes

$$W_E = m\left[c_v(T_{3b} - T_4) + p_3(v_{3b} - v_{3a})\right] \tag{5.40}$$

For the combustion process, Eqs. (5.7 *g, h*) give

$$m_{f2-3a} Q_{\text{LHV}} = mc_v(T_{3a} - T_2) \tag{5.41a}$$

$$m_{f3a-3b} Q_{\text{LHV}} = mc_p(T_{3b} - T_{3a}) \tag{5.41b}$$

or

$$m_f Q_{\text{LHV}} = m\left[c_v(T_{3a} - T_2) + c_p(T_{3b} - T_{3a})\right] \tag{5.41c}$$

for a working fluid with c_v and c_p constant throughout the cycle.

Combining Eqs. (5.1) , (5.3), and (5.39) to (5.41) and simplifying gives

$$\eta_{f,i} = 1 - \frac{T_4 - T_1}{(T_{3a} - T_2) + \gamma(T_{3b} - T_{3a})}$$

Use of the isentropic relationships for the working fluid along 1-2 and 3 *b*-4, with the substitutions

$$\alpha = \frac{p_3}{p_2} \qquad \beta = \frac{V_{3b}}{V_{3a}} \tag{5.42a, b}$$

leads to the result

$$\eta_{f,i} = 1 - \frac{1}{r_c^{\gamma-1}}\left[\frac{\alpha\beta^{\gamma} - 1}{\alpha\gamma(\beta - 1) + \alpha - 1}\right] \tag{5.43}$$

For $b = 1$ this result becomes the constant-volume cycle efficiency [Eq. (5.31)]. For $a = 1$, this result gives the constant-pressure cycle efficiency as a special case.

The mean effective pressure is related to p_1 and p_3 via

$$\frac{\text{imep}}{p_1} = \frac{Q^*}{c_v T_1(\gamma - 1)}\left(\frac{r_c}{r_c - 1}\right)\eta_{f,i} \tag{5.44}$$

$$\frac{\text{imep}}{p_3} = \frac{1}{\alpha r_c^{\gamma}}\left(\frac{Q^*}{c_v T_1}\right)\left(\frac{1}{\gamma - 1}\right)\left(\frac{r_c}{r_c - 1}\right)\eta_{f,i} \tag{5.45}$$

5.4.3 Cycle Comparison

The above expressions are most useful if values for γ and $Q^*/(c_v T_1)$ are chosen to match real working fluid properties. Figure 5.5 has already shown the sensitivity of $\eta_{f,i}$ for the constant-volume cycle to the value of γ chosen. In Sec. 4.4 , average values of γ_u and γ_b were determined which match real working fluid properties over the compression and expansion strokes, respectively. Values for a stoichiometric mixture appropriate to an SI engine are $\gamma_u \approx 1.3$, $\gamma_b \approx 1.2$. However, analysis of pressure-volume data for real engine cycles indicates that pV^n, where $n \approx 1.3$, is also a good fit to the expansion stroke p-V data. [2] Heat transfer from the burned gases increases the exponent above the value corresponding to γ_b. A value of $\gamma = 1.3$ for the entire cycle is thus a reasonable compromise.

Q^*, defined by Eq. (5.29) , is the enthalpy decrease during isothermal combustion per unit mass of working fluid. Hence

$$Q^* = \left(\frac{m_f}{m_a}\right) Q_{\mathrm{LHV}} \left(\frac{m_a}{m}\right) \tag{5.46}$$

A simple approximation for (m_a/m) is ($r_c - 1$)/ r_c; that is, fresh air fills the displaced volume and the residual gas fills the clearance volume at the same density. Then, for isooctane fuel, for a stoichiometric mixture, Q^* is given by 2.92×10^6 ($r_c - 1$)/ r_c J/kg air. For $\gamma = 1.3$ and an average molecular weight $M = 29.3$, $c_v = 946$ J/kg · K. For $T_1 = 333$ K, $Q^*/(c_v T_1)$ becomes 9.3 ($r_c - 1$)/ r_c. For this value of $Q^*/(c_v T_1)$ all cycles would be burning a stoichiometric mixture with an appropriate residual gas fraction.

Pressure-volume diagrams for the three ideal cycles for the same compression ratio and unburned mixture composition are shown in Fig. 5.6. For each cycle, $\gamma = 1.3$, $r_c = 12$, $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c = 8.525$. Overall performance characteristics for each of these cycles are summarized in Table 5.2. The constant-volume cycle has the highest efficiency, the constant-pressure cycle the lowest efficiency. This can be seen from Eq. (5.43) where the term in square brackets is equal to unity for the constant-volume cycle and greater than unity for the limited- and constant-pressure cycles. The imep values are proportional to *hf,i* since the mass of fuel burned per cycle is the same in all three cases.

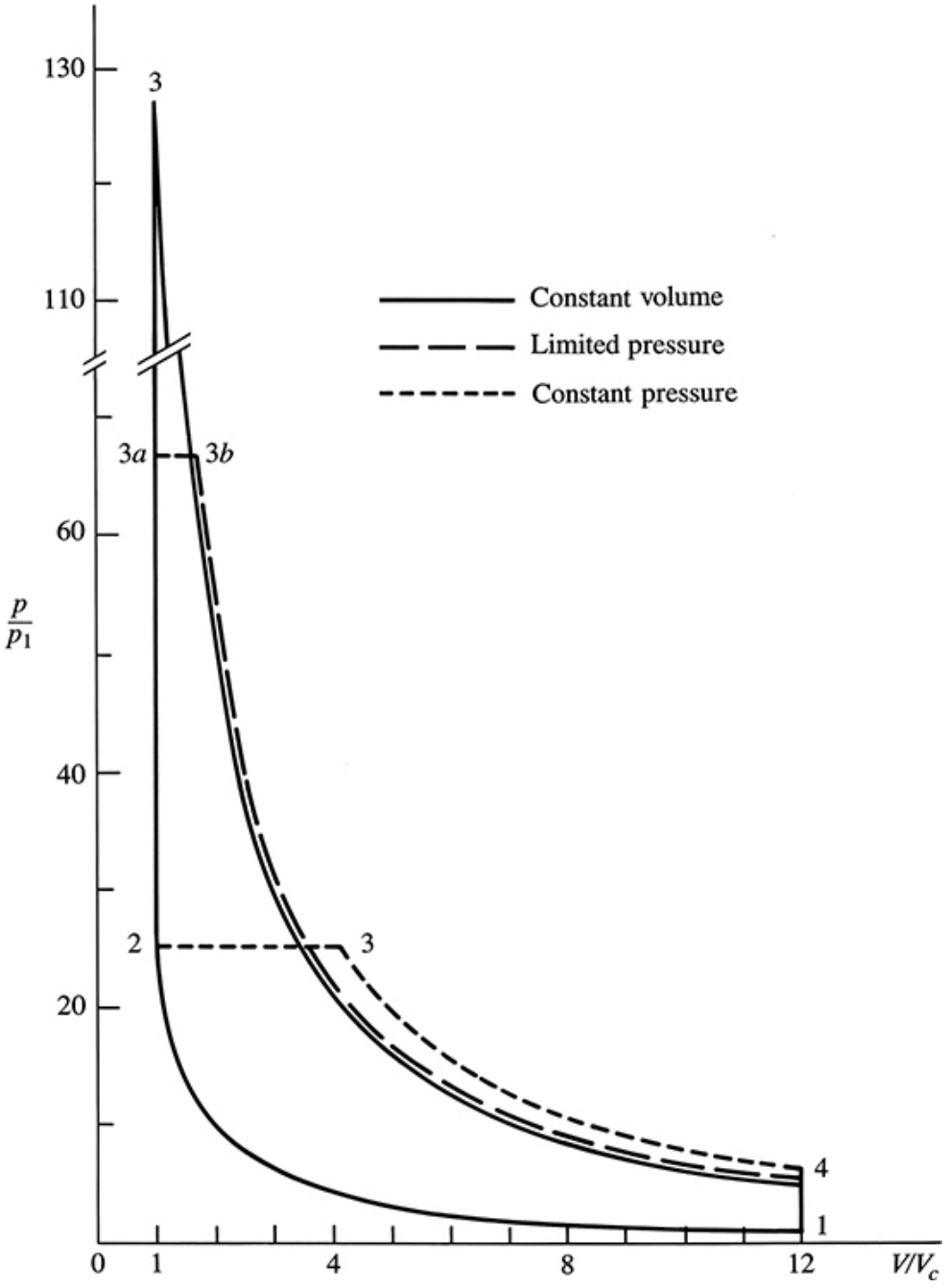


Figure 5.6 Pressure-volume diagrams for constant-volume, limited-pressure, and constant-pressure ideal gas standard cycles. $r_c = 12$, $\gamma = 1.3$, $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c = 8.525$, $p_3 a/p_1 = 67$.

TABLE 5.2 Comparison of ideal cycle results

	$\eta_{f,i}$	$\frac{\text{imep}}{p_1}$	$\frac{\text{imep}}{p_3}$	$\frac{p_{max}}{p_1}$
Constant volume	0.525	16.3	0.128	128
Limited pressure	0.500	15.5	0.231	67
Constant pressure	0.380	11.8	0.466	25.3

$\gamma = 1.3$; $r_c = 12$; $Q^*/(c_v T_1) = 8.525$.

As the peak pressure p_3 is decreased, the ratio of imep to p_3 increases. This ratio is important because imep is a measure of the useful pressure on the piston, and the maximum pressure chiefly affects the strength required of the engine structure.

A more extensive comparison of the three cycles is given in Figs. 5.7 and 5.8, over a range of compression ratios. For all cases $\gamma = 1.3$ and $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c$. At any given r_c, the constant-volume cycle has the highest efficiency and lowest imep/p_3. For a given maximum pressure p_3, the constant-pressure cycle has the highest efficiency (and the highest compression ratio). For the limited-pressure cycle, at constant p_3/p_1, there is little improvement in efficiency and imep above a compression ratio of about 10 as r_c is increased.

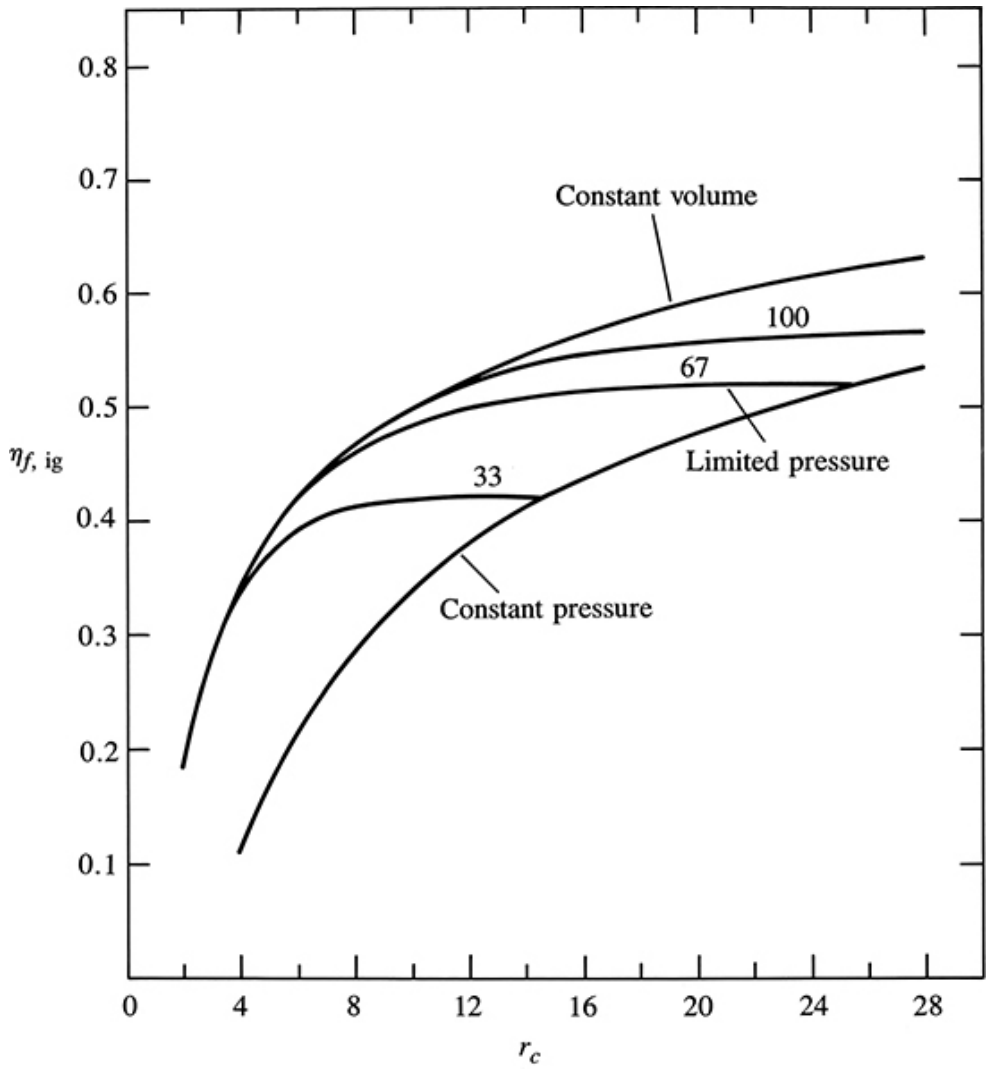


Figure 5.7 Fuel conversion efficiency as a function of compression ratio, for constant-volume, constant-pressure, and limited-pressure ideal gas cycles. $\gamma = 1.3$, $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c$. For limited-pressure cycle, $p_3/p_1 = 33, 67, 100$.

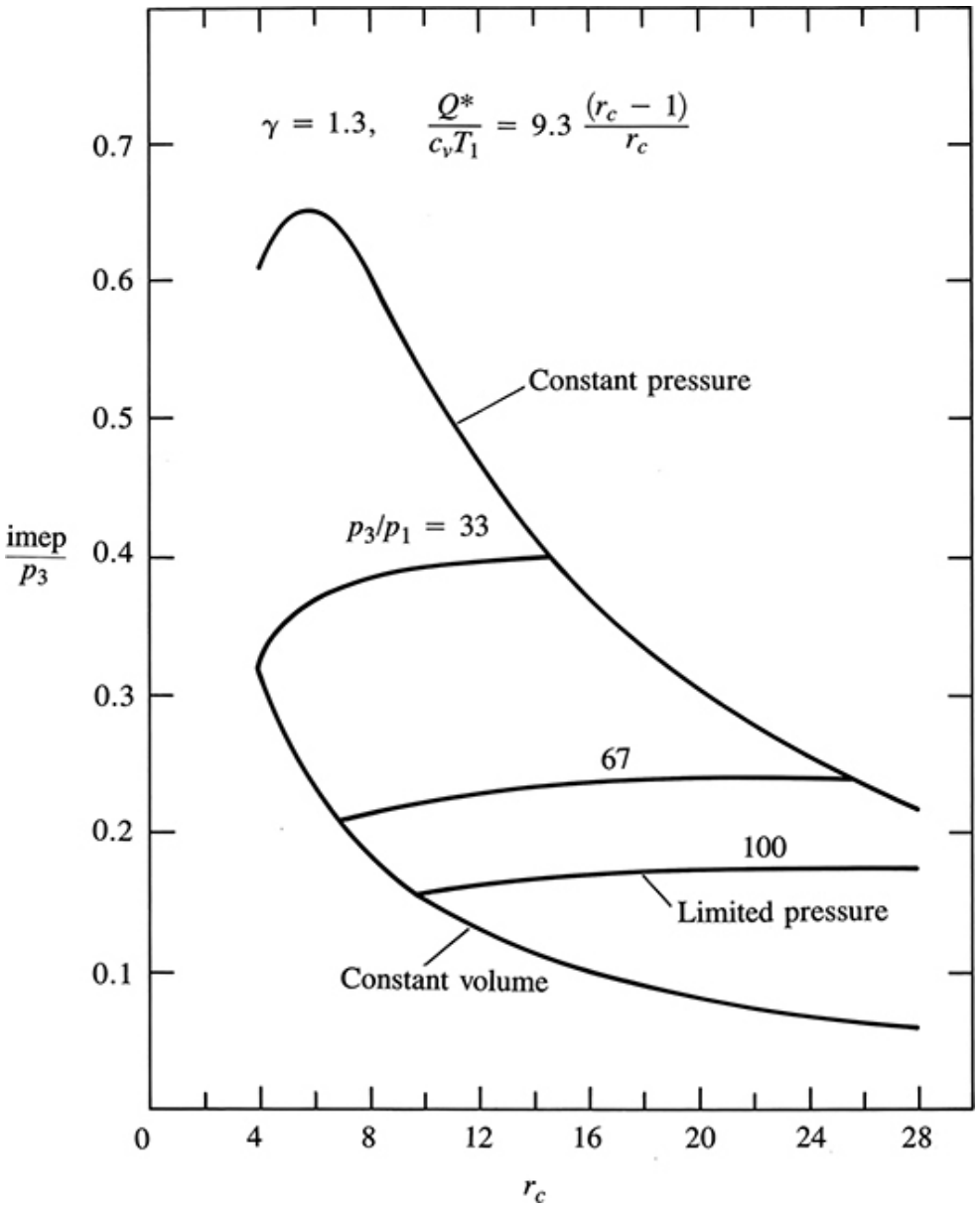


Figure 5.8 Indicated mean effective pressure (imep) divided by maximum cycle pressure (p_3) as a function of compression ratio for constant-volume, constant-pressure, and limited-pressure cycles. Details same as Fig. 5.7.

Example 5.1 shows how ideal cycle equations relate residual and intake conditions with the gas state at point 1 in the cycle. An iterative procedure is required if intake conditions are specified.

EXAMPLE 5.1

For $\gamma = 1.3$, compression ratio $r_c = 6$, and a stoichiometric mixture with intake temperature 300 K, find the residual gas fraction, residual gas temperature, and mixture temperature at point 1 in the constant-volume cycle for $p_e/p_i = 1$ (unthrottled operation) and 2 (throttled operation).

For a stoichiometric mixture, for isooctane,

$$Q^* = \frac{m_f}{m} Q_{\text{LHV}} = \left(\frac{m_f}{m_i}\right)\left(\frac{m_i}{m}\right) Q_{\text{LHV}} = \frac{44.38}{16.14}(1 - x_r) = 2.75(1 - x_r) \quad \text{MJ/kg}$$

For $\gamma = 1.3$, $c_v = 946$ J/kg · K,

$$\frac{Q^*}{c_v T_1} = \frac{2.75 \times 10^6}{946 T_1}(1 - x_r) = \frac{2910}{T_1}(1 - x_r) \qquad (a)$$

and Eqs. (5.35) , (5.36), and (5.38), for $r_c = 6$ and $\gamma = 1.3$, become

$$x_r = \frac{1}{6} \frac{(p_e/p_i)^{0.769}}{6\left[1 + Q^*/(c_v T_1 \times 6^{0.3})\right]^{0.769}} \qquad (b)$$

$$\frac{T_r}{T_1} = \left(\frac{p_e}{p_i}\right)^{0.23} \left(1 + \frac{Q^*}{c_v T_1 \times 6^{0.3}}\right)^{0.769} \qquad (c)$$

$$\frac{T_1}{300} = \frac{1 - x_r}{1 - [1/(1.3 \times 6)](p_e/p_i + 0.3)} \qquad (d)$$

A trial-and-error solution of Eqs. (*a*) to (*d*) is required. It is easiest to estimate x_r, solve for T_1 from (*d*), evaluate $Q^*/(c_v T_1)$ from (*a*), and check the value of x_r assumed with that given by (*b*).

For (p_e/p_i) = 1 (unthrottled operation) the following solution is obtained:

$$x_r = 0.044, \quad T_1 = 344 \text{ K}, \quad \frac{Q^*}{c_v T_1} = 8.1, \quad T_r = 1316 \text{ K}$$

For (p_e / p_i) = 2 the following solution is obtained:

$$x_r = 0.082, \quad T_1 = 391\,\text{K}, \quad \frac{Q^*}{c_v T_1} = 6.8, \quad T_r = 1580\,\text{K}$$

5.5 FUEL-AIR CYCLE ANALYSIS

A more accurate representation of the properties of the working fluid inside the engine cylinder is to treat the unburned mixture as frozen in composition and the burned gas mixture as in equilibrium. Values for thermodynamic properties for these working fluid models can be obtained with the charts for unburned and burned gas mixtures described in Sec. 4.5 , or computer codes such as those summarized in Sec. 4.7 . When these working fluid models are combined with the ideal engine process models in Table 5.1, the resulting cycles are called fuel-air cycles. [1] The sequence of processes and assumptions are (with the notation of Fig. 5.2):

1-2 Reversible adiabatic compression of a mixture of air, fuel vapor, and residual gas without change in chemical composition.

2-3 Complete combustion (at constant volume or limited pressure or constant pressure), without heat loss, to burned gases in chemical equilibrium.

3-4 Reversible adiabatic expansion of the burned gases which remain in chemical equilibrium.

4-5-6 Ideal adiabatic exhaust blowdown and displacement processes with the burned gases fixed in chemical composition.

6-7-1 Ideal intake process with adiabatic mixing between residual gas and fresh mixture, both of which are fixed in chemical composition.

The basic equations for each of these processes have already been presented in Sec. 5.3 . The use of the charts for a complete engine cycle calculation will now be illustrated.

5.5.1 SI Engine Cycle Simulation

The mixture conditions at point 1 must be known or must be estimated. The following approximate relationships can be used for this purpose: [3]

$$x_r = \left\{1 + \frac{T_r}{T_i}\left[r_c\left(\frac{p_i}{p_e}\right) - \left(\frac{p_i}{p_e}\right)^{(\gamma-1)/\gamma}\right]\right\}^{-1} \tag{5.47}$$

$$T_1 = T_r r_c x_r \left(\frac{p_i}{p_e}\right) \tag{5.48}$$

where $T_r = 1400$ K and $(\gamma - 1)/\gamma = 0.24$ are appropriate average values to use for initial estimates.

Given the equivalence ratio ϕ and initial conditions T_1 (K), $p_1 = p_i$ (Pa), and v_1 (m^3/kg air), the state at point 2 at the end of compression through a volume ratio $v_1/v_2 = r_c$ is obtained from Eq. (4.25 *a*) and the isentropic compression chart (Fig. 4.4). The compression work W_c (J/kg air) is found from Eq. (5.6) with the internal energy change determined from the unburned mixture chart (Fig. 4.3).

The use of charts to relate the state of the burned mixture to the state of the unburned mixture prior to combustion, for adiabatic constant-volume and constant-pressure combustion, has already been illustrated in Sec. 4.5.3 .

For the *constant-volume cycle*,

$$u_3 = u_{s2} + \Delta u^\circ_{f,u} \quad \text{J/kg air} \tag{5.49}$$

where u_{s2} is the sensible internal energy of the unburned mixture at T_2 from Fig. 4.3 and $\Delta u^\circ_{f,u}$ is the internal energy of formation of the unburned mixture [given by Eq. (4.32)]. Since $v_3 = v_2$, the burned gas state at point 3 can be located on the appropriate burned gas chart (Figs. 4.5 to 4.9).

For the *constant-pressure cycle*,

$$h_3 = h_{s2} + \Delta h^\circ_{f,u} \quad \text{J/kg air} \tag{5.50}$$

Since $p_3 = p_2$, the burned gas state at point 3 can be located (by iteration) on the high-temperature burned gas charts, as illustrated by Example 4.5.

For the *limited-pressure cycle*, application of the first law to the mixture between states 2 and 3 *b* gives

$$h_{3b} = u_{3b} + p_3 v_{3b} = u_2 + p_3 v_2 = u_{s2} + \Delta u^{\circ}_{f,u} + p_3 v_2 \qquad \text{J/kg air} \tag{5.51}$$

Since p_3 for a limited-pressure cycle is given, point 3 *b* can be located on the appropriate burned gas chart.

The expansion process 3-4 follows an isentropic line from v_3 to v_4 ($v_4 = v_1$) on the burned mixture charts. Equation (5.9) [or (5.11) or (5.13)] now gives the expansion work W_E. The state of the residual gas at points 5 and 6 in the cycle is obtained by continuing this isentropic expansion from state 4 to $p = p_e$. The residual gas temperature can be read from the equilibrium burned gas chart; the residual gas fraction is obtained from Eq. (5.17) . If values of T_r and x_r were assumed at the start of the cycle calculation to determine T_1, the assumed values can be checked against the calculated values and an additional cycle computation carried out with the new calculated values if required. The convergence is rapid.

The indicated fuel conversion efficiency is obtained from Eq. (5.1) . The indicated mean effective pressure is obtained from Eq. (5.2) . The volumetric efficiency (see Sec. 2.10) for a four-stroke cycle engine is given by

$$\eta_v = \frac{r_c(1 - x_r)}{v_1 \rho_{a,i}(r_c - 1)} \tag{5.52}$$

where $\rho_{a,i}$ is the inlet air density (in kilograms per cubic meter) and v_1 is the chart mixture specific volume (in cubic meters per kilogram of air in the original mixture).

EXAMPLE 5.2

Calculate the performance characteristics of the constant-volume fuel-air cycle defined by the initial conditions of Examples 4.2, 4.3, and 4.5. The compression ratio is 8; the fuel is isooctane and the mixture is stoichiometric;

the pressure and temperature inside the cylinder at the start of compression are 1 atm and 350 K, respectively. The notation of Fig. 5.2*a* is used to define the states at the beginning and end of each process.

Example 4.2 analyzed the compression process:

$$T_1 = 350\ \text{K}, \quad p_1 = 101.3\ \text{kPa}, \quad v_1 = 1\ \text{m}^3/\text{kg air}, \quad u_{s1} = 40\ \text{kJ/kg air}$$
$$T_2 = 682\ \text{K}, \quad p_2 = 1.57\ \text{MPa}, \quad v_2 = 0.125\ \text{m}^3/\text{kg air}, \quad u_{s2} = 350\ \text{kJ/kg air}$$
$$W_{1\text{-}2} = W_C = -310\ \text{kJ/kg air}$$

Example 4.5 analyzed the constant-volume adiabatic combustion process (it was assumed that the residual gas fraction was 0.08):

$$u_{b3} = u_{u2} = u_{s,u2} + \Delta u^\circ_{f,u} = -5\ \text{kJ/kg air}, \quad s_3 = 9.33\ \text{kJ/kg air} \cdot \text{K}$$
$$v_3 = v_2 = 0.125\ \text{m}^3/\text{kg air}, \quad T_3 = 2825\ \text{K}, \quad p_3 = 7100\ \text{kPa}$$

Example 4.3 analyzed the expansion process, from these conditions after combustion at TC, to the volume v_4 at BC of 1 m^3/kg air:

$$T_4 = 1840\ \text{K}, \quad p_4 = 570\ \text{kPa}, \quad u_4 = -1540\ \text{kJ/kg air}$$
$$W_{3\text{-}4} = W_E = 1535\ \text{kJ/kg air}$$

To check the assumed residual gas fraction, the constant entropy expansion process on the chart in Fig. 4.8 is continued from state 4 to the exhaust pressure p_5 of 1 atm = 101.3 kPa. This gives $v_5 = 4.0$ m^3/kg air and $T_5 = 1320$ K. The residual fraction from Eq. (5.17) is

$$x_r = \frac{v_2}{v_5} = \frac{0.125}{4.0} = 0.031$$

which is significantly different from the assumed value of 0.08. The combustion and expansion calculations are now repeated with the new residual fraction of 0.031 (the compression process will not be changed significantly and the initial temperature is assumed fixed):

$$u_{b3} = 350 - 118.2 - 2956 \times 0.031 = 140\ \text{kJ/kg air}$$

With $v_3 = 0.125$ m^3/kg air, Fig. 4.8 gives

$$p_3 = 7270 \text{ kPa}, \quad T_3 = 2890 \text{ K}$$

Expand at constant entropy to $v_4 = 1$ m^3/kgair:

$$p_4 = 595 \text{ kPa}, \quad T_4 = 1920 \text{ K}, \quad u_4 = -1457 \text{ kJ/kg air}$$
$$W_{3\text{-}4} = W_E = 1597 \text{ kJ/kg air}$$

Continue expansion at constant entropy to the exhaust pressure, $p_5 = 1$ atm:

$$v_5 = 4 \text{ m}^3\text{/kg air}, \quad T_5 = 1360 \text{ K}$$

Equation (5.17) now gives the residual fraction

$$x_r = \frac{v_2}{v_5} = \frac{0.125}{4} = 0.031$$

which agrees with the value assumed for the second iteration.

The fuel conversion efficiency can now be calculated:

$$\eta_{f,i} = \frac{W_E + W_C}{m_f Q_{\text{LHV}}}$$

where

$$m_f = \text{kg fuel/kg air at state 1} = \left(\frac{F}{A}\right)(1 - x_r)$$

Thus

$$\eta_{f,i} = \frac{1597 - 310}{0.0661 \times (1 - 0.031) \times 44.4 \times 10^3} = 0.45$$

The indicated mean effective pressure is

$$\text{imep} = \frac{W_E + W_C}{V_d} = \frac{1597 - 310}{1 - 0.125} = 1470 \text{ kPa}$$

or

$$\frac{\text{imep}}{p_1} = 14.6$$

5.5.2 CI Engine Cycle Simulation

With a diesel engine fuel-air cycle calculation, additional factors must be taken into account. The mixture during compression is air plus a small amount of residual gas. At point 2 liquid fuel is injected into the hot compressed air at temperature $T_{2'}$; as the fuel vaporizes and heats up, the air is cooled. For a constant-volume mixing process which is adiabatic overall, the mixture internal energy is unchanged, that is:

$$m_f\left[u_{fg} + c_{v,f}(T_{2'} - T_0)\right] + m_a c_{v,a}(T_{2'} - T_2) = 0 \tag{5.53}$$

where m_f is the mass of fuel injected, u_{fg} is the latent heat of vaporization of the fuel, $c_{v,f}$ cv,f is the specific heat at constant volume of the fuel vapor, $T_{2'}$ is the mixture temperature (assumed uniform) after vaporization and mixing is complete, m_a is the mass of air used, and $c_{v,a}$ is the specific heat at constant volume of air. Substitution of typical values for fuel and air properties gives $(T_{2'} - T_2) \approx 70$ K at full load. Localized cooling in a real engine will be greater.

The limited-pressure cycle is a better approximation to the diesel engine than the constant-pressure or constant-volume cycles. Note that because nonuniformities in the fuel/air ratio exist during and after combustion in the CI engine, the burned gas charts, which assume uniform composition, will not be as accurate an approximation to working fluid properties as they are for SI engines.

5.5.3 Results of Cycle Calculations

Extensive results of constant-volume fuel-air cycle calculations are available. [1, 3, 4] Efficiency is little affected by variables other than the compression ratio r_c and equivalence ratio ϕ. Figures 5.9 and 5.10 show the

effect of variations in these two parameters on indicated fuel conversion efficiency and mean effective pressure. From the available results, the following conclusions can be drawn:

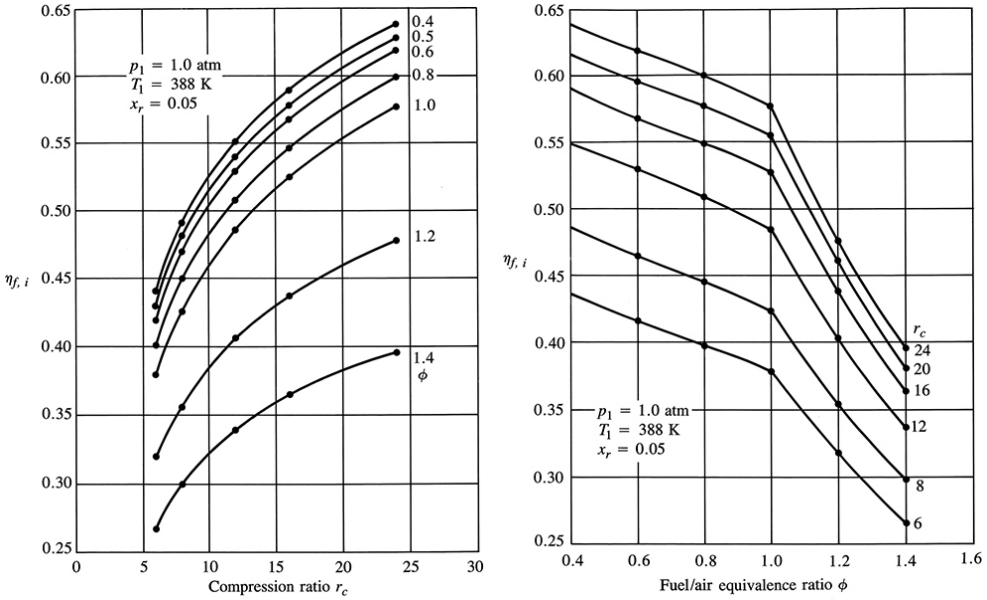


Figure 5.9 Fuel-air cycle results for indicated fuel conversion efficiency as a function of compression ratio and equivalence ratio. Fuel: octene. $p_1 = 1$ atm, $T_1 = 388$ K, $x_r = 0.05$. (*From Edson and Taylor.* [4])

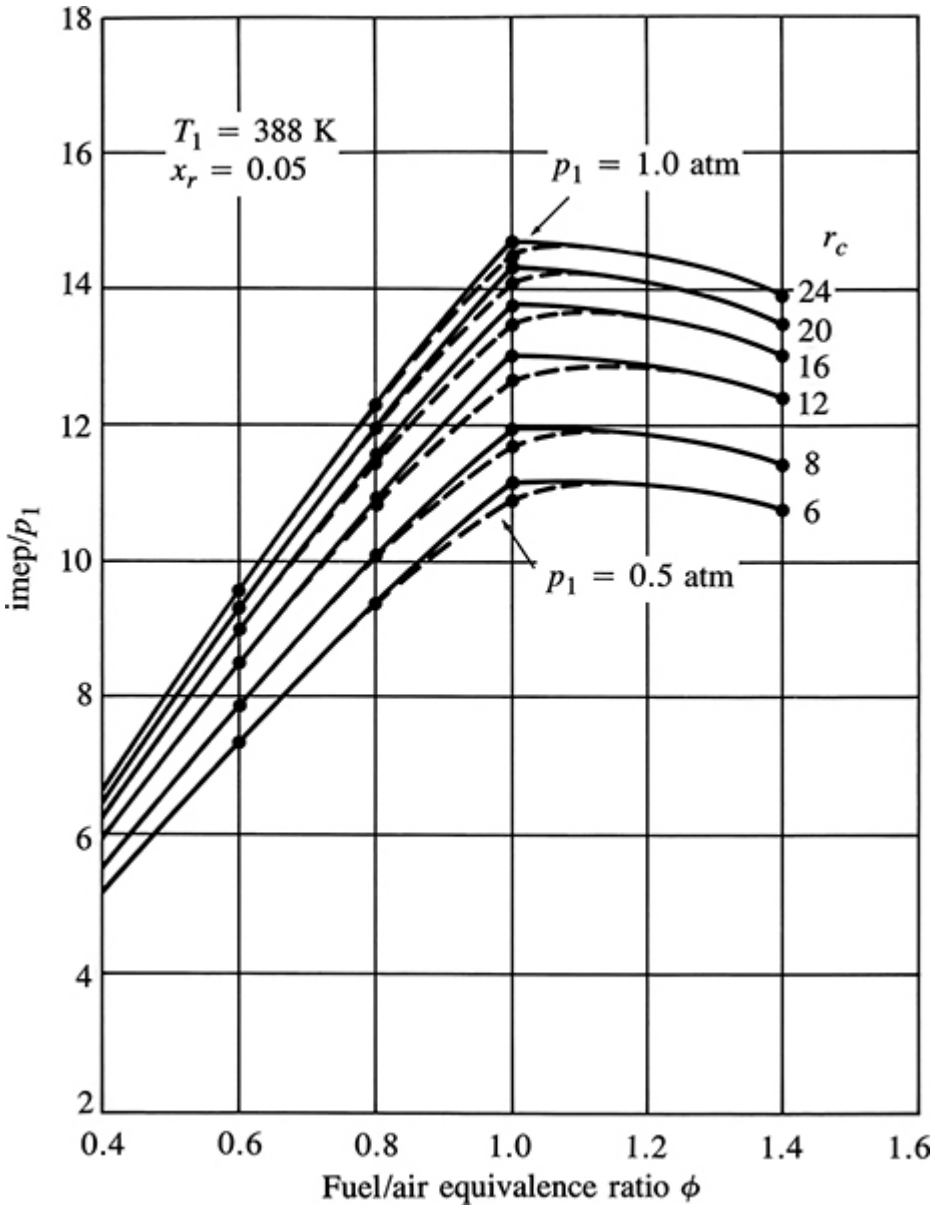


Figure 5.10 Fuel-air cycle results for indicated mean effective pressure as a function of equivalence ratio and compression ratio. Fuel: octene. $p_1 = 1$ atm, $T_1 = 388$ K, $x_r = 0.05$. (*From Edson and Taylor.* [4])

1. The effect of increasing the compression ratio on efficiency at a constant equivalence ratio is similar to that demonstrated by the constant γ constant-volume cycle analysis provided the appropriate value of γ is used. For a stoichiometric mixture, $\gamma = 1.27$ in Eq. (5.31) matches the fuel-air cycle $\phi = 1$ data well.

2. As the equivalence ratio is decreased below unity (i.e., the fuel-air mixture is made progressively leaner than stoichiometric), the efficiency increases. This occurs because the burned gas temperatures after combustion decrease, decreasing the burned gas specific heats and thereby increasing the effective value of γ over the expansion stroke. The efficiency increases because, for a given volume-expansion ratio, the burned gases expand through a larger temperature ratio prior to exhaust; therefore, per unit mass of fuel, the expansion stroke work is increased.
3. As the equivalence ratio increases above unity (i.e., the mixture is made progressively richer than stoichiometric), the efficiency decreases because lack of sufficient air for complete oxidation of the fuel more than offsets the effect of decreasing burned gas temperatures which decrease the mixture's specific heats.
4. The mean effective pressure, from Eq. (5.2) , is proportional to the product $\phi_h f,i$. This product is essentially constant between $\phi \approx 1.0$ and $\phi \approx 1.2$ (i.e., slightly rich of stoichiometric). For ϕ less than the value corresponding to this maximum, the decreasing fuel mass per unit displaced volume more than offsets the increasing fuel conversion efficiency. For ϕ greater than about 1.2, the decreasing fuel conversion efficiency (due to decreasing combustion efficiency) more than offsets the increasing fuel mass.
5. Variations in initial pressure, inlet temperature, residual gas fraction, and atmospheric moisture fraction have only a modest effect on the fuel conversion efficiency. The effects of variations in these variables on imep are more substantial, however, because imep depends directly on the initial charge density. [1]
6. Comparison of results from limited-pressure and constant-volume fuel-air cycles [1] shows that placing a realistic limit on the maximum pressure reduces the advantages of increased compression ratio on both efficiency and imep.

5.6 OVEREXPANDED ENGINE CYCLES

The gas pressure within the cylinder of a conventional four-stroke engine at exhaust valve opening is greater than the exhaust pressure. The available energy of the cylinder gases at this point in the cycle is then dissipated in the exhaust blowdown process. Additional expansion within the engine cylinder would increase the indicated work per cycle, as shown in Fig. 5.11, where expansion continues beyond point 4′ (the conventional ideal cycle exhaust valve opening point) at $V_{4'} = r_c V_c$ to point 4 at $V_4 = r_e V_c$, where r_e is the volume expansion ratio. The exhaust stroke in this overexpanded cycle is 4-5-6. The intake stroke is 6-1. The area 14'451 has been added to the conventional cycle p-V diagram area, for the same fuel input, thereby increasing the engine's efficiency.

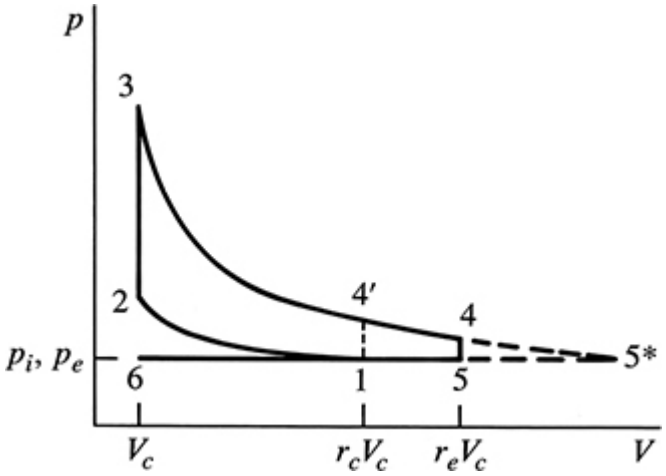


Figure 5.11 Pressure-volume diagram for overexpanded engine cycle (1234561) and Atkinson cycle (1235*61). r_c and r_e are volumetric compression and expansion ratios, respectively.

Complete expansion within the cylinder to exhaust pressure p_e (point 5*) is called the *Atkinson cycle*. Unthrottled operation is shown in Fig. 5.11; throttled operating cycles can also be generated. Many crank and valve mechanisms have been proposed to achieve this additional expansion. For example, it can be achieved in a conventional four-stroke cycle engine by suitable choice of exhaust valve opening and intake valve closing positions relative to BC. If the crank angle between exhaust valve opening and BC on the expansion stroke is less than the crank angle between BC and intake valve closing on the compression stroke, then the actual volumetric

expansion ratio is greater than the actual volumetric compression ratio (these *actual* ratios are both less than the *nominal* compression ratio with normal valve timing).

The effect of overexpansion on efficiency can be estimated from an analysis of the ideal cycle shown in Fig. 5.11. An ideal gas working fluid with specific heats constant throughout the cycle will be assumed. The indicated work per cycle for the overexpanded cycle is

$$W_{c,i} = m\left[(u_3 - u_4) - (u_2 - u_1) - p_1(v_4 - v_1)\right] \tag{5.54}$$

The isentropic relations for 1-2 and 3-4 are

$$\frac{T_2}{T_1} = r_c^{\gamma-1} \qquad \frac{T_3}{T_4} = r_e^{\gamma-1}$$

With Eq. (5.33) to relate T_3 and T_2, the following expression for indicated fuel conversion efficiency can be derived from Eqs. (5.1) , (5.29), and (5.54):

$$\eta_{f,i} = 1 - \frac{1}{(rr_c)^{\gamma-1}}\left\{1 + \frac{c_v T_1}{Q^*} r_c^{\gamma-1}\left[1 - \gamma r^{\gamma-1} + (\gamma - 1)r^{\gamma}\right]\right\} \tag{5.55}$$

where

$$r = \frac{r_e}{r_c}$$

Note that the efficiency given by Eq. (5.55) is a function of load (via Q^*), and is a maximum at maximum load. This contrasts with the ideal constant-volume cycle efficiency [Eq. (5.31)], which is independent of load. The ratio r_e / r_c for complete expansion is given by

$$r^{\gamma} = 1 + \frac{Q^*}{c_v T_1 r_c^{\gamma-1}} \tag{5.56}$$

The effect of overexpansion on fuel conversion efficiency is shown in Fig. 5.12 for $r_c = 4$, 8, and 16 with $\gamma = 1.3$. The ratio of overexpanded cycle

efficiency to the standard cycle efficiency is plotted against r. The Atkinson cycle (complete expansion) values are indicated by the transition from a continuous line to a dashed line. Significant increases in efficiency can be achieved, especially at low compression ratios.

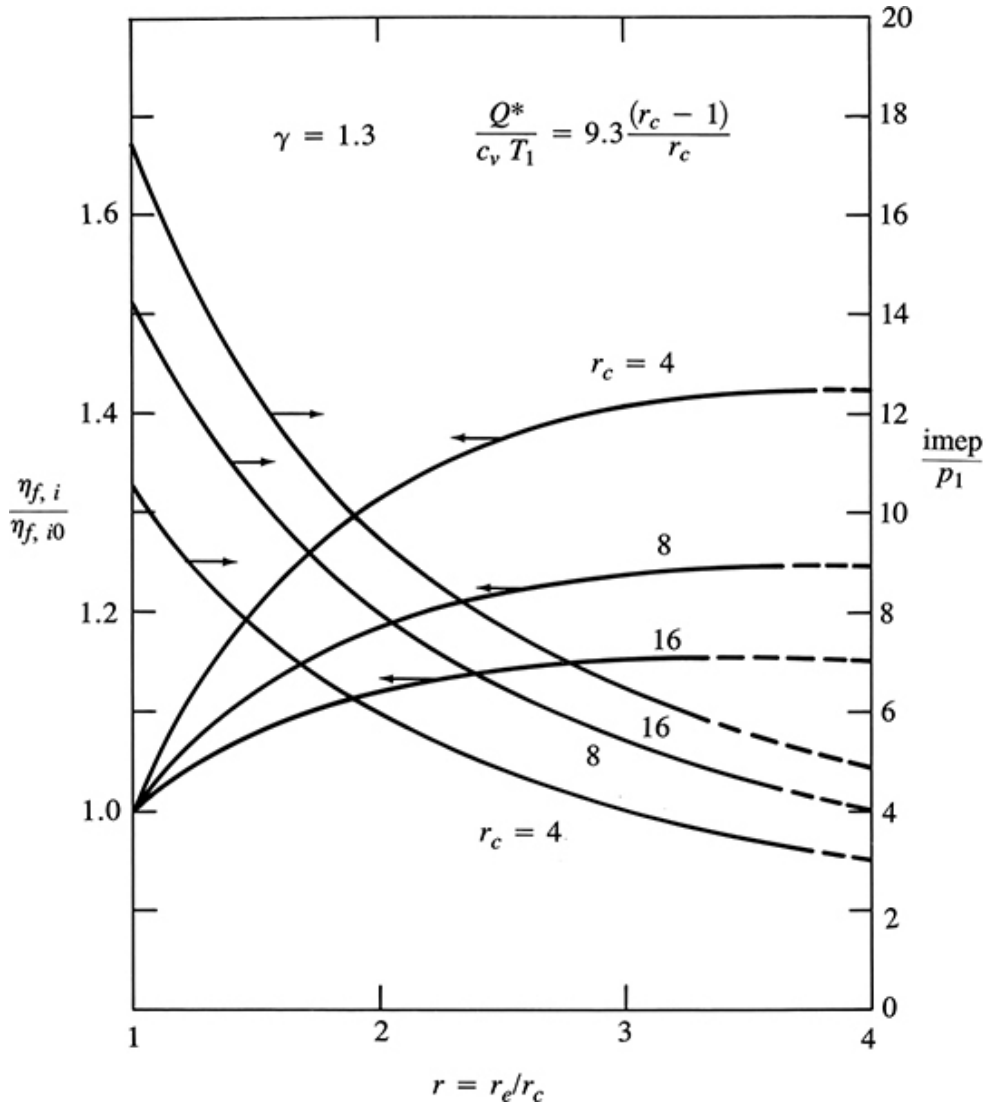


Figure 5.12 Indicated fuel conversion efficiency and mean effective pressure for overexpanded engine cycle as a function of r_e/r_c. Efficiencies given relative to $r_e = r_c$ value, h_f, i_0, $\gamma = 1.3$, $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c$. Solid to dashed line transition marks the complete expansion point (Atkinson cycle).

One major disadvantage of this cycle is that imep and power density decrease significantly because only part of the total displaced volume is filled with fresh charge. From Eqs. (5.2) , (5.29), and the relations $V_d = V_1(r_e - 1)/r_c$ and $p_1V_1 = mRT_1$, it follows that imep for the overexpanded cycle is given by

$$\frac{\text{imep}}{p_1} = \left(\frac{Q^*}{c_v T_1}\right)\left(\frac{1}{\gamma - 1}\right)\left(\frac{r_c}{r_e - 1}\right)\eta_{f,i} \tag{5.57}$$

Values of imep/ p_1 are plotted in Fig. 5.12 as a function of $r(= r_e/ r_c)$. The substantial decrease from the standard constant-volume cycle values at $r = 1$ is clear.

5.7 AVAILABILITY ANALYSIS OF ENGINE PROCESSES

5.7.1 Availability Relationships

Of interest in engine performance analysis is the amount of *useful work* that can be extracted from the gases within the cylinder at each point in the operating cycle. The problem is that of determining the maximum possible work output (or minimum work input) when a system (the charge within the cylinder) is taken from one specified state to another in the presence of a specified environment (the atmosphere). The first and second laws of thermodynamics together define this maximum or minimum work, which is best expressed in terms of the property of such a system-environment combination called *availability* [5] or sometimes *exergy*. [6, 7]

Consider the system-atmosphere combination shown in Fig. 5.13. In the absence of mass flow across the system boundary, as the system changes from state 1 to state 2, the first and second laws give

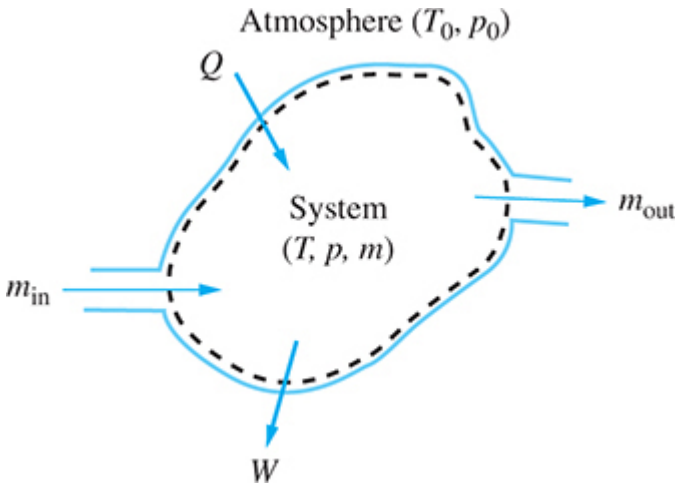


Figure 5.13 System-atmosphere configuration for availability analysis.

$$W_{1\text{-}2} = -(U_2 - U_1) + Q_{1\text{-}2}$$
$$Q_{1\text{-}2} \leq T_0(S_2 - S_1)$$

Combining these two equations gives the *total work* transfer:

$$W_{t,1\text{-}2} \leq -\left[(U_2 - U_1) - T_0(S_2 - S_1)\right] \tag{5.58}$$

The work done by the system against the atmosphere is not available for productive use. It must, therefore, be subtracted from the total work to obtain the *useful work transfer*:

$$W_{U,1\text{-}2} \leq -\left[(U_2 - U_1) + p_0(V_2 - V) - T_0(S_2 - S_1)\right] \tag{5.59}$$

The *maximum* useful work will be obtained when the final state of the system is in thermal and mechanical equilibrium with the atmosphere. [d] The *availability* of this system which is in communication with the atmosphere

$$A = U + p_0V - T_0S \tag{5.60}$$

is thus the property of the system-atmosphere combination which defines its capacity for useful work. The useful work such a system-atmosphere combination can provide, as the system changes from state 1 to state 2, is less than or equal to the change in availability:

$$W_{U,1\text{-}2} \leq -(A_2 - A_1) \tag{5.61}$$

When mass flow across the system boundary occurs, the availability associated with this mass flow is

$$B = H - T_0 S \tag{5.62}$$

B is usually called the *steady-flow availability function*.

With these relations, an availability balance for the gas working-fluid system around the engine cycle can be carried out. For any process between specified end states which this system undergoes (interacting only with the atmosphere), the change in availability ΔA is given by

$$\Delta A = A_{\text{in}} - A_{\text{out}} - A_{\text{destroyed}} \tag{5.63}$$

The availability transfers in and out occur as a result of work transfers, heat transfers, and mass transfers across the system boundary. The availability transfer associated with a work transfer is equal to the work transfer. The availability transfer dA_Q associated with a heat transfer δQ occurring when the system temperature is T is given by

$$dA_Q = \delta Q\left(1 - \frac{T_0}{T}\right) \tag{5.64}$$

since both an energy and entropy transfer occurs across the system boundary. The availability transfer associated with a mass transfer is given by Eq. (5.62) . Availability is destroyed by the irreversibilities that occur in any real process. The availability destroyed is given by

$$A_{\text{destroyed}} = T_0 \Delta S_{\text{irrev}} \tag{5.65}$$

where ΔS_{irrev} is the entropy increase associated with the irreversibilities occurring within the system boundary. [7, 8]

5.7.2 Entropy Changes in Ideal Cycles

The ideal models of engine processes examined earlier in this chapter provide useful illustrative examples for availability analysis. First, however, we will consider the variation in the entropy of the cylinder gases as they proceed through these ideal operating cycles.

For an adiabatic reversible compression process, the entropy is constant. For the combustion process in each of the ideal gas standard cycles, the entropy increase can be calculated from the relations of Eq. (4.14) (with constant specific heats):

$$s - s_0 = c_v \ln\left(\frac{T}{T_0}\right) + R \ln\left(\frac{v}{v_0}\right) = c_p \ln\left(\frac{T}{T_0}\right) - R \ln\left(\frac{p}{p_0}\right)$$

For the constant-volume cycle:

$$S_3 - S_2 = m(s_3 - s_2) = mc_v \ln\left(\frac{T_3}{T_2}\right) \tag{5.66a}$$

For the constant-pressure cycle:

$$S_3 - S_2 = m(s_3 - s_2) = mc_p \ln\left(\frac{T_3}{T_2}\right) \tag{5.66b}$$

For the limited-pressure cycle:

$$S_{3b} - S_2 = mc_v \ln\left(\frac{T_{3a}}{T_2}\right) + mc_p \ln\left(\frac{T_{3b}}{T_{3a}}\right) = mc_v \ln \alpha + mc_p \ln \beta \tag{5.66c}$$

with α and β defined by Eq. (5.42) .

Since the expansion process, after combustion is complete, is adiabatic and reversible, there is no further change in entropy, 3 to 4 (or 3 *b* to 4). Figure 5.14 shows the entropy changes that occur during each process of these three ideal engine operating cycles, calculated from the above equations, on a *T*-*s* diagram. The three cycles shown correspond to those of the *p*-*V* diagrams of Fig. 5.6 with $r_c = 12$, $\gamma = 1.3$, and $Q^*/(c_v T_1) = 8.525$. Since the combustion process was assumed to be adiabatic, the increase in

entropy during combustion clearly demonstrates the irreversible nature of this process.

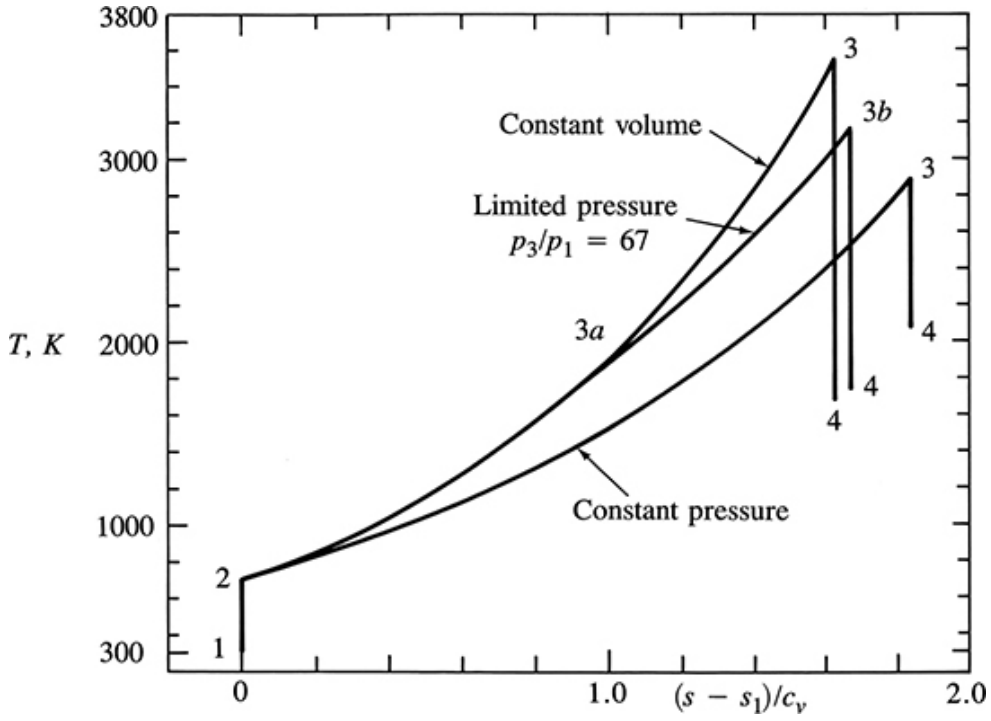


Figure 5.14 Temperature-entropy diagram for ideal gas constant-volume, constant-pressure, and limited-pressure cycles. Assumptions same as in Fig. 5.6.

5.7.3 Availability Analysis of Ideal Cycles

An availability analysis for each process in the ideal cycle illustrates the magnitude of the availability transfers and where the losses in availability occur. [9] In general, for the system of Fig. 5.4 in communication with an atmosphere at p_0, T_0 as indicated in Fig. 5.13, the change in availability between states i and j during the portion of the cycle when the valves are closed is given by

$$A_j - A_i = m(a_j - a_i) = m\left[(u_j - u_i) + p_0(v_j - v_i) - T_0(s_j - s_i)\right] \tag{5.67}$$

The appropriate normalizing quantity for these changes in availability is the availability of the fuel supplied to the engine cylinder each cycle, $m_f(-\Delta g$

$_{298})$ [e] (see Sec. 3.6.2). However, it is more convenient to use $m_f(-\Delta h_{298})$ [f] $= m_f Q_{\text{LHV}}$ as the normalizing quantity since it can be related to the temperature rise during combustion via Eq. (5.28) . As shown in Table 3.3, these two quantities differ by only a few percent for common hydrocarbon fuels. Equation (5.67) , with Eq. (5.29) , then becomes

$$\frac{A_j - A_i}{m_f Q_{\text{LHV}}} = \frac{m(a_j - a_i)}{m_f Q_{\text{LHV}}} = \frac{a_j - a_i}{Q^*} \tag{5.68}$$

The compression process is isentropic, so:

$$\begin{aligned} \frac{A_2 - A_1}{m_f Q_{\text{LHV}}} &= \frac{a_2 - a_1}{Q^*} = \frac{(u_2 - u_1) + p_0(v_2 - v_1)}{Q^*} \\ &= \frac{c_v T_1}{Q^*}\left[\left(\frac{T_2}{T_1} - 1\right) - (\gamma - 1)\left(1 - \frac{V_2}{V_1}\right)\right] \\ &= \frac{c_v T_1}{Q^*}\left[(r_c^{\gamma} - 1) - (\gamma - 1)\left(1 - \frac{1}{r_c}\right)\right] \end{aligned} \tag{5.69}$$

where we have assumed $p_0 = p_1$. The first term in the square brackets is the compression stroke work transfer. The second term is the work done by the atmosphere on the system, which is subtracted because it does not increase the *useful work* which the system-atmosphere combination can perform.

During combustion, for the constant-volume cycle, the volume and internal energy remain unchanged [Eqs. (5.7 *a, b*)]. Thus

$$\begin{aligned} \frac{A_3 - A_2}{m_f Q_{\text{LHV}}} &= \frac{a_3 - a_2}{Q^*} = -\frac{T_0(s_3 - s_2)}{Q^*} \\ &= -\frac{c_v T_0}{Q^*} \ln\left(\frac{T_3}{T_2}\right) = -\frac{c_v T_0}{Q^*} \ln\left(1 + \frac{Q^*}{c_v T_1 r_c^{\gamma - 1}}\right) \end{aligned} \tag{5.70}$$

This loss in availability results from the increase in entropy associated with the irreversibilities of the combustion process. This lost or destroyed availability, as a fraction of the initial availability of the fuel-air mixture, decreases as the compression ratio increases (since T_2 increases as the

compression ratio increases, T_3/T_2 decreases for fixed heat addition) and increases as Q^* decreases [e.g., when the mixture is made leaner; see Eq. (5.46)]. The changes in availability during combustion for the constant-pressure and limited-pressure cycles are more complex because there is a transfer of availability out of the system equal to the expansion work transfer which occurs.

For the constant-volume cycle expansion stroke:

$$\begin{aligned}\frac{A_4 - A_3}{m_f Q_{\mathrm{LHV}}} &= \frac{a_4 - a_3}{Q^*} = \frac{(u_4 - u_3) + p_0(v_4 - v_3)}{Q^*} \\ &= \frac{c_v T_3}{Q^*}\left[\left(\frac{T_4}{T_3} - 1\right) + (\gamma - 1)\left(\frac{p_0}{p_3}\right)\left(\frac{V_4}{V_3} - 1\right)\right] \\ &= \left[\left(1 + \frac{c_v T_1 r_c^{\gamma-1}}{Q^*}\right)\left(1 - \frac{1}{r_c^{\gamma-1}}\right) - (\gamma - 1)\frac{c_v T_1}{Q^*}\left(\frac{r_c - 1}{r_c}\right)\right]\end{aligned} \tag{5.71}$$

The availability of the exhaust gas at state 4 relative to its availability at (T_1, p_1) is given by

$$\frac{A_4 - A_1}{m_f Q_{\mathrm{LHV}}} = \frac{c_v T_1}{Q^*}\left[\left(\frac{T_4}{T_1} - 1\right) - \frac{T_0}{T_1}\ln\left(\frac{T_4}{T_1}\right)\right] \tag{5.72}$$

The availability of the gases inside the cylinder relative to their availability at (T_1, p_1) over the compression and expansion strokes of the constant-volume operating cycle example used in Figs. 5.6 and 5.14 is shown in Fig. 5.15. Equations (5.69) and (5.71), with T_2 and T_4 replaced by the appropriate temperatures intermediate between T_1 and T_2 and T_3 and T_4, respectively, were used to compute the variations during compression and expansion. Table 5.3 summarizes the changes in availability during each process and the availability of the cylinder gases, at the beginning and end of each process, relative to the datum for the atmosphere (1 atm, 300 K). The availability at state 1 of the fuel, air, residual-gas mixture is (1.0286 + 0.0008) $m_f Q_{\mathrm{LHV}}$. 1.0286 is the ratio $(-\Delta g^\circ_{298})/(-\Delta h^\circ_{298})$ for isooctane (see Table 3.3). The second number, 0.0008, allows for the difference between T_1 and T_0. Because both work-transfer processes in this ideal cycle case are

reversible, the fuel conversion efficiency $\eta_{f,i}$ is given by $(A_3 - A_4)/(m_f Q_{LHV}) - (A_2 - A_1)/(m_f Q_{LHV})$. It is, of course, equal to the value obtained for $r_c = 12$ and $\gamma = 1.3$ from the formula for efficiency Eq. (5.31) , obtained previously. The availability conversion efficiency is $\eta_{f,i}/1.0286$. Note that it is the availability destroyed during combustion, plus the inability of this ideal constant-volume cycle to use the availability remaining in the gas at state 4, that decrease the availability conversion efficiency below unity. Both these loss mechanisms decrease in magnitude, relative to the fuel availability, as the compression ratio increases. This is the fundamental reason why engine indicated efficiency increases with an increasing compression ratio.

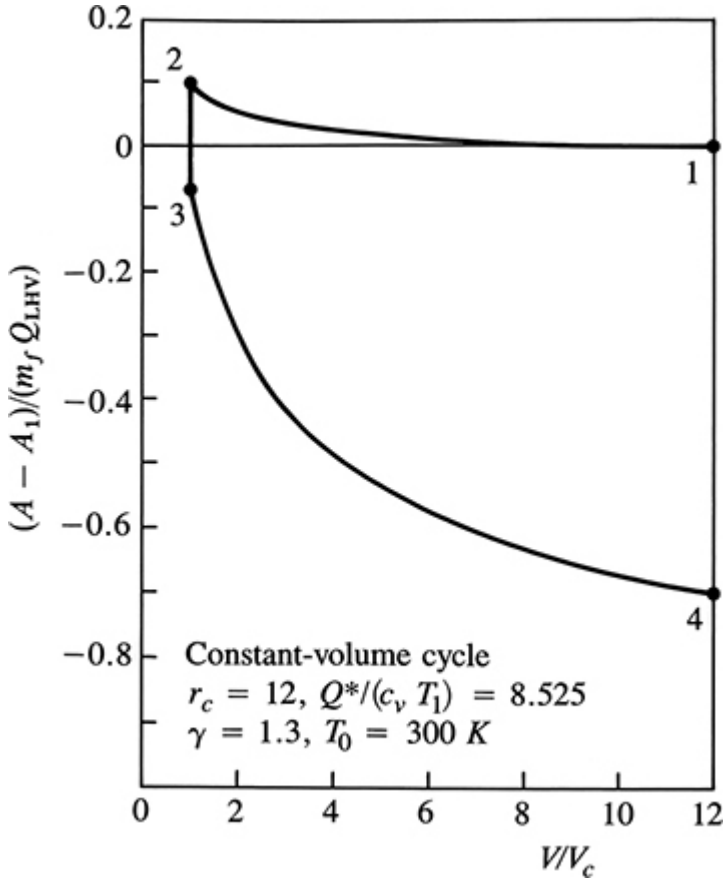


Figure 5.15 Availability of cylinder charge relative to availability at state 1 for constant-volume ideal gas cycle as a function of cylinder volume. Availability made dimensionless by $m_f Q_{LHV}$. Assumptions as in Fig. 5.6.

TABLE 5.3 Availability changes in constant-volume cycle

Process or state	$\frac{A_j - A_i}{m_f Q_{LHV}}$	$\frac{A_i}{m_f Q_{LHV}}$
1		1.0294
1-2	0.0976	
2		1.1270
2-3	−0.1710	
3		0.9560
3-4	−0.6237	
4		0.3323
Fuel conversion efficiency $\eta_{f,i}$	0.526	
Availability conversion efficiency $\eta_{a,i}$	0.511	

$r_c = 12$, $\gamma = 1.3$, $Q^*/(c_v T_1) = 8.525$, $T_0 = 300\,\text{K}$, $T_1 = 333\,\text{K}$.

5.7.4 Effect of Equivalence Ratio

The fuel-air cycle with its more accurate models for working fluid properties can be used to examine the effect of variations in the fuel/air equivalence ratio on the availability conversion efficiency. Figure 5.16 shows the temperature attained and the entropy rise that occurs in constant-volume combustion of a fuel-air mixture of different equivalence ratios, following isentropic compression from ambient temperature and pressure through different volumetric compression ratios.[8] The entropy increase is the result of irreversibilities in the combustion process and mixing of complete combustion products with excess air. The significance of these combustion-related losses the destruction of availability that occurs in this process is shown in Fig. 5.17 where the availability after constant-volume combustion divided by the availability of the initial fuel-air mixture is shown as a function of equivalence ratio for compression ratios of 12 and 36. [8] The loss of availability increases as the equivalence ratio decreases below 1.0. [g] The combustion loss is a stronger function of the rise in temperature and pressure which occurs than of the change in the specific heat ratio that occurs.

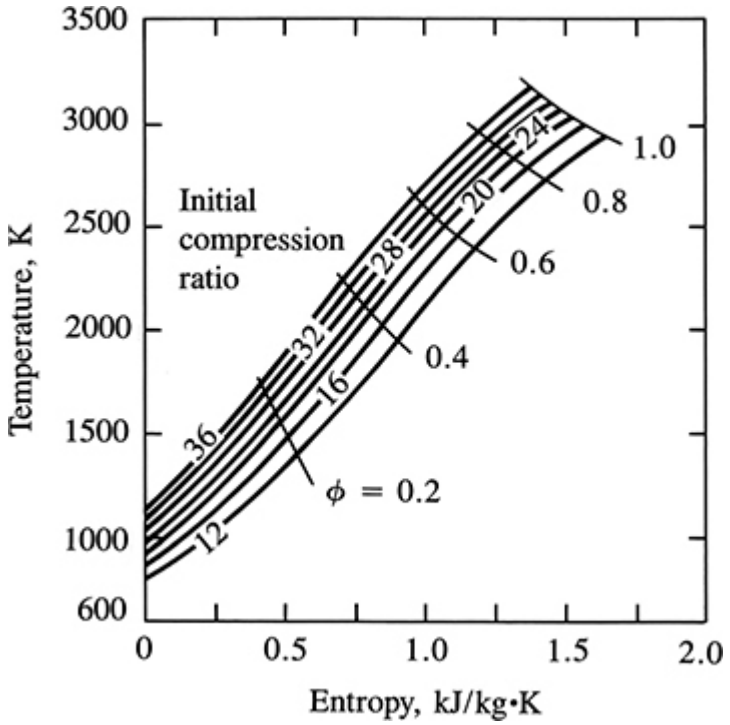


Figure 5.16 Temperature and entropy of combustion products after constant-volume combustion following isentropic compression from ambient conditions through specified compression ratio as a function of compression ratio and equivalence ratio. (*From Flynn et al.* [8])

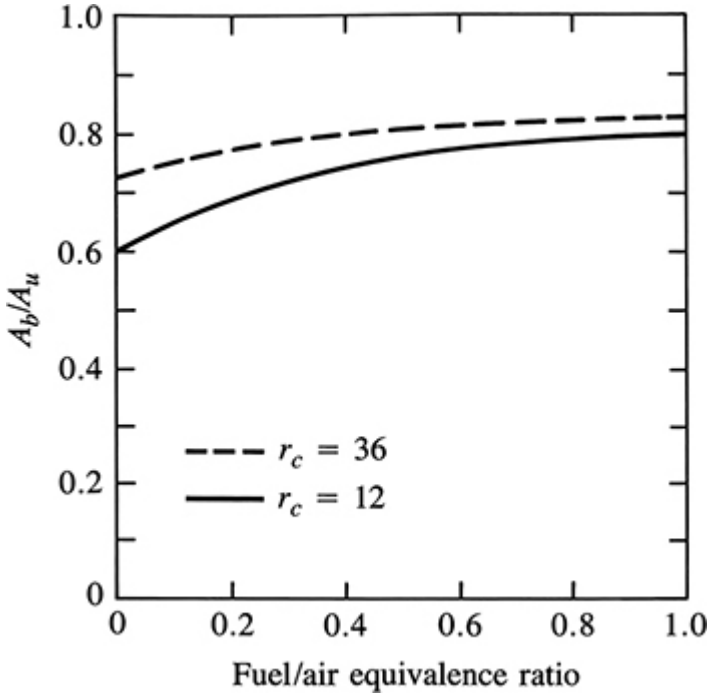


Figure 5.17 Availability of combustion products after constant-volume combustion relative to availability before combustion following isentropic compression from ambient through specified compression ratio as a function of equivalence ratio. (*From Flynn et al.* [8])

Why then does engine efficiency increase with a decreasing equivalence ratio as shown in Fig. 5.9? The reason is that the expansion stroke work transfer, as a fraction of the fuel availability, increases as the equivalence ratio decreases; hence, the availability lost in the exhaust process, again expressed as a fraction of the fuel availability, decreases. This increase in the expansion stroke work as a fraction of the fuel availability, as the equivalence ratio decreases, more than offsets the increase in the availability lost during combustion; so the availability conversion efficiency (or the fuel conversion efficiency which closely approximates it) increases.

5.8 COMPARISON WITH REAL ENGINE CYCLES

To put these ideal models of engine processes in perspective, this chapter will conclude with a brief discussion of the additional effects which are important in real engine processes.

A comparison of a real engine p-V diagram over the compression and expansion strokes with an equivalent fuel-air cycle analysis is shown in Fig. 5.18. [4] The real engine and the fuel-air cycle have the same geometric compression ratio, fuel chemical composition and equivalence ratio, residual fraction, and mixture density before compression. Midway through the compression stroke, the pressure in the fuel-air cycle has been made equal to the real cycle pressure. [h] The compression stroke pressures for the two cycles essentially coincide. Modest differences in pressure during intake and the early part of the compression process result from the pressure drop across the intake valve during the intake process and the closing of the intake valve 40 to 60° after BC in the real engine. The expansion stroke pressures for the engine fall below the fuel-air cycle pressures for the following reasons: (1) heat transfer from the burned gases to the walls; (2) finite time

required to burn the charge; (3) exhaust blowdown loss due to opening the exhaust valve before BC; (4) gas flow into crevice regions and leakage past the piston rings; (5) incomplete combustion of the charge.

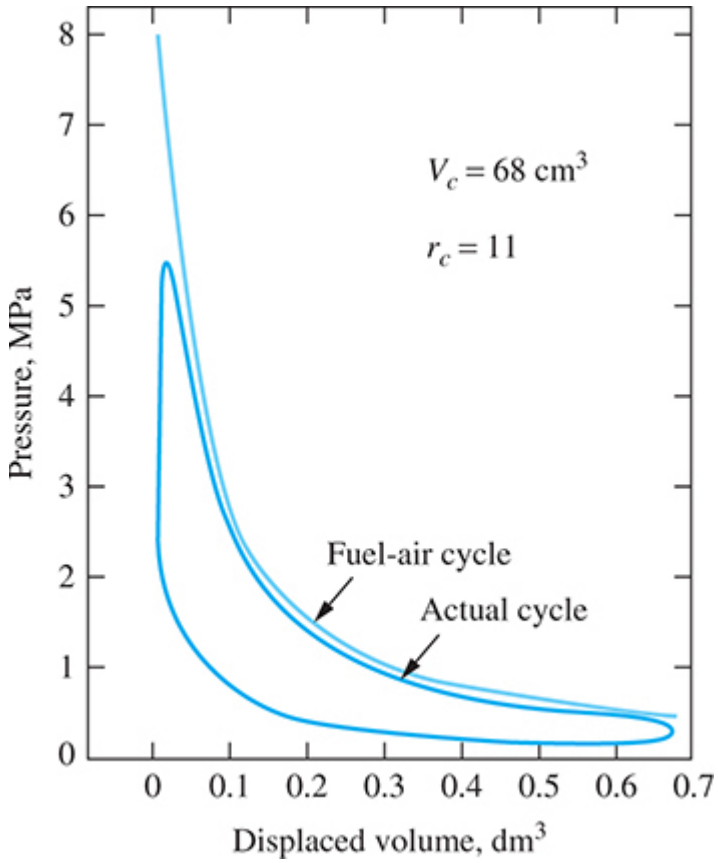


Figure 5.18 Pressure-volume diagram for actual spark-ignition engine compared with that for equivalent fuel-air cycle. $r_c = 11$. (*From Edson and Taylor.* [4])

These differences, in decreasing order of importance, are described below. Together, they contribute to the enclosed area on the *p-V* diagram for a properly adjusted engine with optimum timing being about 85% of the enclosed area of an equivalent fuel-air cycle *p-V* diagram. The indicated fuel conversion or availability conversion efficiency of the actual engine is therefore about 0.85 times the efficiency calculated for the fuel-air cycle. Use is often made of this ratio to estimate the performance of actual engines from fuel-air cycle results. [i] To estimate the imep of actual engines from the fuel-air cycle imep/ p_1 data in Fig. 5.10, an additional adjustment is required for

the mass of mixture in the cylinder. The indicated efficiency is the ratio of work transferred to the piston to the in-cylinder fuel chemical energy. If the actual engine in-cylinder mass is different from the in-cylinder mass in the equivalent fuel-air cycle, the work produced will be different by about the same percentage, so mass discrepancies cancel out. However, imep—the normalized work per cycle—is a performance parameter which does depend on the amount of fuel chemical energy in the cylinder. Mixture temperature and pressure, and cylinder volume, at time of intake valve closing affect the mass trapped inside the cylinder. So differences between values in the actual engine and the equivalent fuel-air cycle will affect the actual engine imep, and adjustments should be made. Typically imep (actual engine) is about 0.85 to 1.0 times the imep of the equivalent fuel-air cycle. The lower end of the range corresponds to part load (lower imep) levels; the higher end of the range corresponds to higher loads.

1. *Heat transfer*. Heat transfer from the unburned mixture to the cylinder walls has a negligible effect on the *p-V* line for the compression process. Heat transfer from the burned gases is much more important (see Chap. 12). Due to heat transfer during combustion, the pressure at the end of combustion in the real cycle will be lower. During expansion, heat transfer will cause the gas pressure in the real cycle to fall below an isentropic expansion line as the volume increases. A decrease in efficiency results from this heat loss.
2. *Finite combustion time*. In an SI engine with spark-timing adjusted for optimum efficiency, combustion typically starts 10 to 40 crank angle degrees before TC, is half complete at about 8° after TC, and is essentially complete 30 to 40° after TC. Peak pressure occurs at about 13 to 15° after TC (see Fig. 1.9). In a diesel engine, the burning process starts shortly before TC. The pressure rises rapidly to a peak some 5 to 10° after TC since the initial rate of burning is fast. However, the final stages of burning are much slower, and combustion continues until 40 to 50° after TC (see Fig. 1.24). Thus, the peak pressure in the engine is substantially below the fuel-air cycle peak pressure value because combustion continues until well after TC when the cylinder volume is much greater than the clearance volume. After peak pressure, expansion stroke pressures in the engine

are higher than fuel-air cycle values in the absence of other loss mechanisms because less work has been extracted from the cylinder gases. A comparison of the constant-volume and limited-pressure cycles in Fig. 5.6 demonstrates this point.

For spark or fuel-injection timing which is retarded from the optimum for maximum efficiency, the peak pressure in the real cycle will be lower, and expansion stroke pressures after the peak pressure will be higher than in the optimum timing cycle.

3. *Exhaust blowdown loss*. In the real engine operating cycle, the exhaust valve is opened some 50° before BC to reduce the pressure during the first part of the exhaust stroke in four-stroke engines, and to allow time for scavenging in two-stroke engines. The gas pressure at the end of the expansion stroke is therefore reduced below the isentropic line. A decrease in expansion-stroke work transfer results.
4. *Crevice effects and leakage*. As the cylinder pressure increases, gas flows into crevices such as the regions between the piston, piston rings, and cylinder wall. These crevice regions can comprise a percent or so of the clearance volume. This flow reduces the mass in the volume above the piston crown, and this flow is cooled by heat transfer to the crevice walls. In premixed charge engines, some of this crevice gas is unburned and some of it will not burn. Though much of this gas returns to the cylinder later in the expansion, a fraction, from behind and between the piston rings, flows into the crankcase. However, leakage in a well-designed and maintained engine is small (usually less than one percent of the charge). All these effects reduce the cylinder pressure during the latter stages of compression, during combustion, and during expansion below the value that would result if crevice and leakage effects were absent.
5. *Incomplete combustion*. Combustion of the cylinder charge is incomplete; the exhaust gases contain combustible species. For example, in spark-ignition engines the hydrocarbon emissions from a warmed-up engine (which come largely from the crevice regions) are about 2% of the fuel mass under normal operating conditions; carbon monoxide and hydrogen in the exhaust contain an additional 2 to 3% or more of the fuel energy, even with excess air present (see Sec. 4.9). Hence, the chemical energy of the fuel which is released in the

actual engine is about 5% less than the chemical energy of the fuel inducted, that is, the combustion efficiency (see Sec. 3.5.5) is about 95%. The fuel-air cycle pressures after combustion will be higher than engine pressures because complete combustion is assumed. In diesel engines, the combustion inefficiency is usually less, about 1 to 2%, so this effect is smaller.

Summary

The effect of all these loss mechanisms on engine efficiency is best defined by an availability balance for the real engine cycle. A limited number of such calculations have been published (e.g., Refs. 8, 10, and 11). Table 5.4 shows the magnitude of the loss in availability (as a fraction of the initial availability) that occurs due to real cycle effects in a typical naturally-aspirated diesel engine.[10] The combustion and exhaust losses are present in the ideal cycle models also (they are smaller, however[9]). The loss in availability due to heat losses, flow or aerodynamic losses, and mechanical friction are real engine effects.

TABLE 5.4 Availability losses in naturally-aspirated diesel

Loss mechanism	Loss, fraction of fuel availability
Combustion	0.225
Exhaust	0.144
Heat transfer	0.135
Aerodynamic	0.047
Mechanical friction	0.048
Total losses	0.599
Availability conversion efficiency (brake)	0.401

Source: Traupel.[10]

Figure 5.19 shows standard and fuel-air cycle efficiencies as a function of the compression ratio compared with engine indicated efficiency data. The

top three sets of engine data are for the best efficiency air/fuel ratio. Differences in the data are in part due to different fuels [(12) isooctane; (13) gasoline; (14) propane] which affect efficiency slightly through their different composition and heating values (see Table D.4). They also result from different combustion chamber shapes which affect the combustion rate and heat transfer. The trends in the data with increasing compression ratio and the $\phi = 0.8$ fuel-air cycle curve (which corresponds approximately to the actual air/fuel ratios used) are similar. The factor of 0.8 relating these older engine and fuel-air cycle efficiencies holds approximately. At compression ratios above about 15, however, the data show that the indicated efficiency of actual engines becomes essentially constant. Increasing crevice and heat losses offset the calculated ideal cycle efficiency increase as the compression ratio is raised above this value. The standard ideal gas cycle analysis results, with an appropriate choice for the value of γ(1.25 to 1.3), correspond closely to the fuel-air cycle analysis results.

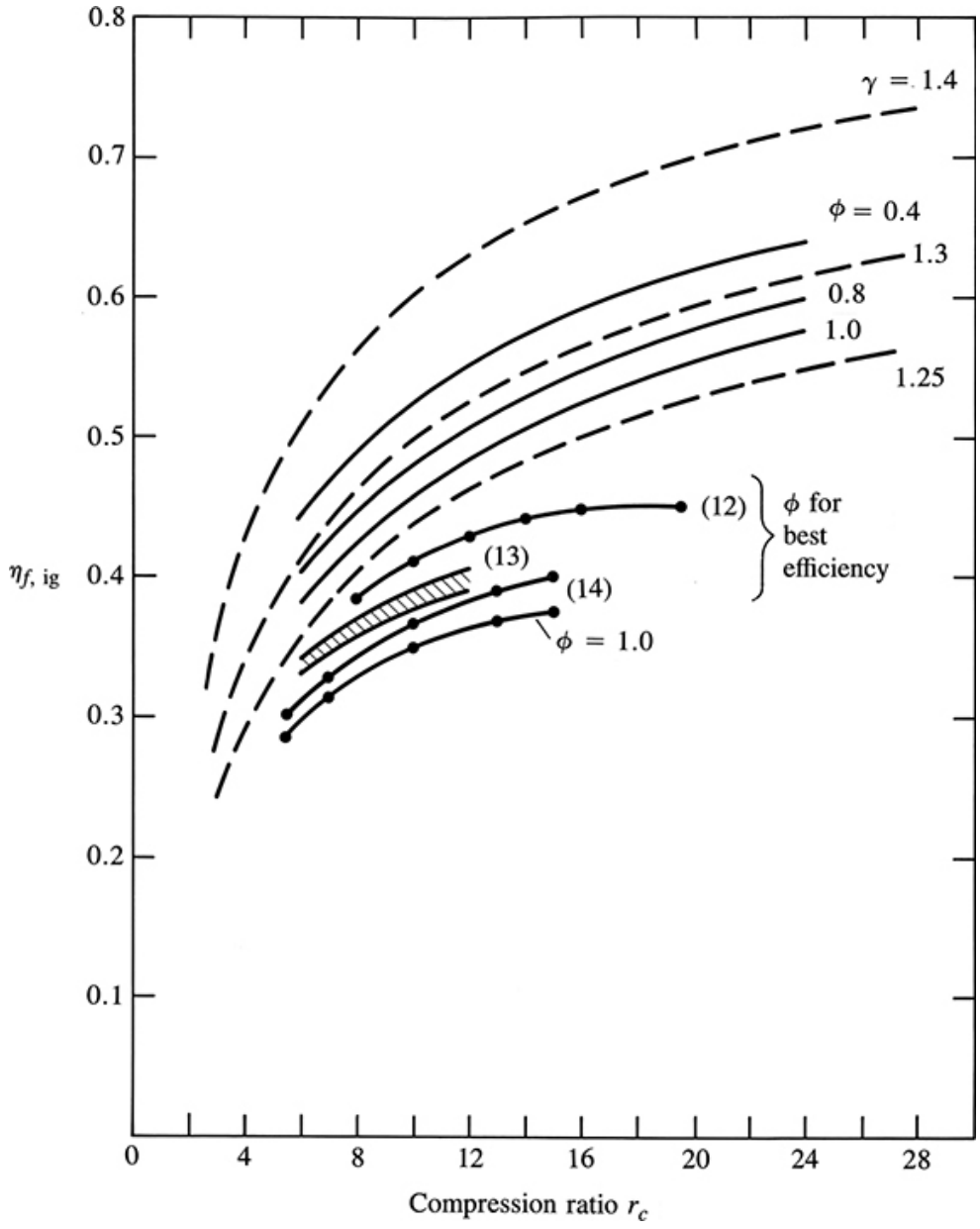


Figure 5.19 Indicated fuel conversion efficiency as a function of compression ratio for ideal gas constant-volume cycle (dashed lines, γ = 1.25, 1.3, 1.4) and fuel-air cycle (solid lines, ϕ = 0.4, 0.8, 0.1). Also shown are available engine data for equivalence ratios given: best efficiency ϕ; [12–14] $\phi = 1$. [14]

The ideal cycle provides a convenient but crude approximation to the real engine operating cycle. It is useful for illustrating the thermodynamic principles of engine operation. It can also provide approximate estimates of

trends as major engine parameters change. The weakest link in these ideal cycles is the modeling of the combustion processes in SI and CI engines. None of the models examined in this chapter are sufficiently close to reality to provide accurate predictions of engine performance. More sophisticated models of the spark-ignition and diesel engine operating cycles have been developed and are discussed in Chap. 14 .

PROBLEMS

5.1 Many diesel engines can be approximated by a limited-pressure cycle. In a limited-pressure cycle, a fraction of the fuel is burnt at constant volume and the remaining fuel is burnt at constant pressure. Use this cycle approximation with $\gamma = c_p / c_v = 1.3$ to analyze the following problem:

Inlet conditions:	$p_1 = 1.0$ bar, $T_1 = 289$ K
Compression ratio:	15:1
Heat added during combustion:	43,000 kJ/kg of fuel
Overall fuel/air ratio:	0.045 kg fuel/kg air

(a) Half of the fuel is burnt at constant volume, then half at constant pressure. Draw a *p-V* diagram and compute the fuel conversion efficiency of the cycle.

(b) Compare the efficiency and peak pressure of the cycle with the efficiency and peak pressure that would be obtained if all of the fuel were burnt at constant pressure or at constant volume.

5.2 It is desired to increase the output of a spark-ignition engine by either (1) raising the compression ratio from 8 to 10 or (2) increasing the intake pressure from 1.0 bar to 1.5 bar. Using the constant-volume cycle as a model for engine operation, which procedure will give:

(a) The highest cycle pressure?

(b) The highest efficiency?

(c) The highest mep?

Assume $\gamma = 1.3$ and $(m_f Q_{HV})/(mc_{v\,T\,1}) = 9.3(r_c = 1)/r_c$.

5.3 **(a)** Use the ideal gas cycle with constant-volume combustion to describe the operation of an SI engine with a compression ratio of 9. Find the pressure and temperature at points 2, 3, 4, and 5 on Fig. 5.2 *a*. Assume a pressure of 100 kPa and a temperature of 320 K at point 1. Assume $m_f/m = 0.06$, $c_v = 946$ J/kg · K, $\gamma = 1.3$. Q_{LHV} for gasoline is 44 MJ/kg.

(b) Find the indicated fuel conversion efficiency and imep for this engine under these operating conditions.

5.4 A spark-ignition engine is throttled when operating at part load (the inlet pressure is reduced) while the fuel/air ratio is held essentially constant. Part-load operation of the engine is modeled by the cycle shown in Fig. 5.2 *d*; the inlet air is at pressure p_1, the exhaust pressure is atmospheric p_a, and the ambient temperature is T_a. Derive an expression for the *decrease* in net indicated fuel conversion efficiency due to throttling from the ideal constant-volume cycle efficiency, and show that it is proportional to $(p_a/p_1 - 1)$. Assume mass fuel ≪ mass air.

5.5 Use a limited-pressure cycle analysis to obtain a plot of indicated fuel conversion efficiency versus p_3/p_1 for a compression ratio of 15 with light diesel oil as fuel. Assume $m_f/m = 0.04$, $T_1 = 45°C$. Use $\gamma = 1.3$ and $c_v = 946$ J/kg · K.

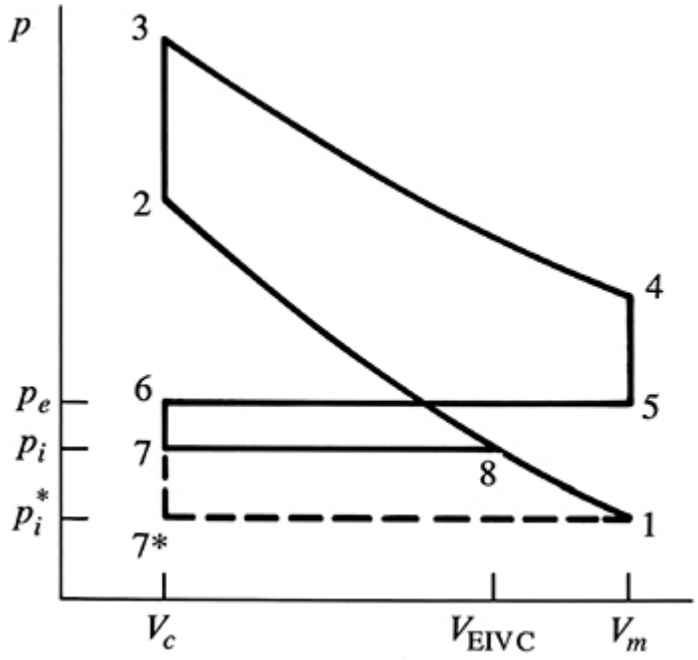
5.6 Explain why constant-volume combustion gives a higher indicated fuel conversion efficiency than constant-pressure combustion for the same compression ratio.

5.7 Two engines are running at a bmep of 250 kPa. One is an SI engine with the throttle partially closed to maintain the correct load. The second engine is a naturally-aspirated CI engine which requires no throttle. Mechanical friction mep for both engines is 100 kPa. If the intake manifold pressures for the SI and CI engines are 25 kPa and 100 kPa, respectively, and both exhaust manifold pressures are 105 kPa, use an ideal cycle model to estimate and compare the *gross* and *net* imep of the two engines. You may neglect the pressure drop across the valves during the intake and exhaust processes.

5.8 **(a)** Plot net imep versus p_i for 20 kPa < p_i < 100 kPa for a constant-volume cycle using the following conditions: $m_f/m = 0.06$, $T_1 = 40°C$, $c_v = 946$ J/kg · K, $\gamma = 1.3$, $r_c = 9.5$, $Q_{LHV} = 44$ MJ/kg fuel. Assume $p_e = 100$ kPa.
(b) What additional information is necessary to draw a similar plot for the engine's indicated torque, and indicated power?

5.9 **(a)** Draw a diagram similar to those in Fig. 5.2 for a supercharged cycle with constant-pressure combustion.
(b) Use the ideal gas cycle with constant-pressure combustion to model an engine with a compression ratio of 14 through such a supercharged cycle. Find the pressure and temperature at points corresponding to 2, 3, 4, and 5 in Fig. 5.2. Assume a pressure of 200 kPa and temperature of 325 K at point 1, and a pressure of 100 kPa at points 5 and 6. $m_f/m = 0.03$ and the fuel is a light diesel oil.
(c) Calculate the gross and net indicated fuel conversion efficiency and imep for this engine under these operating conditions.

5.10 One method proposed for reducing the pumping work in throttled spark-ignition engines is *early intake valve closing* (EIVC). The ideal cycle *p-V* diagram shown illustrates the concept. The EIVC cycle is 1-2-3-4-5-6-7-8-1 (the conventional throttled cycle is 1-2-3-4-5-6-7*-1). With EIVC, the inlet manifold is held at a pressure p_i (which is higher than the normal engine intake pressure, p_i^*), and the inlet valve is closed *during* the inlet stroke at 8. The trapped fresh charge and residual is then *expanded* to the normal cycle (lower) intake pressure, p_i^*. You can assume that both cycles have the same mass of gas in the cylinder, temperature, and pressure at state 1 of the cycle.
(a) On a sketch of the intake and exhaust process *p-V* diagram, shade in the area that corresponds to the difference between the pumping work of the EIVC cycle and that of the normal cycle.
(b) What value of p_i and V_{EIVC} will give the maximum reduction in pumping work for the EIVC cycle?
(c) Derive an expression for this maximum difference in pumping work between the normal cycle and the EIVC cycle in terms of p_e, p_i^*, V_c, and V_m. You can make the appropriate ideal cycle assumptions.



5.11 Calculate the following parameters for a constant-volume fuel-air cycle (Fig. 5.2*a*):

(a) The pressures and temperatures at states 1, 2, 3, 4, 5, and 6

(b) The indicated fuel conversion efficiency

(c) The imep

(d) The residual gas fraction

(e) The volumetric efficiency

Inlet pressure = 1 atm, exhaust pressure = 1 atm, inlet temperature = 300 K, compression ratio = 8:1, equivalence ratio = $\phi = 1$.

Calculate the above parameters (points a-e) using the SI unit's charts. Use 44.4 MJ/kg for heating value of the fuel. *Hint:* Start the calculations using the residual mass fraction 0.03 and the residual gas temperature 1370 K.

5.12 Calculate the following parameters for a constant-pressure fuel-air cycle (Fig. 5.2*b*):

(a) The pressures and temperatures at states 1, 2, 3, 4, 5, and 6

(b) The indicated fuel conversion efficiency

(c) The imep

(d) The residual fraction

(e) The volumetric efficiency

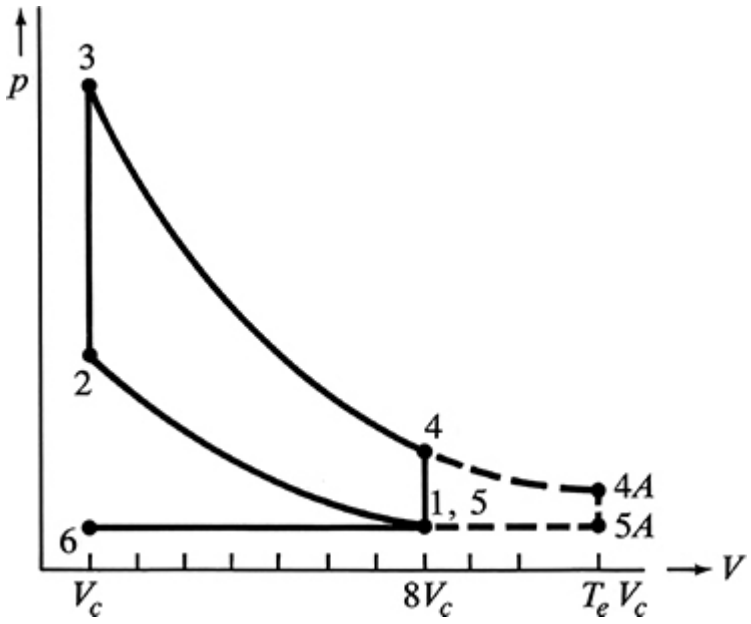
Intake pressure = 1 bar, exhaust pressure = 1 bar, intake temperature = 300 K, compression ratio = 18:1, equivalence ratio = $\phi = 0.6$.

Calculate the above parameters (points a-e) using the SI unit's charts. Use 44.4 MJ/kg for heating value of the fuel. *Hint* : Start the calculations using the residual mass fraction 0.03 and the residual gas temperature 900 K.

5.13 The cycle 1-2-3-4-5-6-1 is a conventional constant-volume fuel-air cycle with a compression ratio of 8. The fuel is isooctane, C_8H_{18}, with a lower heating value of 44.4 MJ/kg. The gas state at 1 is $T_1 = 300$ K, $p_1 = 1$ atm with an equivalence ratio of 1.0 and zero residual fraction. The specific volume at state 1 is 0.9 m^3/kg air in the mixture. The temperature at the end of compression at state 2 is 600 K.

(a) Find the indicated fuel conversion efficiency and mean effective pressure of this fuel-air cycle model of a spark-ignition engine.

(b) The efficiency of the cycle can be increased by increasing the expansion ratio r_e while maintaining the same compression ratio r_c (cycle 1-2-3-4*A*-5*A*-6-1). (This can be done with valve timing.) If the expansion ratio r_e is 12, while the compression ratio and other details of the cycle remain the same as in (*a*), what is the indicated efficiency and mean effective pressure (based on the new, larger, displaced volume) of this new engine cycle?



5.14 We would like to examine the effects of using reformulated gasoline in an SI engine. Reformulated gasoline (for lower emissions) must contain 2% by weight of oxygen. One approach is to blend ethanol (C_2H_6O) with the gasoline (approximate as C_7H_{13}) in the appropriate ratio to provide this 2% oxygen. The heating value of gasoline is 44 MJ/kg; the heating value of ethanol is 26.9 MJ/kg. The density of the two fuels is essentially the same, 0.76 Mg/M^3.

(a) Calculate the required mass fraction of ethanol in the ethanol-gasoline mixture.

(b) The stoichiometric air/fuel for ethanol and gasoline are 9.0 and 14.6, respectively. Determine the stoichiometric A/F ratio for the mixture.

(c) The fuel injection system injects a given volume of fuel based on the measured mass flow rate of air. So in a spark-ignition engine designed for stoichiometric operation with gasoline, the engine will run lean with the ethanol-gasoline mixture. At what lean relative air/fuel ratio will it operate?

(d) If the fuel-air cycle results in Fig. 5.9 apply for both fuels, what is the ratio of brake specific fuel consumption of the ethanol-gasoline fuel to that of the gasoline fuel? For a given driving pattern what will be the ratio of miles per gallon for the two fuels?

5.15 In spark-ignition engines, exhaust gas is recycled to the intake at part load to reduce the peak burned gas temperatures and lower emissions of

nitrogen oxides.

(a) Calculate the reduction in burned gas temperature that occurs when, due to exhaust gas recycle, the burned gas fraction in the unburned gas mixture (x_b) inside the cylinder is increased from 10% (the normal residual fraction) to 30%. Assume combustion occurs at top-center, at constant volume, and is adiabatic. Conditions at the end of compression for both cases are: $T = 700$ K, $p = 1000$ kPa, $v = 0.2$ m^3/kg air in the original mixture; the equivalence ratio is 1.0. The fuel can be modeled as isooctane.

(b) The compression ratio is 8. The compression stroke work is 300 kJ/kg air in the original mixture. Find the indicated work per cycle for the compression and expansion strokes, per kilogram of air in the original mixture, for these two cases.

(c) Briefly explain how you would increase the work per cycle with 30% burned gas fraction in the unburned mixture to the value obtained with 10% burned gas fraction, with fixed engine geometry. (A qualitative answer, only, is required here.)

5.16 The following cycle has been proposed for improving the operation of a four-stroke cycle engine. Its aim is to expand the post-combustion cylinder gases to a lower pressure and temperature by extending the expansion stroke, and hence extract more work per cycle.

The cycle consists of: (1) an intake stroke; (2) a compression stroke, where the inlet valve remains open (and the cylinder pressure is constant) for the first portion of the stroke; (3) a combustion process, which occurs rapidly close to top-center; (4) an expansion stroke, where the exhaust valve remains closed until the end of the stroke; (5) an exhaust stroke, where the cylinder pressure blows down to the exhaust pressure rapidly and most of the remaining combustion products are expelled as the piston moves from the BC to the TC position. Thus, for this engine concept, the compression ratio r_c (ratio of cylinder volume at inlet valve closing to clearance volume) is less than the expansion ratio r_e (ratio of cylinder volume at exhaust valve opening to clearance volume).

(a) Sketch a p-V diagram for the cylinder gases for this cycle operating unthrottled.

(b) Using the charts in SI units developed for fuel-air cycle calculations, carry out an analysis of an appropriate ideal model for this cycle where the compression ratio r_c is 8 and the expansion ratio r_e is (1) 8; (2) 16.

Assume the following: pressure in the cylinder at inlet valve close 1 bar; mixture temperature at inlet valve close 300 K; mixture equivalence ratio = 1.0; fuel: isooctane C_8H_{18}; fuel lower heating value = 44.4 MJ/kg; residual gas mass fraction at inlet valve close 0.05; stoichiometric fuel/air ratio = 0.066.

Calculate the indicated work per cycle per kg of air in the original mixture (the standard chart units) and the indicated mean effective pressure for these two expansion ratios. Base the mean effective pressure on the volume displaced by the piston during the *expansion* stroke. Tabulate your answers. (*Note:* You are given the initial conditions for the cycle calculation; changing the value of r_e requires only modest changes in the cycle calculation.)

(c) Comment briefly on the effect of increasing the ratio r_e/r_c above 1.0 with this concept on engine efficiency and specific power (power per unit engine weight). Additional calculations are not required.

5.17 The earliest successful reciprocating internal combustion engine was an engine developed by Lenoir in the 1860s. The operating cycle of this engine consisted of two strokes (i.e., one crankshaft revolution). During the first half of the first stroke, as the piston moves away from its top-center position, fuel-air mixture is drawn into the cylinder through the inlet valve. When half the total cylinder volume is filled with fresh mixture, the inlet valve is closed. The mixture is then ignited and burns rapidly. During the second half of the first stroke, power is delivered from the high-pressure burned gases to the piston. With the piston in its bottom-center position, the exhaust valve is opened. The second stroke, the exhaust stroke, completes the cycle as the piston returns to top-center.

(a) Sketch a cylinder pressure versus cylinder volume diagram for this engine.

(b) Using the charts in SI units developed for fuel-air cycle calculations, carry out a cycle analysis and determine the indicated fuel conversion efficiency and mean effective pressure for the Lenoir engine.

Assume the following: intake pressure = 1 bar; intake mixture temperature = 300 K;

mixture equivalence ratio = 1.0; fuel: isooctane C_8H_{18}, lower heating value = 44.4 MJ/kg. The clearance volume can be neglected.

(c) Compare these values with typical values for the constant-volume fuel-air cycle. Explain (with thermodynamic arguments) why the two cycles have such different indicated mean effective pressures and efficiencies.

(d) Explain briefly why the real Lenoir engine would have a lower efficiency than the value you calculated in (*b*) (the actual *brake* fuel conversion efficiency of the engine was about 5%).

5.18 Estimate from fuel-air cycle results the indicated fuel conversion efficiency, the indicated mean effective pressure, and the maximum indicated power (in kilowatts) at wide-open throttle of these two four-stroke cycle spark-ignition engines:

1. A six-cylinder engine with a 9.2-cm bore, 9-cm stroke, compression ratio of 7, operated at an equivalence ratio of 0.8
2. A six-cylinder engine with an 8.3-cm bore, 8-cm stroke, compression ratio of 10, operated at an equivalence ratio of 1.1

Assume that actual indicated engine efficiency is 0.85 times the appropriate fuel-air cycle efficiency. The inlet manifold pressure is close to 1 atmosphere. The maximum permitted value of the mean piston speed is 15 m/s. Briefly summarize the reasons why:

(a) The efficiency of these two engines is approximately the same despite their different compression ratios.

(b) The maximum power of the smaller displacement engine is approximately the same as that of the larger displacement engine.

5.19 In this problem we would like to compare the CO_2-producing potential of natural gas, methanol, and gasoline in vehicles with spark-ignition engines. Assume all three engine/fuel combinations are operating at stoichiometric and that the knock-limited compression ratios for the fuels are:

Fuel	Knock-limited compression ratio
Natural gas	12.0
Methanol	10.5
Gasoline	9.0

(a) Calculate the mass of CO_2 produced per unit of each fuel's chemical energy released (i.e., heating value) in gCO_2/MJ.

(b) Estimate the best *brake* fuel conversion efficiency of the three engine/fuel combinations at the given operating conditions, and hence their relative CO_2 emissions per unit engine power output. (Normalize by the gasoline value to simplify the comparison.) You will need to make appropriate assumptions in developing these estimates. Assume the air-fuel cycle results in Figs. 5.9 and 5.10 in the text are valid for these three different fuels. At its best efficiency, the gasoline engine has a mechanical efficiency of 0.85.

(c) Will these engine/fuel combinations produce the same output? Estimate the maximum bmep for these three engine/fuel combinations, allowing (approximately) for differences, if any, in volumetric efficiency, mechanical efficiency, fuel conversion efficiency, etc. (Look at Fig. 6.3 in the text first.) Assume that the mechanical efficiency at maximum torque can be approximated by the expression $\eta_m = 1 - r_c/50$.

5.20 The constant-volume combustion fuel-air cycle model can be used to estimate the effect of changes in internal combustion engine design and operating variables on engine efficiency. The following table gives the major differences between a diesel and a spark-ignition engine both operating at half maximum power.

	Diesel engine	Spark-ignition engine
Compression ratio	16:1	9:1
Fuel/air equivalence ratio	0.4	1.0
Inlet manifold pressure	1 bar	0.55 bar

(a) Use the graphs of fuel-air cycle results (Figs. 5.9 and 5.10) to estimate the ratio of the diesel engine *brake* fuel conversion efficiency to the spark-ignition engine *brake* fuel conversion efficiency.
(b) Estimate what percentage of the higher diesel brake fuel conversion efficiency comes from:

1. The higher diesel compression ratio
2. The leaner diesel equivalence ratio
3. The lack of intake throttling in the diesel compared with the spark-ignition engine

The values of fuel conversion efficiency and mean effective pressure given in the graphs are *gross* indicated values (i.e., values obtained from $\int pdV$ over the compression and expansion strokes only).

You may assume, if necessary, that for the real engines, the gross indicated efficiency and gross indicated mean effective pressure are 0.85 times the fuel-air cycle values. Also, the mechanical rubbing friction for each engine is 30% of the *net* indicated power or mep.

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[a] These cycles are also referred to by the titles Otto cycle, Diesel cycle, and dual cycle, respectively, for historical reasons. The more descriptive titles used above are preferred because they avoid the often-made assumption that the spark-ignition or Otto engine is best approximated by the constant-volume cycle, and the compression-ignition or diesel engine is best approximated by the constant-pressure cycle. These assumptions are not necessarily correct.

[b] Note that the underside of the piston, as it moves up and down the cylinder, does work on the gas in the crankcase. If the crankcase gas pressure is constant (a good assumption, especially in multicylinder

four-stroke engines), then the intake and exhaust stroke crankcase gas work transfer are opposite in sign and cancel out.

[c] Note that if insufficient air is available for complete combustion of the fuel, Eq. (5.28) must be modified. The right-hand side of the equation should then be $\eta_c m_f Q_{\text{LHV}}$, where η_c is the combustion efficiency given by Eq. (3.27) .

[d] The issue of chemical equilibrium with the atmosphere must also be considered. Attainment of chemical equilibrium with the environment requires the capacity to extract work from the partial pressure differences between the various species in the working fluid and the partial pressures of those same species in the environment. This would require such devices as ideal semipermeable membranes and efficient low input pressure, high-pressure ratio, expansion devices (which are not generally available for mobile power plant systems). Inclusion of these additional steps to achieve full equilibrium beyond equality of temperature and pressure is inappropriate. [8]

[e] Δg_{298} is the Gibbs free energy change for the combustion reaction, per unit mass of fuel.

[f] Δh_{298} is the enthalpy change for the combustion reaction, again per unit mass of fuel.

[g] This is consistent with the ideal gas standard cycle result [Eq. (5.70)]. As ϕ decreases, so does $Q^*/(c_v T_1)$. The factor which multiplies the natural logarithm (which increases) has a greater impact than the logarithmic term (which decreases).

[h] Note that in the fuel-air cycle with idealized valve timing, the compression process starts immediately after BC. In most engines, charge compression starts later, close to the time that the intake valve closes some 30 to 60° after BC. This matching process is approximate.

[i] As actual engines continue to improve, this ratio increases. Historically, a value of about 0.8 has been used to relate the indicated performance of actual engines to equivalent fuel-air cycle predictions. [1] The indicated efficiency of modern engines has steadily increased over the past few decades: the indicated efficiency of a spark-ignition or diesel engine is now about 0.85 times the equivalent fuel-air cycle efficiency for the same values of r_c, ϕ, and initial conditions.

CHAPTER 6

Gas Exchange Processes

This chapter deals with the fundamentals of the gas exchange process—intake and exhaust in four-stroke cycle engines and scavenging in two-stroke cycle engines. The purpose of the exhaust and intake processes, or of the scavenging process, is to remove the burned gases at the end of the power stroke and admit the fresh charge for the next cycle. Equation (2.45) shows that the brake power of an internal combustion engine at a given speed is proportional to the mass flow rate of air into the cylinders. Inducting the maximum amount of air into the engine at wide-open throttle (full load) and retaining that mass within each cylinder maximizes engine power and is thus a primary goal of the gas exchange process. Engine gas exchange processes are characterized by overall parameters such as volumetric efficiency (for four-stroke cycles), and scavenging efficiency and trapping efficiency (for two-stroke cycles). These overall parameters depend on the design of engine subsystems such as manifolds, valves, and ports, and the rest of the exhaust and intake systems, as well as engine operating conditions. Thus, the flow through individual components in the engine intake and exhaust system has been extensively studied. Supercharging and turbocharging are used to increase air flow through engines, and hence power density, b y increasing the density of the air entering each cylinder. Obviously, whether the engine is naturally aspirated (draws air directly from the atmosphere), or is supercharged or turbocharged, significantly affects the gas exchange processes. The above topics are the subject of this chapter.

The amount of air inducted into each cylinder limits the amount of fuel that can be burned. Thus the ability of the total engine system to flow as much air as possible into each cylinder—its breathing capacity at a given speed—is a

very important design objective. The degree to which the burned gases from the previous cycle are expelled from the cylinder also affects the airflow into the cylinder. The exhaust process is therefore important, too. In a spark-ignition engine, the torque produced at any given speed is regulated by controlling the air flow rate into the engine. A valve—usually a throttle plate—partly closes off the intake to limit air flow. This causes the pressure, and hence air density, in the intake manifold to decrease thereby reducing the torque and power. The maximum airflow at any given speed occurs at wide-open throttle or full load.

For spark-ignition engines, the fresh charge comprises fuel, air, and (if used for emission control) recycled exhaust, so mixture preparation is also an important goal of the intake process. Mixture preparation includes both achieving the appropriate mixture composition and achieving equal distribution of air, fuel, and recycled exhaust amongst the different cylinders. In diesels, only air (or air plus recycled exhaust) is inducted. Mixture preparation and manifold flow phenomena are discussed in Chap. 7 . A third goal of the gas exchange processes is to set up the flow field within the engine cylinders that will give a fast-enough combustion process for satisfactory engine operation. In-cylinder flows are the subject of Chap. 8 .

6.1 INTAKE AND EXHAUST PROCESSES IN THE FOUR-STROKE CYCLE

In a four-stroke spark-ignition engine, the intake system typically consists of an air filter, an air duct, a throttle, an intake manifold, and individual cylinder fuel injectors in each intake port or cylinder head. During the induction process, pressure losses occur as the mixture passes through or by each of these components. There is a significant pressure drop across the intake port and valve. The exhaust system typically consists of an exhaust manifold, exhaust pipe, one or more catalytic converters for emission control, and one or more mufflers (or silencers). Figure 6.1 illustrates the intake and exhaust gas flow processes in a conventional spark-ignition engine where the valve opening and closing crank angles are fixed at well-established values. Many modern engines now include variable camshaft phasing, so the valve opening and closing timings can be varied. Other means of varying the valve timing

and lift are also used. The flows to each cylinder are pulsating. However, many aspects of these flows can be analyzed on a quasi-steady basis, and the pressures indicated in the intake system in Fig. 6.1*a* represent time-averaged values for a multicylinder engine.

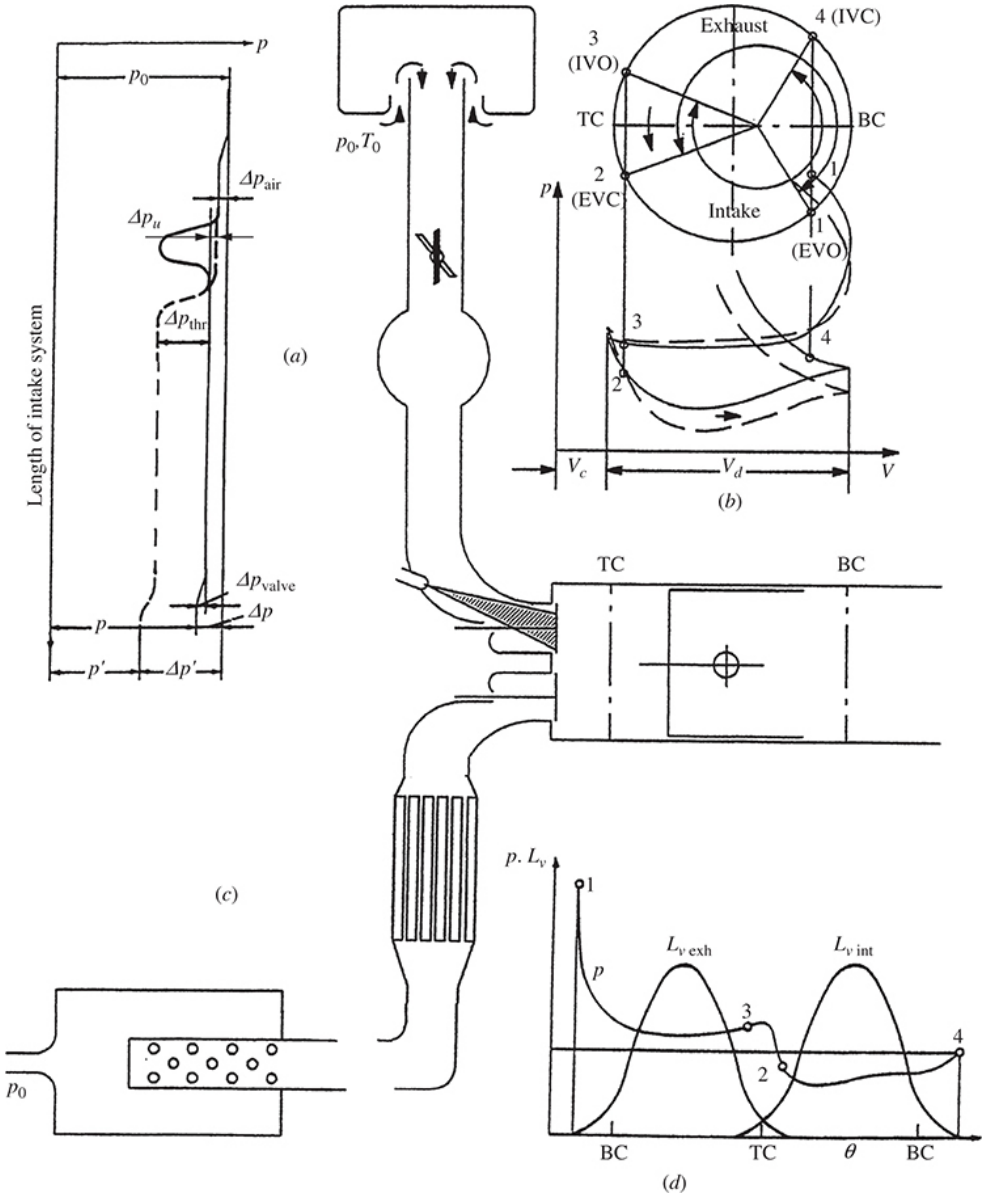


Figure 6.1 Intake and exhaust processes for four-stroke cycle spark-ignition engine: (*a*) intake system; (*b*) valve timing and pressure-volume diagrams; (*c*) exhaust system; (*d*) cylinder pressure p and valve lift Lv versus crank angle θ. Solid lines are for wide-open throttle, dashed lines are for part throttle; p_0, T_0 are atmospheric conditions; Δp_{air} = pressure losses in

air cleaner; Δp_u = intake losses upstream of throttle; Δp_{thr} = losses across throttle; Δp_{valve} = losses across the intake valve. [1]

The drop in pressure along the intake system depends on engine speed, the flow resistance of the elements in the system, the cross-sectional area through which the fresh charge moves, and the charge density. Figure 6.1 *d* shows the intake and exhaust valve lifts versus crank angle. The usual practice is to extend the valve open phases beyond the intake and exhaust strokes to improve emptying and charging of the cylinders and make the best use of the inertia of the gases in the intake and exhaust systems. The exhaust process usually begins 40 to 60° before BC (point 1, EVO, Fig. 6.1*b*). Until about BC the burned cylinder gases are discharged due to the pressure difference between the cylinder and the exhaust system. After BC, the cylinder is scavenged by the piston as it moves toward TC. The terms *blowdown* and *displacement* are used to denote these two phases of the exhaust process. Typically, the exhaust valve closes about 15° after TC (point 2, Fig. 6.1*b*), and the intake valve opens 10 to 15° before TC (point 3, Fig. 6.1*b*). Both valves are open during an *overlap period*, and when $p_i / p_e < 1$, backflow of exhausted gas into the cylinder and of cylinder gases into the intake will usually occur. The advantage of valve overlap occurs at high engine speeds when the longer valve open periods improve volumetric efficiency. As the piston moves past TC and the cylinder pressure falls below the intake pressure, gas flows from the intake into the cylinder. The intake valve remains open until 50 to 60° after BC (point 4, Fig. 6.1*b*) so that fresh charge may continue to flow into the cylinder after BC at higher engine speeds.

In a diesel engine intake system, the port fuel injection system and the throttle plate are absent. Diesel engines are more frequently turbocharged. A set of diagrams illustrating the intake and exhaust processes for a turbocharged four-stroke diesel is shown in Fig. 6.2. When the exhaust valve opens (point 1), the burned cylinder gases are fed to a turbine (T), which drives a compressor (C) which compresses the air prior to entry to the cylinder. Charge air cooling is usually employed; a heat exchanger, called an intercooler (ICHE) is placed between the compressor exit and the intake plenum or manifolds to reduce the compressed air temperature and increase its density.

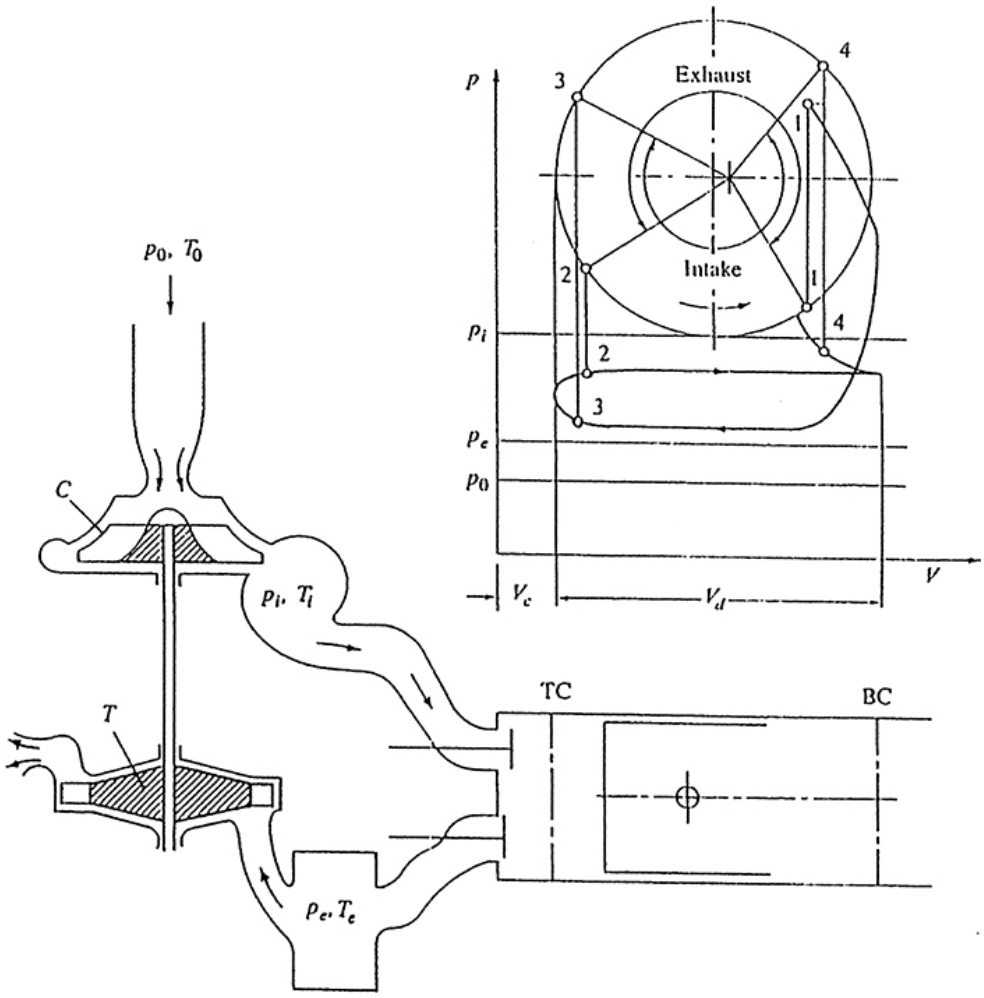


Figure 6.2 Intake and exhaust process for turbocharged four-stroke cycle engine. The turbocharger compressor C raises air pressure and temperature from ambient p_0, T_0 to p_i, T_i. Cylinder pressure during intake is less than p_i. During exhaust, the cylinder gases flow through the exhaust manifold to the turbocharger turbine T. Manifold pressure p_e may vary during the exhaust process and lies between cylinder pressure and ambient. [1]

Due to the time-varying valve open area and cylinder volume, gas inertia effects, and wave propagation in the intake and exhaust systems, the pressures in the intake, the cylinder, and the exhaust during these gas exchange processes vary in complicated ways. Analysis of these processes is complex (see Secs. 7.6.2 and 14.3 for a review of available methods). In practice, these processes are often treated empirically using overall

parameters such as volumetric efficiency to define intake and exhaust system performance.

Figure 6.3 shows an example of an intake manifold system for a V-6 gasoline SI engine. Air ducts lead through two throttles, one on each side of the engine, mechanically connected together, to two plenums from each of which three runners lead to the intake ports. Each plenum has larger cross-sectional flow area than the manifolds so that the different length flow paths through the plenum to the intake ports have little effect on the individual manifold runner airflows. With this intake manifold, a valve connects the two runners closest to the throttle. The valve is closed at low and high speeds so the two plenums are isolated from each other; this valve is opened at mid-speeds so the two intake manifold branches are connected. Changing the intake manifold geometry as engine speed varies can take advantage of different resonant behaviors.

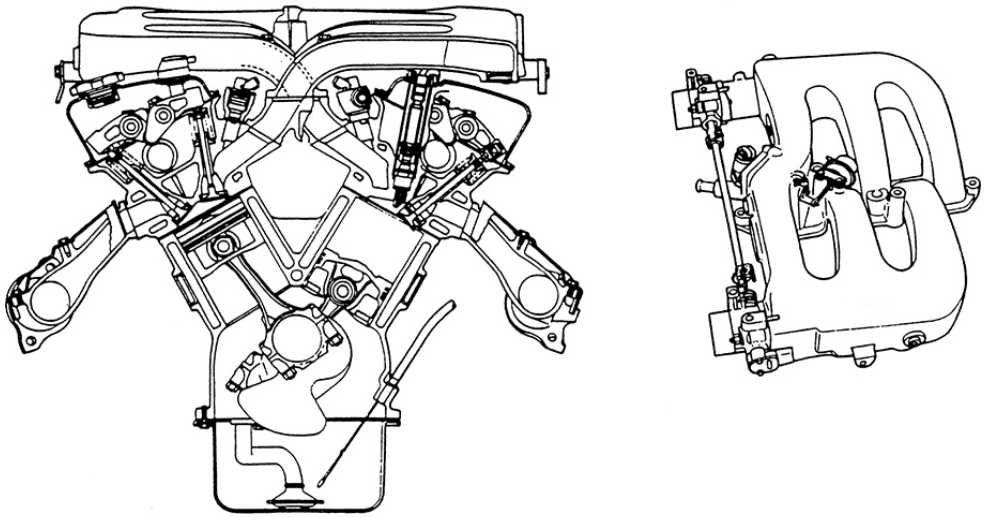


Figure 6.3 Example of a port-fuel-injected gasoline SI engine intake system and cylinder head design: 3.5-liter V-6 engine.[2]

Figure 6.4 shows the components and their layout in the exhaust system of a multicylinder automobile gasoline engine. It consists of an exhaust manifold, a catalytic converter close coupled to the engine for rapid light off which removes the pollutants NO_x, CO, and HC, one or more mufflers to absorb the noise produced by the pulsating exhaust flow, and connecting pipework. For V-6 and V-8 engines, two parallel exhaust systems are often

employed. In a spark-ignition engine the exhaust gases are extremely hot so the exhaust system components are made of high-temperature alloy steels.

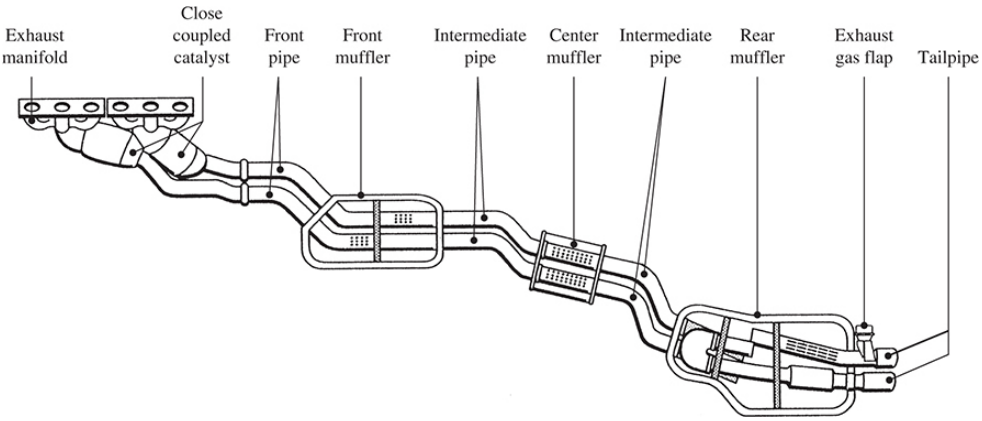


Figure 6.4 Gasoline 6-cylinder SI engine exhaust system, with close-coupled catalytic converters and three mufflers.[3] (*Courtesy Robert Bosch GmbH and SAE.*)

In diesel engines, the exhaust gas is cooler than in a gasoline engine since the diesel always operates well lean of stoichiometric and the engine extracts more of the high temperature burned gas energy as work. The gasoline engine catalyst is replaced by a particulate trap and a diesel NO_x catalyst (see Chap. 11). Diesel engines are normally turbocharged, and the turbine is close coupled to the exhaust manifold to maximize its effectiveness. In turbocharged gasoline SI engines the turbocharger is also close coupled, and due to the higher exhaust gas temperature, a turbine bypass valve or wastegate is incorporated to bypass some of the exhaust gas flow to avoid over-boosting the engine.

6.2 VOLUMETRIC EFFICIENCY

Volumetric efficiency is used as an overall measure of the effectiveness of a naturally-aspirated four-stroke cycle engine, and its intake and exhaust systems, as an air pumping device. It is defined [see Sec. 2.10 , Eq. (2.33)] as

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,0} V_d N}$$

The air density $\rho_{a,0}$ is usually evaluated at atmospheric conditions; η_v is then the overall volumetric efficiency. It can be evaluated at inlet manifold conditions, $\rho_{a,i}$; η_v then measures the pumping performance of the cylinder, intake port, and valve alone. This discussion will cover unthrottled (or wide-open-throttle) engine operation only. It is the air flow under these conditions that constrains maximum engine power. Lesser airflows in SI engines are obtained by restricting the intake system flow area with the throttle valve.

We now examine the several phenomena or factors that have a significant effect on volumetric efficiency: that is, an engine's airflow capability. These factors are:

1. *Quasi-static effects*. Fuel composition, fuel/air ratio, fuel vaporization, intake air temperature, ratio of exhaust to intake manifold pressures, compression ratio, all affect the amount of air inducted.
2. *Intake and exhaust system flow friction*. Due to flow friction, pressure drops occur in the intake and in the exhaust systems.
3. *Port and valve flow friction*. The intake valve and port are the largest flow resistance in the intake system. There is a significant pressure drop across the port and valve due to flow friction.
4. *Intake system heat transfer*. Heat transfer to the mixture in the intake manifold and port decreases the air density.
5. *Valve timing effects*. At low engine speed, a significant amount of mixture which entered the cylinder during the intake stroke is pushed back into the intake by the moving piston during the compression stroke prior to inlet valve closing. At high engine speed, the high momentum of the air in the intake manifold toward the cylinder set-up during the intake stroke forces additional air into the cylinder while the intake valve remains open after BC.
6. *Airflow choking at intake valve and port*. At very high piston speeds, the velocity at the point where the flow area in the intake port

and valve is a minimum can become sonic. This phenomenon, called choking, limits the airflow rate into the cylinder.

7. *In-cylinder heat transfer*. The incoming mixture is cooler than the combustion chamber walls. Heat transfer to the fresh charge within the cylinder decreases its density.
8. *Intake (and exhaust) tuning*. By suitable design of intake and exhaust systems, advantage can be taken of wave phenomena resulting from the pulsating flows to increase the pressure in the intake port at the end of the intake process, and lower the pressure in the exhaust port, at specific engine speeds.
9. *Turbocharging or supercharging*. By compressing the intake air before it enters the cylinder in a compressor or blower, the air density is increased and the mass of air entering each cylinder is thus increased.

In addition to the variables listed in (1) above, these factors are influenced by the design, layout, and detailed geometry of the intake and exhaust system components, as well as details of the intake and exhaust ports and valves in the cylinder head and how these valves are controlled by the cam mechanism during the engine cycle. Except for the quasi-static effects (item 1 above), the magnitude of these various factors or phenomena vary significantly with intake mixture velocity in the intake system, manifold, port, and intake valve. Local velocities for quasi-steady flow are equal to the volume flow rate divided by the local cross-sectional area. Since the intake system, port and valve dimensions typically scale with the cylinder bore, air velocities in the intake will scale with piston speed, since it is piston motion that pulls the air into each cylinder in turn. Hence, volumetric efficiency will depend strongly on piston speed. Therefore, volumetric efficiencies for different engines should be compared at constant mean piston speed rather than engine speed.

We will now discuss the phenomena or factors that have a significant impact on airflow and volumetric efficiency, and especially their dependence on speed.

6.2.1 Quasi-Static Effects

Volumetric Efficiency of an Ideal Cycle

For the ideal cycles of Fig. 5.2*d* and *e*, an expression for volumetric efficiency can be derived which is a function of the following variables: intake mixture pressure p_i, temperature T_i, and fuel/air ratio (F / A); compression ratio r_c; exhaust pressure p_e; and molecular weight M and γ for the cycle working fluid. The overall volumetric efficiency is

$$\eta_v = \frac{m_a}{\rho_{a,0} V_d} = \frac{m(1-x_r)}{\rho_{a,0}\left[1+(F/A)\right]} \frac{r_c}{(r_c-1)V_1} \tag{6.1}$$

where m is the mass in the cylinder at point 1 in the cycle. Since

$$p_i V_1 = m \frac{\tilde{R}}{M} T_1 \quad \text{and} \quad p_{a,0} = \rho_{a,0} \frac{\tilde{R}}{M_a} T_{a,0}$$

and Eq. (5.38) relates T_1 to T_i, the above expression for η_v can be written

$$\eta_v = \left(\frac{M}{M_a}\right)\left(\frac{p_i}{p_{a,0}}\right)\left(\frac{T_{a,0}}{T_i}\right)\frac{1}{\left[1+(F/A)\right]}\left\{\frac{r_c}{r_c-1} - \frac{1}{\gamma(r_c-1)}\left[\left(\frac{p_e}{p_i}\right)+(\gamma-1)\right]\right\} \tag{6.2}$$

For $p_e / p_i = 1$, the term in { } is unity. We see from Eq. (6.2) that there are static effects from variations in p_i, p_e, T_i, fuel characteristics F/A and r_c. There are also dynamic effects from variations in engine speed.

Effect of Fuel Composition, Phase, and Fuel/Air Ratio

In a spark-ignition engine, the presence of gaseous fuel (and water vapor) in the intake system reduces the air partial pressure below the mixture pressure. For mixtures of air, water vapor, and gaseous or evaporated fuel we can write the intake manifold pressure as the sum of each component's partial pressure:

$$p_i = p_{a,i} + p_{f,i} + p_{w,i}$$

which, with the ideal gas law, gives

$$\frac{p_{a,i}}{p_i} = \left[1 + \left(\frac{\dot{m}_j}{\dot{m}_a}\right)\left(\frac{M_a}{M_f}\right) + \left(\frac{\dot{m}_w}{\dot{m}_a}\right)\left(\frac{M_a}{M_w}\right)\right]^{-1} \tag{6.3}$$

The water vapor correction is usually small (≤0.03). This ratio, $p_{a,i}/\ p_i$, for several common fuels as a function of $(\dot{m}_f/\dot{m}_a)$ is shown in Fig. 6.5. Note that $(\dot{m}_f/\dot{m}_a)$ only equals the engine's operating fuel/air ratio if the fuel is fully vaporized.

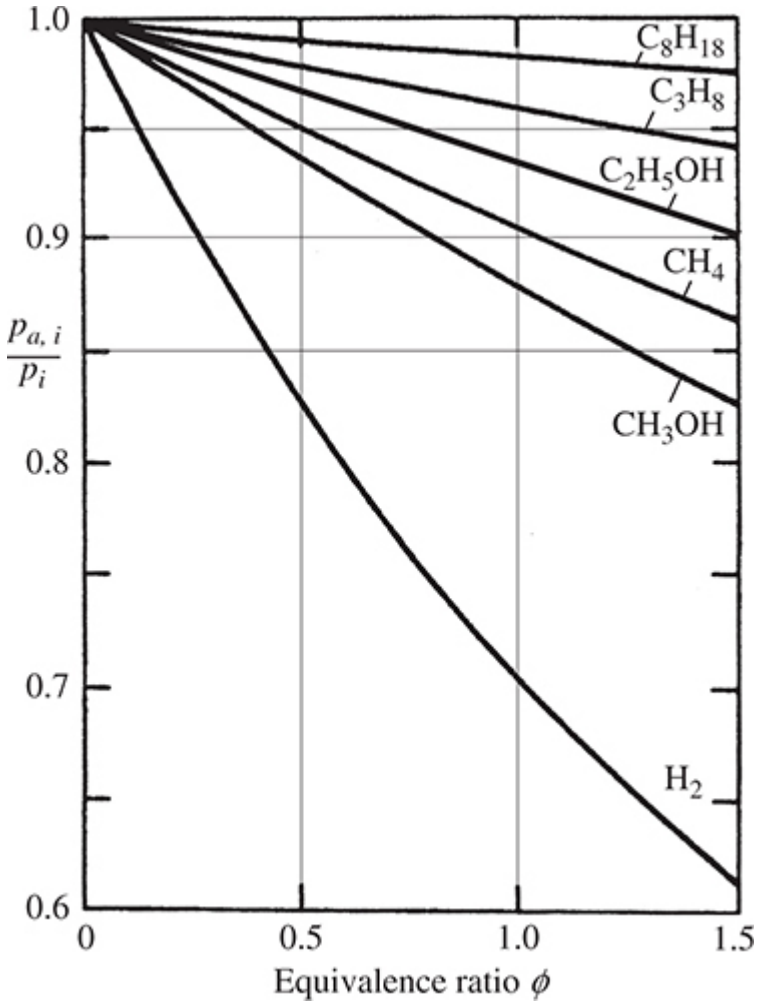


Figure 6.5 Effect of fuel (vapor) on inlet air partial pressure. Ratio of air inlet pressure $p_{a,i}$ to mixture inlet pressure p_i versus fuel/air equivalence ratio ϕ for isooctane vapor, propane, ethanol vapor, methane, methanol vapor, and hydrogen.

For conventional liquid fuels such as gasoline, which have higher molecular weights, the effect of fuel vapor, and therefore fuel/air ratio, is relatively small. For gaseous fuels and for ethanol and methanol vapor, the volumetric efficiency is significantly reduced by the fuel vapor in the intake mixture.

Fraction Fuel Vaporized and Heat of Vaporization

For a constant-pressure flow with liquid fuel evaporation and with heat transfer, the steady-flow energy equation is

$$\left[\dot{m}_a h_a + (x_{f,e})\dot{m}_f h_{f,L} + x_{f,e}\dot{m}_f h_{f,V}\right]_A = \dot{Q} + (\dot{m}_a h_a + \dot{m}_a h_a + \dot{m}_f h_{f,L})_B \qquad (6.4)$$

where $x_{f,e}$ is the mass fraction of fuel evaporated and the subscripts denote: a, air properties; f, fuel properties; L, liquid; V, vapor; B, before evaporation; A, after evaporation. Approximating the change in enthalpy per unit mass of each component of the mixture by $c_p \Delta T$, and with $h_{f,V} - h_{f,L} = h_{f,LV}$ (the enthalpy of vaporization), Eq. (6.4) becomes

$$T_A - T_B = \frac{(\dot{Q}/\dot{m}_a) - x_{f,e}(F/A)h_{f,LV}}{c_{p,a} + (F/A)c_{f,L}} \qquad (6.5)$$

Since $c_{f,L} \approx 2c_{p,a}$ the last term in the denominator can often be neglected.

With liquid-fueled port-injected SI engines, much of the injected fuel impinges on the intake valve and port walls. With gasoline, the thermal energy required to vaporize this fuel mostly comes from these walls rather than from the airflow. Thus, only a fraction of the full potential evaporative cooling of the air is realized. As the fuel drop size in the injected spray decreases, the fraction of the fuel evaporated via heat transfer with the airflow increases. With direct injection of a volatile liquid fuel into the cylinder, most of the fuel drops are suspended in the air (and residual gas) charge and undergo an adiabatic evaporation process. If no heat transfer to this mixture occurs, the mixture temperature decreases in proportion to the amount of liquid fuel vaporized. For complete evaporation of isooctane, with $\phi = 1.0$, $T_A - T_B = -20$ deg. C. For ethanol and methanol under the same conditions, $T_A - T_B$ would be −73 and −122 deg. C, respectively. In practice

some air heating occurs: also, with port injection, the fuel is not necessarily fully evaporated prior to entry to the cylinder.

The ideal cycle equation for volumetric efficiency [Eq. (6.2)] shows that the effect of gas temperature variations, measured at entry to the cylinder, is through the factor ($T_{a,0}/T_i$). This temperature scaling is appropriate at lower engine speeds. However, engine test data indicate that a square root dependence of volumetric efficiency on temperature ratio is closer to real engine behavior at high engine speeds. The square root dependence on temperature is a standard assumption made in correcting engine maximum power test data (see Sec. 2.12).

Effect of Intake and Exhaust Pressure Ratio and Compression Ratio

As the pressure ratio (p_e/p_i) and the compression ratio are varied, the fraction of the cylinder volume occupied by the residual gas *at the intake pressure* varies. As this volume increases, volumetric efficiency decreases. These effects on ideal-cycle volumetric efficiency are given by the { } term in Eq. (6.2) . For $\gamma = 1.3$ these effects are shown in Fig. 6.6. A more detailed discussion of residual gas behavior is given in Sec. 6.4 .

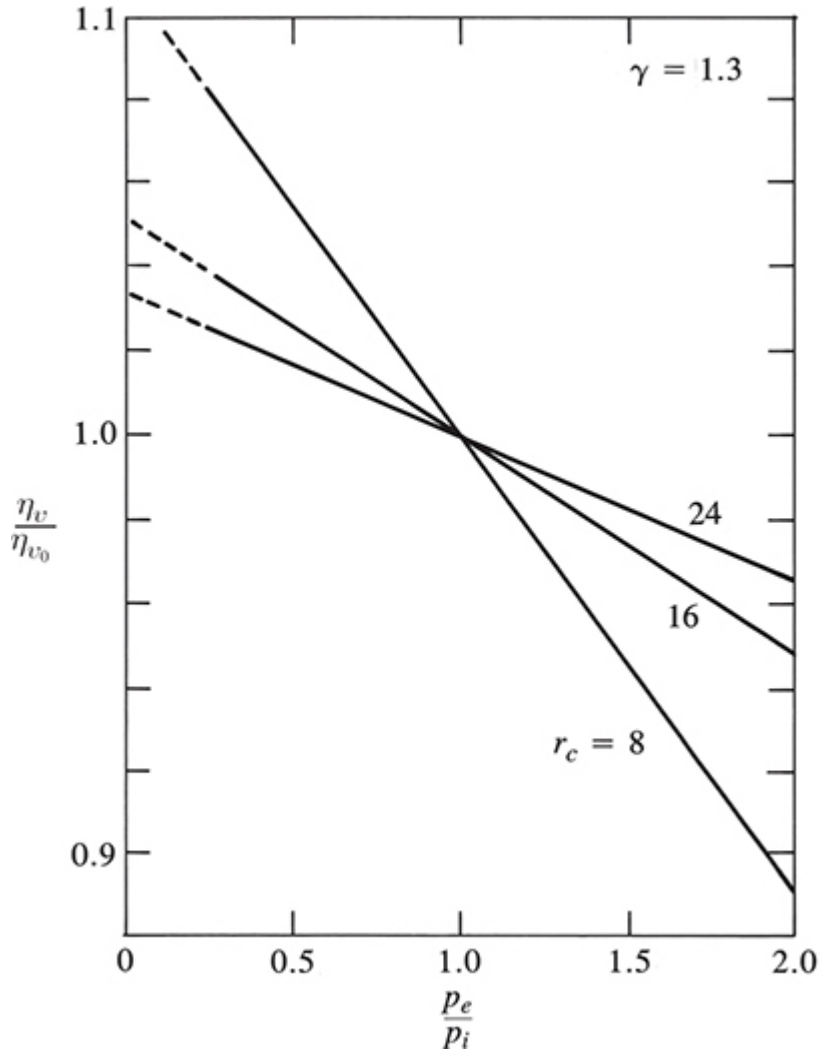


Figure 6.6 Effect of exhaust to inlet pressure ratio on ideal-cycle volumetric efficiency.

6.2.2 Intake and Exhaust Flow Resistances

When gas flows unsteadily through a system of pipes, chambers, ports, and valves, friction, pressure, and inertial forces are present. The relative importance of these forces depends on gas velocity and the size and shape of these passages and their junctions. Both quasi-steady and dynamic effects are usually significant. While the effects of changes in engine speed, and intake and exhaust manifold, port and valve design are interrelated, several separate phenomena which affect volumetric efficiency can be identified.

Intake Systems

During the intake stroke, due to flow-induced friction in each part of the intake system, the pressure in the cylinder p_c is less than the intake pressure [a] p_i by an amount dependent on the square of the flow velocity. This total pressure drop is the sum of the pressure loss in each component of the intake system: air filter, ducting, throttle, manifold, intake port, and intake valve. In a boosted engine, the compressor raises the air pressure above atmospheric; each loss is then a few percent, with the cylinder head's ports and valves contributing the largest drop. As a result, the pressure in the cylinder during the period in the intake process when the piston is moving at high speed, can be 10 to 20% lower than p_i. For each component in the intake (and in the exhaust) system, Bernoulli's equation gives

$$\Delta p_j = \xi_j \rho v_j^2$$

where ξ_j is the flow's resistance coefficient for that component which depends on its geometric details and v_j is the local velocity. Assuming the flow is quasi-steady, since the piston motion pulls the intake flow v_j is related to the piston speed S_p by

$$v_j A_j = S_p A_p$$

where A_j and A_p are the component minimum flow area and the piston area, respectively. Hence, the total quasi-steady average pressure loss due to friction is

$$p_{atm}(\text{or } p_i) - p_c = \sum \Delta p_j = \sum \xi j \rho v_j^2 = \rho \bar{S}_p^2 \sum \xi_j \left(\frac{A_p}{A_j}\right)^2 \tag{6.6}$$

where, for the average Δp, we have used the mean piston speed $\bar{S}_p$ in place of S_p. Equation (6.6) indicates the importance of large component flow areas for reducing frictional losses, and the dependence of these losses on engine speed. Figure 6.7 shows an example of the pressure losses due to friction across the air cleaner, throttle, and manifold plenum of a standard four-cylinder automobile engine intake system. These steady-flow tests, conducted

over the full engine speed range, [4] show that the pressure loss depends on speed squared.

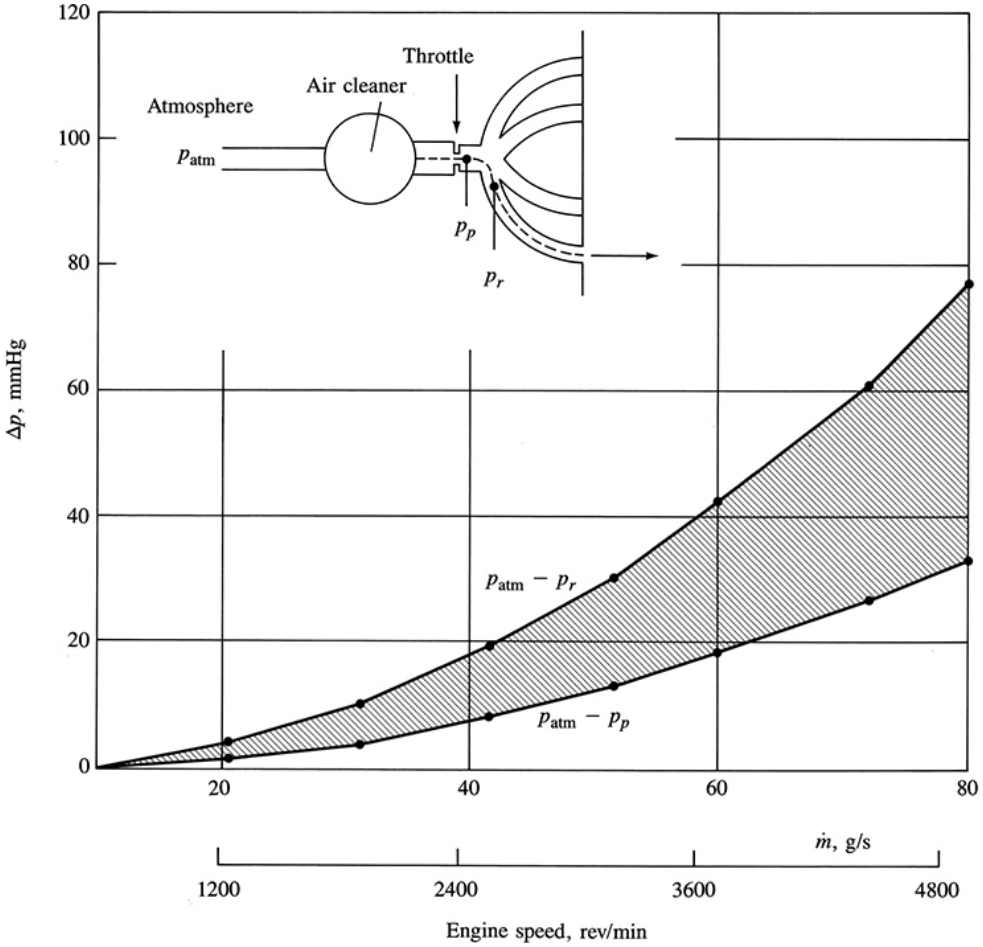


Figure 6.7 Pressure losses in the intake system of a four-stroke cycle spark-ignition engine determined under steady-flow conditions.[4] Stroke = 89 mm, bore = 84 mm.

Intake Ports and Valves

The engine intake ports and valves constitute a comparable flow resistance to that of the intake system. The relative size of the valve heads and seats, and thus their open area, are constrained by the size of the cylinder liner. The open areas they define restrict the airflow. They thus add to the set of flow resistances which are summed in Eq. (6.6) , thus causing an additional pressure loss (and density reduction) that increases with the square of engine speed. Note that as the valves lift, the minimum intake flow area in the

cylinder head transitions (at about half the maximum valve lift) from being between the valve seat and valve head at low lifts, to the port area close to the valve seat at high lifts; see Sec. 6.3.1 .

Exhaust System, Ports, and Valves

Equivalent flow-dependent pressure losses in the exhaust system result in the exhaust ports and manifold having average pressure levels that are higher than atmospheric. Figure 6.8 shows the time-averaged exhaust manifold gauge pressure (i.e., relative to atmospheric) as a function of intake manifold pressure (which varies with engine load) and with speed, for a four-cylinder naturally-aspirated automobile SI engine. At high loads (high absolute intake manifold pressures) the gauge pressure in the exhaust manifold increases faster than the square of engine speed. This is because the exhaust gas density decreases with increasing engine speed, [b] as the exhaust system version of Eq. (6.6) would indicate. In turbocharged engines, the exhaust manifold pressure at higher loads and speeds is higher because there is now a significant pressure drop across the turbine—see Fig. 6.2. That pressure drop depends on exhaust gas temperature, engine mass flow rate, and operating boost level, and whether or not the wastegate is partly open (see Sec. 6.8.5). At low engine speeds and loads, the turbocharger has little impact on engine operation so this turbine pressure drop is small.

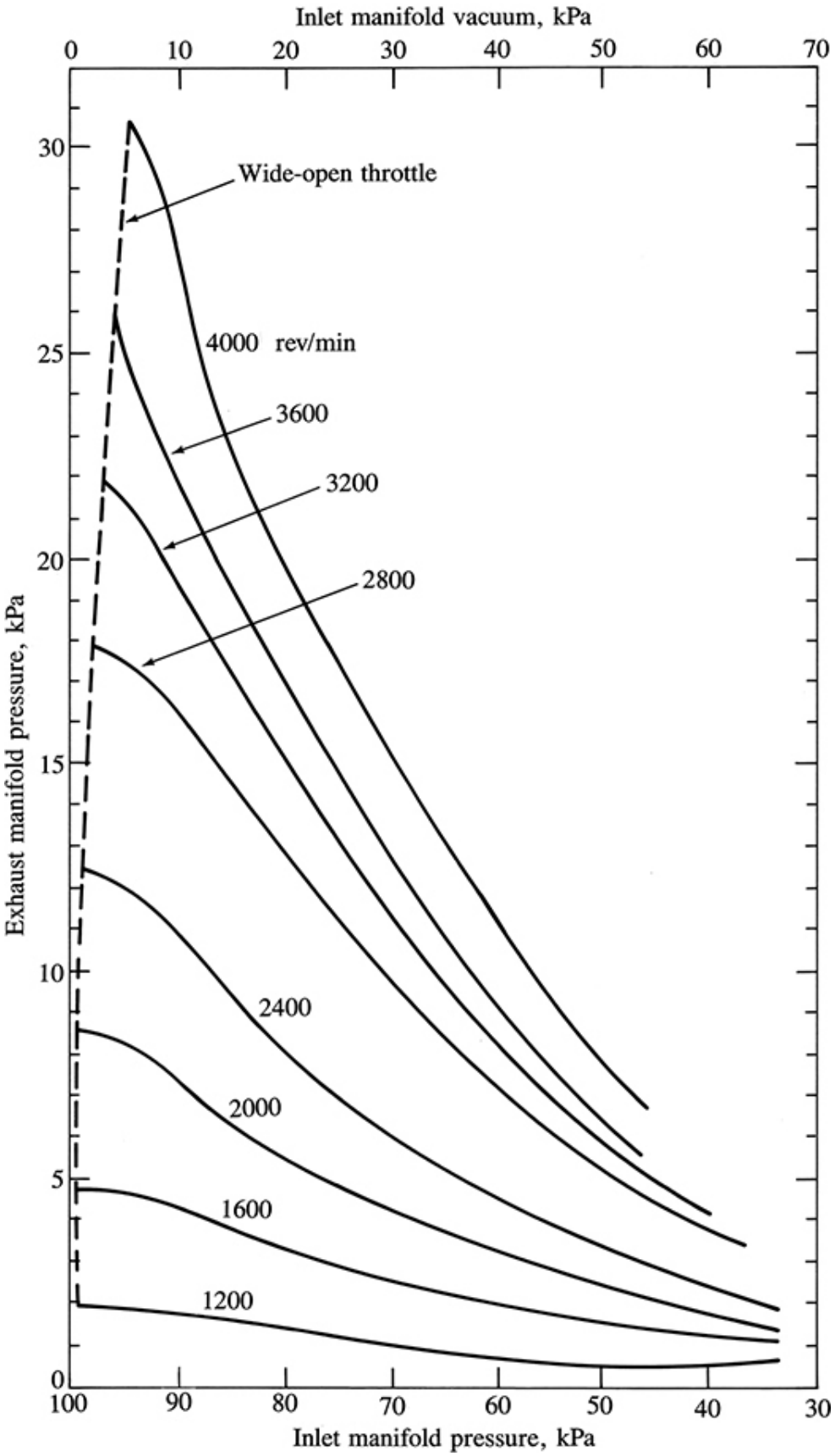

Inlet manifold vacuum, kPa
0
10
20
30
40
50
60
70
Wide-open throttle
4000 rev/min
3600
3200
2800
2400
2000
1600
1200
Exhaust manifold pressure, kPa
0
5
10
15
20
25
30
100
90
80
70
60
50
40
30
Inlet manifold pressure, kPa

Figure 6.8 Exhaust manifold pressure as a function of load (defined by inlet manifold vacuum) and speed, four-stroke cycle, four-cylinder SI engine.[5]

The flow resistances of the exhaust valves and ports are less significant than on the intake side. This is because a substantial fraction of the in-cylinder burned gas is exhausted between exhaust valve opening (EVO) and about BC in a *blowdown* process, which is driven by the pressure difference between the cylinder and the exhaust system. Then, since the exhaust gas is hot, its density (at atmospheric pressure) is lower than that of the intake air by a factor of 3 to 4. So the pressure drop given by the exhaust system version of Eq. (6.6) , due to the exhaust valves and ports during most of the exhaust stroke (BC to TC, when the piston *displaces* the remaining burned gases from the cylinder) is significantly smaller than the cylinder-head flow-resistance pressure drop during intake.

6.2.3 Intake and In-Cylinder Heat Transfer

Intake Heat Transfer

As air flows through each intake port (with its walls close to the coolant temperature, ~90°C) and past the hot intake valve surface (at ~150°C) some heating of the intake air occurs. This causes a 2 to 3% loss in volumetric efficiency. This intake heating decreases modestly with increasing speed since while the heat transfer rate for a turbulent flow in a duct increases with speed to the 0.8 power, [c] the time available per cycle for heat transfer varies as the reciprocal of speed. Thus as speed increases, the time available goes down faster than the heat transfer rate increases. The impact of intake heat transfer on volumetric efficiency depends on the intake air temperature.

In-Cylinder Heat Transfer

As air flows into the cylinder, it mixes with the hot burned residual gases left in the cylinder from the previous cycle. This process raises the temperature of the air (already mixed with fuel in port-fuel-injected SI engines) and residual-gas mixture. Also, the temperatures of the piston, valves, and cylinder head are higher than this mixture temperature, so heat transfer *to* the in-cylinder unburned mixture occurs, further raising its temperature. When

combined, these two processes result in a density decrease (and volumetric efficiency decrease) of some 1 to 2%, since the cylinder has a given displaced volume.

6.2.4 Intake Valve Timing Effects

Ram Effect at High Speed

The mass of air inducted into the cylinder each cycle, and hence the volumetric efficiency, is determined by the total pressure level in the intake port during the latter portion of the intake process. This total pressure is the sum of the static pressure and the dynamic kinetic energy pressure component. At higher engine speeds, the inertia of the gas in the intake system as the intake valve is closing increases the pressure in the port and continues the cylinder charging process as the piston slows down as it approaches BC and starts the compression stroke. This effect becomes progressively greater as engine speed increases. The intake valve in fixed-valve-timing engines is closed some 50 to 60° after BC, in part to take advantage of this so-called ram phenomenon. This effect increases the high-speed engine airflow by some 10%. With variable intake valve timing, these airflow benefits can be increased.

Reverse Flow into the Intake at Low Speed

Because the intake valve closes after the start of the compression stroke, a reverse flow of already-inducted fresh charge from the cylinder back into the intake can occur as the cylinder pressure rises due to piston motion toward TC. This reverse flow is largest at the lowest engine speeds. It is an inevitable low-speed consequence of the intake valve closing (IVC) time chosen to take advantage of the ram effect at high speeds. With fixed-valve-timing engines it results in a substantial loss of air from the cylinder (~15% at 1500 rev/min); with earlier IVC at these conditions, the loss is significantly reduced.

The effects of varying the IVC angle on relative full-load engine torque are shown in Fig. 6.9. [6] Torque is proportional to airflow, and thus volumetric efficiency, except that at the highest engine speeds the negative impact of increasing engine friction plays an increasing role. Note that the typical IVC angle of 50 to 55° ABC is an excellent compromise.

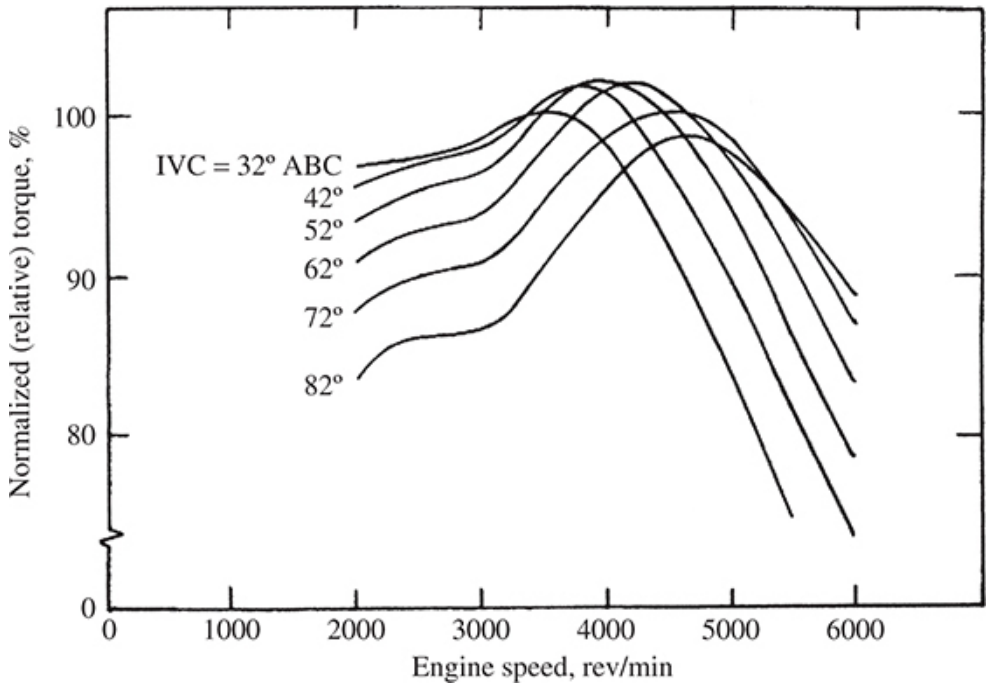


Figure 6.9 Effect of IVC angle on gasoline SI engine torque (normalized) versus engine speed. Typical fixed IVC angle is between 50 and 55° ABC. [6]

6.2.5 Airflow Choking at Intake Valve

As the intake valve opens, its minimum flow area—an annular ring between the valve head and valve seat—increases from zero up to about half the maximum valve lift when the intake port area (at the valve seat) becomes the flow restricting area (see Sec. 6.3.1). The air flow in this minimum flow area in the intake valve and port system is pulled into the cylinder by the downward piston motion. Its scaling with piston speed follows this relationship:

$$U_{a,\mathrm{IV}} A_{\min,\mathrm{IV}} \approx S_p A_p$$

At high engine speeds ($\overline{S}_p \sim 20$ m/s; see Sec. 2.2), during the first portion of the intake stroke, the valve open area is much less than the piston area so the air velocities in this minimum valve open area become large. Typical numbers (for a 4-valve per cylinder automobile SI engine) would be:

Engine details:

Bore = stroke = 85 mm; IV diameter (two intake valves) = 36 mm, maximum valve lift = 10 mm, $\overline{S}_p \approx 20$ m/s. This gives an airflow velocity past the valve seat of 300 m/s at a valve lift of 2 to 3 mm (attained at about 45° after IVO) at an engine speed of 7000 rev/min.

The speed of sound in air at standard atmospheric conditions is close to 330 m/s. Thus the airflow at valve lifts up to this few mm value, is choked—the velocity is higher than the local sound speed. When a flow is choked, decreasing the pressure downstream of the flow restriction (e.g., by further increasing the piston speed) has no effect on the mass flow rate: only increases in *upstream* pressure can increase the flow rate—see App. C, Sec. C.2. While initially the flow is choked for only a short portion of the intake process, as engine speed is increased the duration of the choked flow period increases. Following the onset of this phenomenon, as engine speed increases at wide-open-throttle and higher speeds, the airflow rate decreases increasingly rapidly because the flow frictional losses in the intake system, port, and valve rise as the square of speed. This onset of airflow choking effectively defines (along with the rapidly rising engine friction) the upper speed limit of the engine since engine brake power now starts to decline.

6.2.6 Intake and Exhaust Tuning

Figure 6.10 shows the pressure variations with time in the intake and exhaust systems of a four-cylinder automobile spark-ignition engine at wide-open throttle at low and high speed. The complexity of the gas exchange phenomena is apparent. The amplitude of the pressure fluctuations increases substantially with increasing engine speed. The primary frequency in both the intake and exhaust corresponds to the frequency of individual cylinder intake and exhaust processes. Higher harmonics in both the intake and exhaust are apparent also. This pulsating flow into and out of each cylinder during the intake and exhaust processes sets up pressure waves in the intake and exhaust systems. These pressure waves propagate at the local sound speed relative to the moving air or exhaust gas. The pressure waves interact with the pipe junctions and ends in the ducting, plenums, manifolds, and ports. These interactions cause pressure waves to be reflected back toward the engine cylinder. These pressure waves may aid or inhibit the gas exchange processes. When they aid the process by increasing the pressure in the intake

ports toward the end of the intake process, or by reducing the pressure in the exhaust ports toward the end of the exhaust process, the intake and exhaust systems are said to be *tuned*. [7]

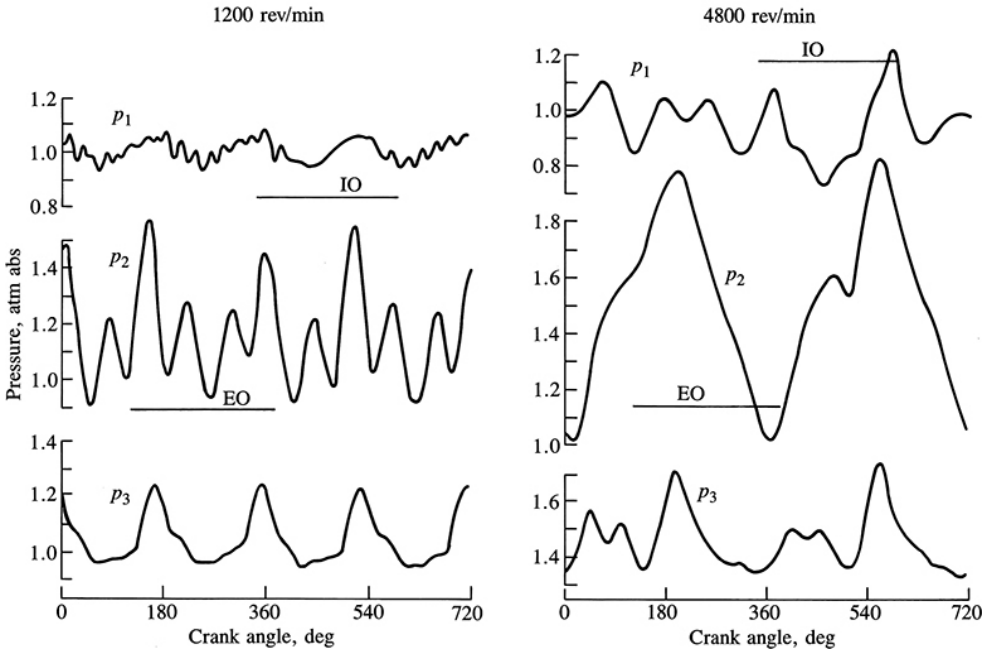


Figure 6.10 Instantaneous pressures in the intake and exhaust manifolds of a four-stroke cycle four-cylinder spark-ignition engine, at wide-open throttle. Locations: p_1, intake manifold runner 150 mm from cylinder 1; p_2, exhaust manifold runner 200 mm from cylinder 1; p_3, exhaust manifold runner 700 mm from cylinder 1. IO and EO, intake and exhaust valve open periods for that cylinder, respectively. Stroke 89 mm, bore 84 mm. [4]

Intake System Tuning

As the time-varying intake flow into the cylinder gets started, an expansion wave is propagated back through the intake manifold. As this expansion wave goes through the open end of the manifold into the plenum it causes a positive pressure wave to be propagated back toward the cylinder. If the timing of these waves is appropriately phased, this positive pressure wave will cause the pressure in the intake port to increase above its nominal value toward the end of the intake process. This will increase the inducted air mass.

Figure 6.11 shows a schematic of the intake system, ports, and valves of a four-cylinder automobile spark-ignition engine. It shows the intake air duct leading air past the throttle valve into an enlarged volume of plenum, which then feeds air through the manifold runners and ports and valves in the cylinder head, into the engine's cylinders. Note the relatively long length of the flow path from the plenum to the valves. Such an intake system configuration can be *tuned* in two basic ways: one, based on the length of the flow path—plenum to intake valve—can provide additional airflow at high engine speeds; the other based on air duct and plenum geometry can enhance the airflow at mid-speeds by causing the plenum pressure to oscillate as a Helmoltz resonator.

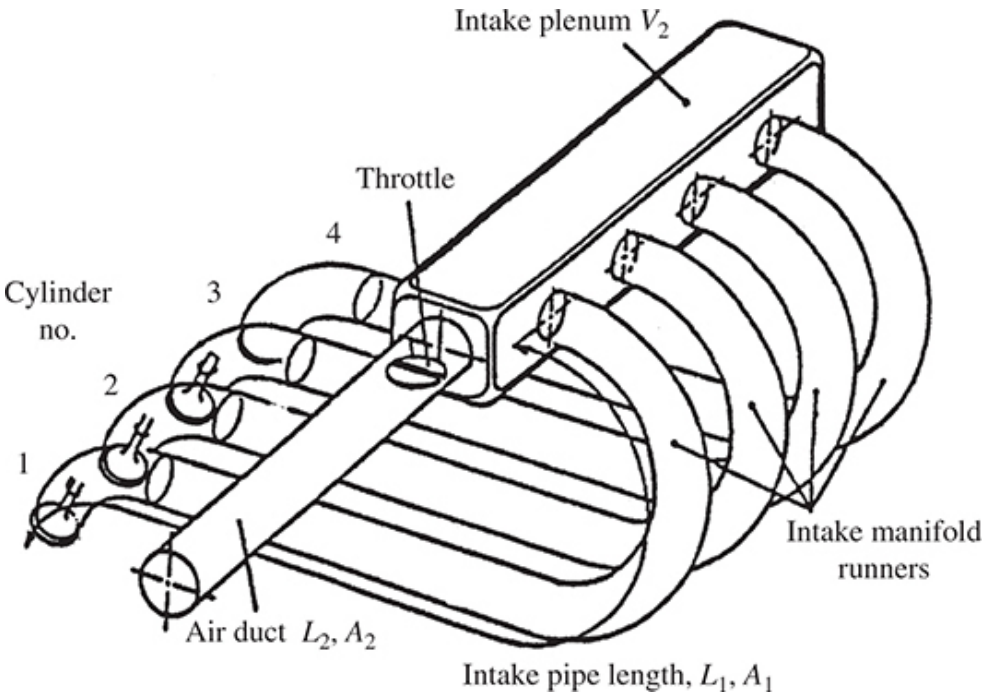


Figure 6.11 Schematic of tuned intake system for a four-cylinder automobile gasoline SI engine: air duct, throttle, plenum, and manifold runners are indicated.[6]

When the intake valve opens (just before TC) and the piston starts accelerating down the cylinder drawing air into the cylinder, the pressure in the intake port falls and a rarefaction (expansion) wave propagates back toward the plenum at the speed of sound. As this expansion wave propagates into the (larger volume) plenum, a compression wave (which increases the

pressure) moves back into the manifold toward the intake valve. If this "high" pressure wave reaches the intake port during the latter part of the intake stroke, then additional air will be "pushed" into the cylinder.

An example of the relevant numbers:

> With L_1 (see Fig. 6.11), the length from the plenum to intake valve seat, equal to 50 cm with the sound speed in air 343 m/s, the pressure wave travel time—intake valve to plenum and back—is 3 ms. This corresponds to the latter 90 degrees of the intake process at about 4500 rev/min, the speed at which the high-speed tuning peak usually occurs.

As engine speed moves away from the tuning peak, the benefits fall off because the match between the pressure wave phenomenon and the phasing of the engine intake flow process becomes increasingly less well matched. Note that to shift these airflow benefits to lower engine speeds would require much longer flow paths from plenum to intake valve (longer manifold runners). This would increase the impact of the manifold runner flow resistance, which is significant at high engine speed; it would also increase the bulk volume of the engine since (as Fig. 6.11 indicates) the runners are already long and the intake system is bulky.

The second tuning mode uses the plenum as a resonator. The flow in the intake air pipe leading to the throttle is the sum of the individual cylinder intake flows. In a four-cylinder four-stroke cycle engine, the beginnings of each cylinder's 720-degree four-stroke cycle are spaced 180 degrees crank angle (CA) apart. Thus, in a four-cylinder engine, there are four flow pulses in this intake pipe every two crank revolutions. This pulsating flow into the intake manifold plenum forces the plenum pressure to oscillate (as a Helmholtz oscillator) at a frequency given by:

$$f_H = \frac{a}{2\pi}\sqrt{\frac{A_2}{L_2 V_2}} \tag{6.7}$$

where a is the speed of sound in the intake air (~340 m/s), and A_2, L_2, and V_2 are defined in Fig. 6.11. If the plenum pressure is higher at about mid-intake-stroke for each cylinder when the airflow "pull" of each piston is at

its maximum, extra air will be "pushed" into each cylinder. Approximate numbers are:

> For a typical automobile four-cylinder gasoline engine, $A_2 \approx 20$ cm^2, $L_2 \approx 40$ cm, $V_2 \approx 1500$ cm^3, Eq. (6.7) gives a Helmholtz resonance frequency of about 100 Hz. This frequency corresponds to high-pressure peaks (twice per revolution for a four-cylinder engine) at an engine speed close to 3500 rev/min: the "in-phase" engine speed for this second tuning mode.

The additional airflow that employing these two tuning modes provide are shown in Fig. 6.12, which adds together all the airflow phenomena or factors discussed in this section (Sec. 6.2) that impact volumetric efficiency. This figure will be discussed fully in Sec. 6.2.7 which follows. Intake manifold tuning provides an additional 10 to 15 percentage points at around 4500 rev/min. Helmholtz resonance in the intake plenum adds an additional 10 or so percentage points between about 2500 and 3500 rev/min.

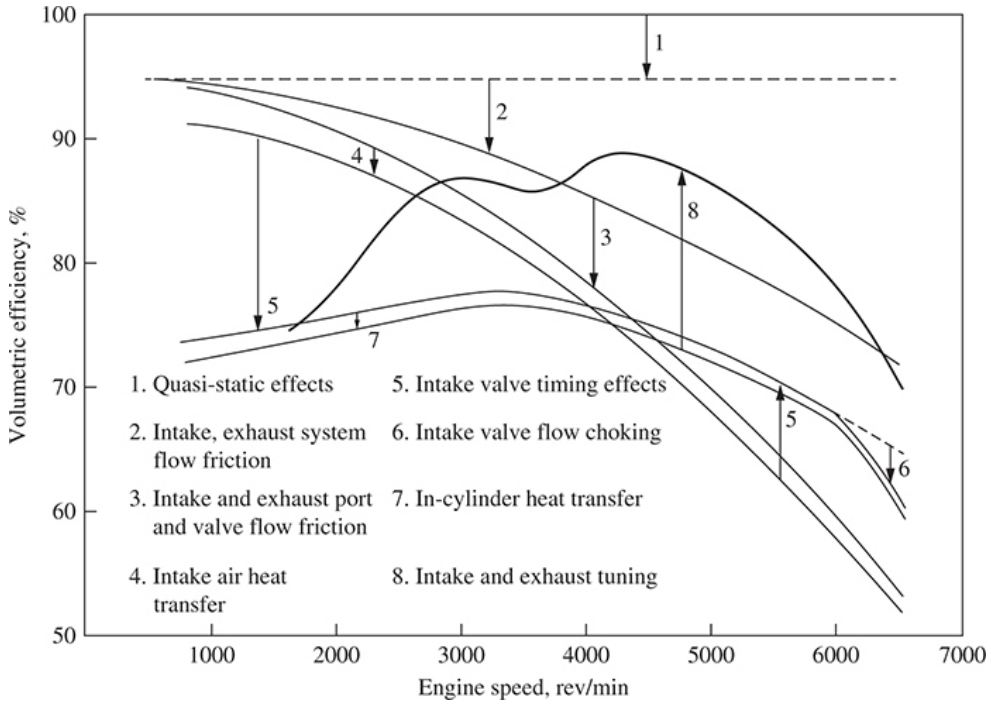


Figure 6.12 Effect on SI engine volumetric efficiency of the different phenomena which affect the airflow rate as a function of engine speed. Bold line is final η_v versus speed curve. [8]

Methods which predict the unsteady flows in the intake and exhaust systems of internal combustion engines with good accuracy have been developed. These methods are complicated, however, so more detailed discussion is deferred to Chap. 14 .

Exhaust System Tuning

The exhaust flow pulses from each individual cylinder are apparent from the low-speed exhaust system pressure traces shown in Fig. 6.10 (also apparent at high speed, but more complex). The total exhaust system—exhaust valves to the atmosphere—has many components (see Fig. 6.4). Some of these components (e.g., mufflers) have additional volume and can act as plenums; these mufflers and the catalyst will provide some damping to flow and pressure oscillations. None-the-less these flow and pressure oscillations will generate wave phenomena which can be used to advantage. If, through suitable layout and geometry, these exhaust pressure wave phenomena result in a *lower* pressure in the exhaust ports during the latter portion of the exhaust stroke, then additional burned gases will be sucked out of the cylinder and the amount of burned gas residual left in the cylinder for the next cycle will be reduced. This increases volumetric efficiency because the (hot) residual gases both occupy volume and, through mixing, heat up the incoming air.

6.2.7 Combined Effects: Naturally-Aspirated Engines

We now combine the effects these several phenomena or factors (discussed in Secs. 6.2.1 through 6.2.6) have on volumetric efficiency, as a function of engine speed.[d] The result shown in Fig. 6.12 is typical of the wide-open-throttle volumetric efficiency versus speed behavior of a modern automobile gasoline engine. The figure shows that the volumetric efficiency of a (fixed-valve-timing, naturally-aspirated engine) steadily increases from about 80% at low speeds to its highest value (95% or more) at about three quarters of the engine's maximum speed, and then declines by about 10% at the speed where airflow choking occurs and the maximum brake power of the engine is reached.

The different phenomena affect volumetric efficiency in the following ways. Quasi-static, non-speed-dependent-effects (such as fuel vapor pressure) drop η_v below 100% (curve 1). Frictional flow losses in the intake and exhaust systems (curve 2), and in the ports and valves in the cylinder head (curve 3), both increase as the square of engine speed. Charge heating in the intake manifold and cylinder head drops the curve modestly to 4. Valve timing effects, the induction ram benefit at higher engine speeds due to the IVC occurring well after BC, and its companion loss of air at lower engine speeds, change curve 4 to curve 5. At the highest engine speeds, the flow into the cylinder during at least part of the intake process becomes chocked. Once this occurs, further increases in speed do not increase the airflow rate (curve 5 to 6) so η_v decreases more steeply. In-cylinder heat transfer heats the intake charge initially, reducing its density and thus the engine's volumetric efficiency, curve 7. Finally, intake system tuning increases the airflow at high speeds by matching the phasing of pressure-wave phenomena generated by the unsteady intake flow in the manifold runners with that of the individual cylinder intake processes; and at mid-speed by creating a Helmholtz resonance in the intake manifold plenum (again appropriately phased to the flows into each cylinder in turn), curve 8. Substantial tuning-generated increases in volumetric efficiency result, as shown.

The factors that have the greatest impacts on this full-load volumetric efficiency versus speed curve are total intake system flow friction, intake-valve closing angle, and intake tuning. In this graph engine speed in rev/min is used: the relevant "speed" is really mean piston speed since piston motion largely drives the intake flow.

6.2.8 Effects of Turbocharging

Turbocharging and supercharging increases the airflow into an engine of a given displaced volume, substantially: see Sec. 6.8 . By increasing the air's density at entry to the engine's cylinders, the inducted air mass can be increased by a factor on the order of two or more. All the phenomena described in this section, and combined together in the previous section, are still relevant and important. But this substantial *boosting*—raising the air's density by raising its pressure significantly [e]—has a major impact. Note that

volumetric efficiency is not as useful an indicator of the total intake system's airflow capabilities: it can be used with $\rho_{a,i}$ in Eq. (6.1) instead of $\rho_{a,0}$ where $\rho_{a,i}$, the air density at the compressor exit (or intercooler exit if utilized), is used to define the performance of that portion of the intake between the compressor or intercooler and the cylinder. A normalized airflow rate parameter—the *specific air consumption*—may be used to quantify the airflow into an engine in a normalized manner (air mass flow rate is divided by the engine's power, analogously to specific fuel consumption, Sec. 2.8), where

$$\text{sac} = \dot{m}_a / P \tag{6.8}$$

which has both brake and indicated values and units of kg of air/kW·h. Effectively, turbocharging substantially changes the baseline level in the combined, naturally-aspirated, volumetric efficiency versus speed plot (Fig. 6.12).

The air exiting the compressor in a turbocharging gasoline engine is at about 100°C (and may be higher if boost pressures above about 2 bar absolute are used). Cooling the intake air in a heat exchanger (intercooler) prior to entry to the engine's cylinders increases its density which increases the mass of air inducted. This both reduces the peak unburned mixture temperatures which reduces the engine's propensity to knock as well as reduces the peak burned gas temperatures which reduces the NO_x emissions. An intercooler, located between the compressor and the throttle, is normally air-cooled since ambient air is available for cooling at temperatures below that of the engine coolant. Coolant-cooled intercoolers are more compact but require a low temperature coolant supply to be effective. These may be used more often, as turbocharged engine outputs for a given displacement increase, as boost levels increase.

Figure 6.13 shows the wide-open-throttle torque and bmep versus speed curves for both a 3.6-liter V-6 turbocharged direct-injection gasoline SI engine and a 5.7-liter V-8 naturally-aspirated port-fuel-injected gasoline engine. The maximum torque and power of the two engines are closely comparable. The maximum bmep values and shapes of the bmep versus speed curves differ significantly. The airflow rates at entry to the cylinders are proportional to bmep from Eq. (2.48) . For the naturally-aspirated engine, the bmep versus speed curve in Fig. 6.13 and the typical volumetric

efficiency curve (Fig. 6.12) have closely comparable shapes.[f] The boosted engine's air mass per cylinder per cycle increases sharply from 1000 to about 1700 rev/min, and then remains roughly constant through the mid-speed range before falling off as the highest speeds are approached. This full load torque curve shape is attractive for automobile applications. The turbocharged gasoline engine behavior shown in Fig. 6.13 is the result of choosing a "smaller" turbocharger (relative to the size of the engine) to achieve rapidly rising torque at low engine speeds: a larger size turbocharger would give a rising air mass per cycle, and bmep (and torque) curves through the mid-speed range.

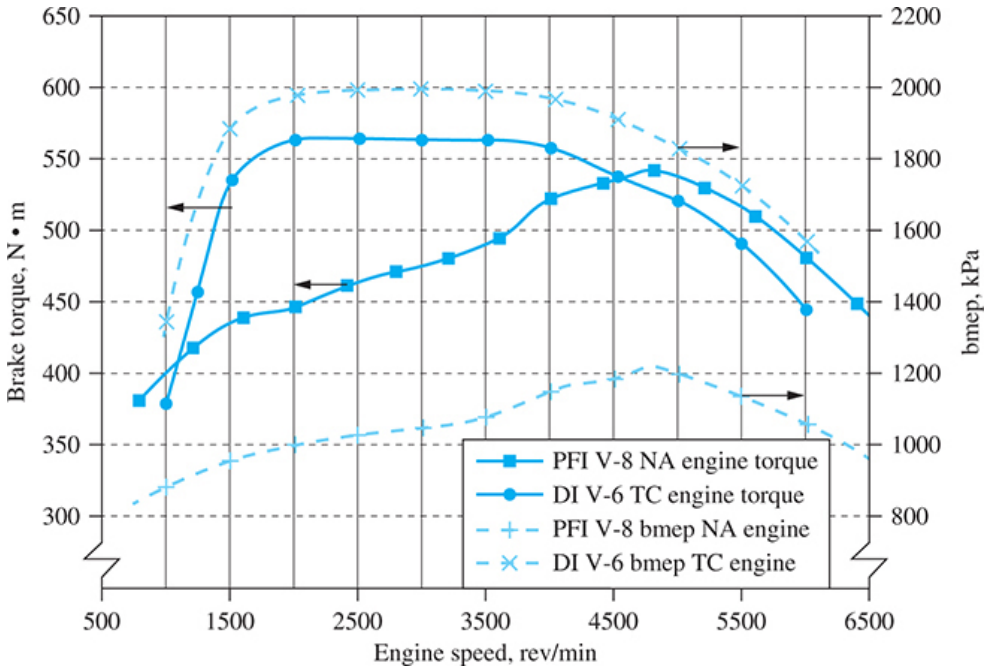


Figure 6.13 Brake torque and bmep as a function of engine speed at wide-open throttle for two gasoline SI engines: a 5.7-liter V-8 two valves per cylinder naturally-aspirated engine, and a 3.6-liter V-6 turbocharged direct-injection engine.

The maximum boost level—intake manifold pressure—for the turbocharged engines shown was about 1.8 bar (absolute) which gives bmep levels of about 2000 kPa. Turbocharged engine boost levels are rising, with bmep values of around 2500 kPa in production and with levels approaching 3000 kPa anticipated.

6.3 FLOW THROUGH VALVES AND PORTS

The valves, or valves and ports together, are usually the most important flow restriction in the intake and the exhaust system of four-stroke cycle engines. The characteristics of flows through engine poppet valves will now be reviewed, in the context of both fixed-valve opening and closing timings and of variable valve timing (VVT) and lift.

6.3.1 Valve and Port Geometry and Operation

Figure 6.14 shows the main geometric parameters of a poppet valve head and seat. Figure 6.15 shows the proportions of typical intake and exhaust valves and ports, relative to the valve seat inner diameter D. The intake port is generally circular, or nearly so, and the cross-sectional area is no larger than is required to achieve the desired power output. For the exhaust port, the importance of good valve seat and guide cooling, with the shortest length of exposed valve stem, leads to a different design. Although a circular cross section is still desirable, a rectangular or oval shape is often essential around the guide boss area.

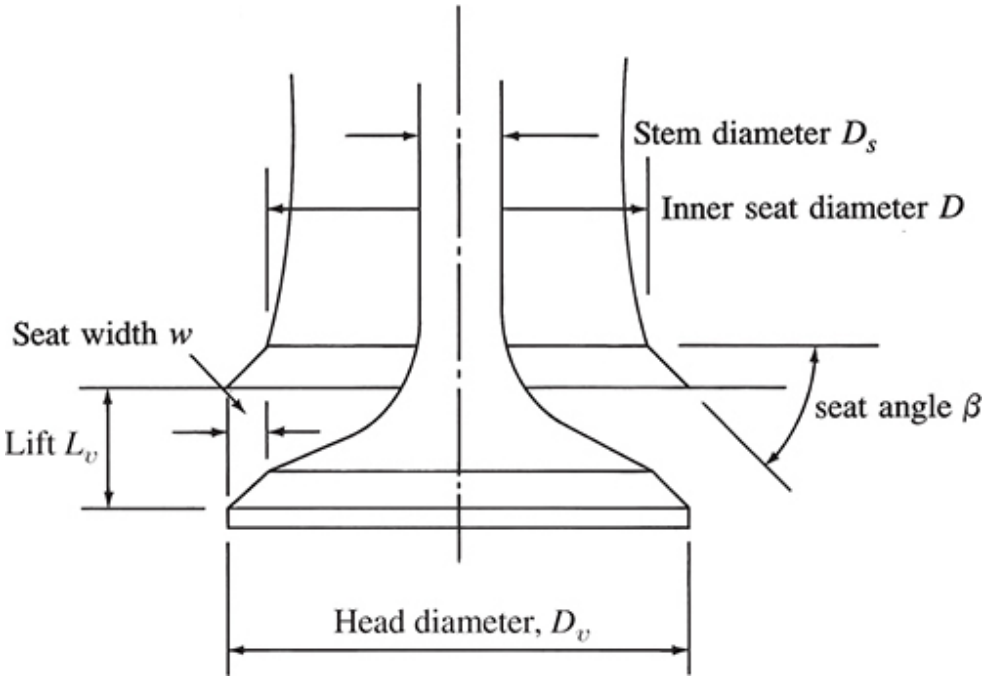


Figure 6.14 Parameters defining poppet valve geometry.

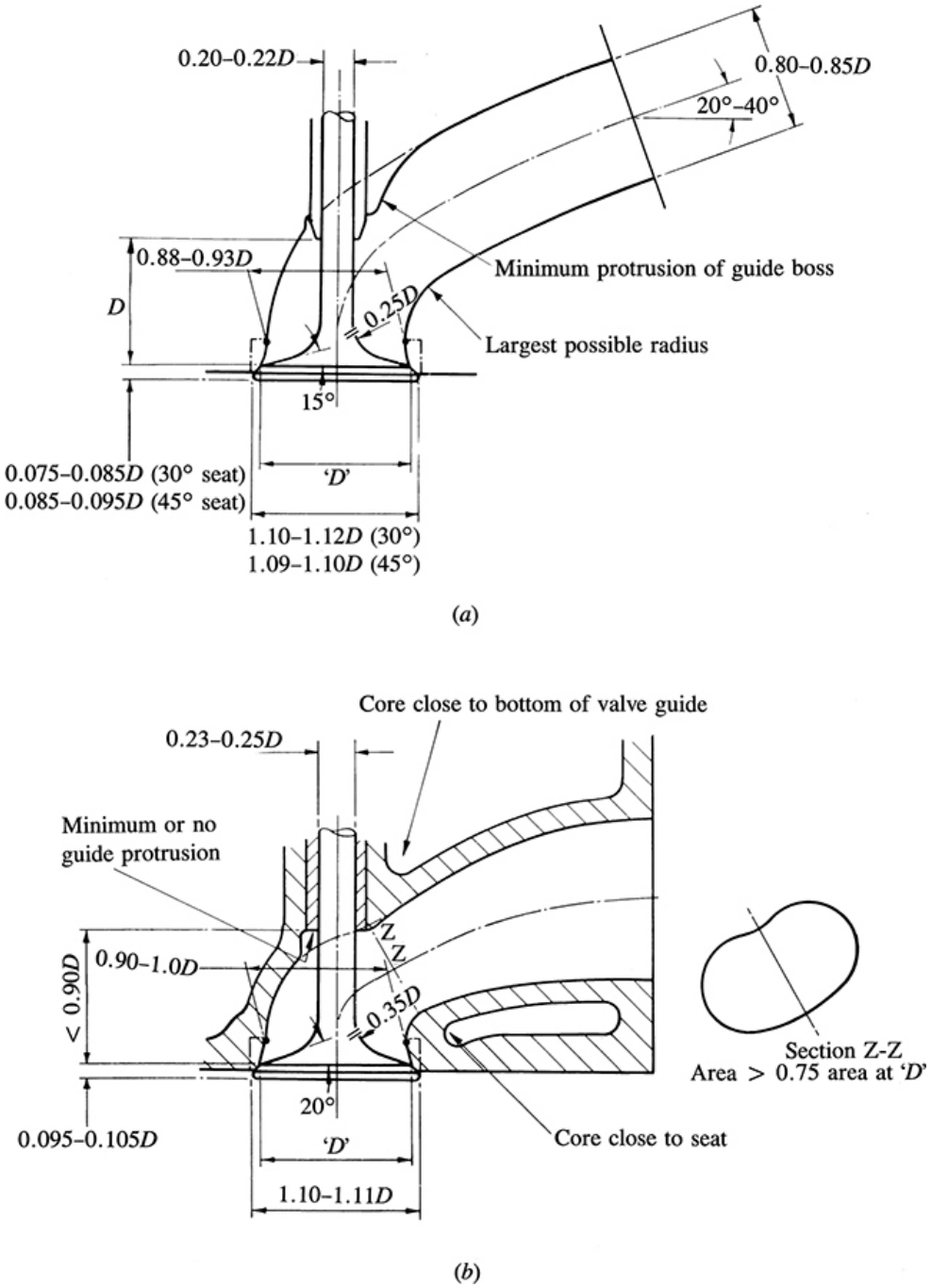


Figure 6.15 Shape, proportions, and critical design areas of typical inlet (top) and exhaust (bottom) valves and ports.[9]

Two, three, four, and even five valves per cylinder engines are in use, in the appropriately shaped cylinder head (of these valves, one, two, or three are intake valves, respectively). With multiple intake and exhaust valves, the

individual valves become smaller while the total valve and port flow area per cylinder increases as the total number of valves increases. Historically, two valves per cylinder designs dominated. Now more than two-thirds of automobile engines have four valves per cylinder. Three valves per cylinder have airflow characteristics close to those of 4-valve engines because the intake valve and port flow areas are comparable and the impact of the smaller single exhaust valve and port area is modest. Variable valve timing and control mechanisms are steadily displacing fixed-valve-timing mechanisms and now dominate because the performance benefits of this flexibility in valve operation (largely through airflow increases) are significant. Variable valve control mechanisms and their operating characteristics are discussed more fully in Sec. 6.3.3 .

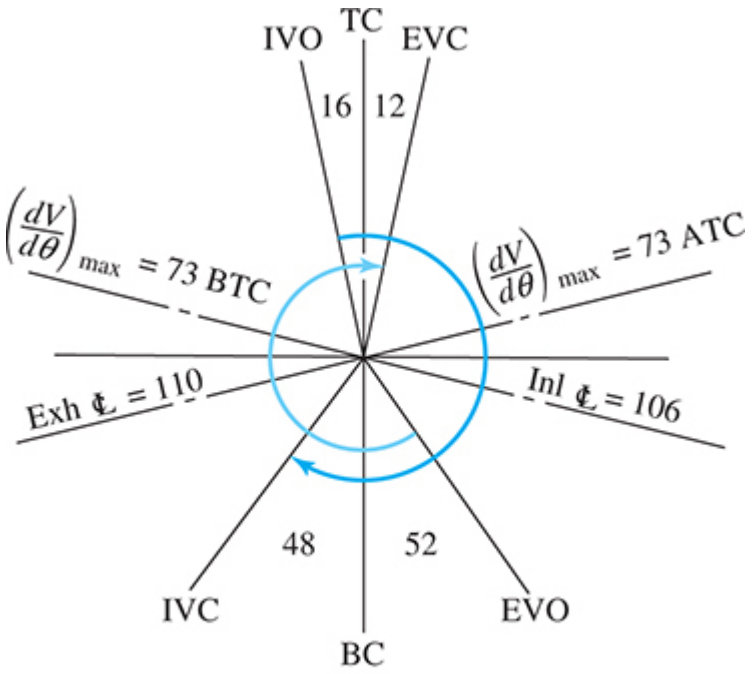
Typical valve head sizes for different shaped combustion chambers in terms of cylinder bore B are given in Table 6.1. [9] Each of these chamber shapes (see Secs. 10.2 and 15.4 for a discussion of spark-ignition and diesel combustion chamber design) imposes different constraints on valve size. Larger valve sizes (or four valves compared with two) allow higher maximum airflows for a given cylinder displacement.

TABLE 6.1 Valve head diameter in terms of cylinder bore B [9]

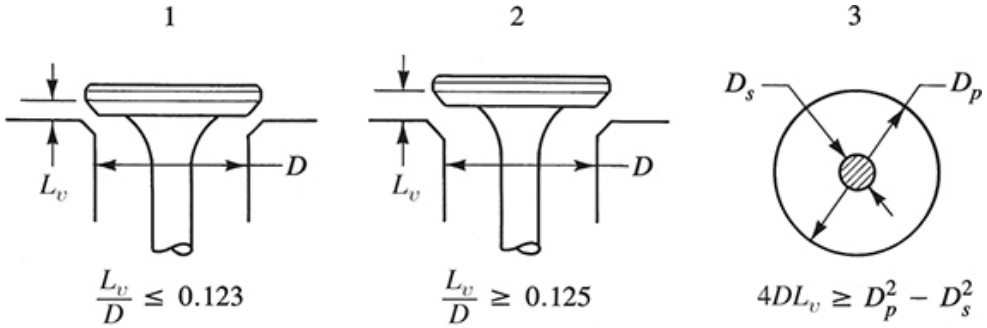
Combustion chamber shape*	Inlet	Exhaust	Approximate mean piston speed, max power, m/s†
Two-valve wedge or bathtub	0.43–0.46B	0.35–0.37B	15–16
Two-valve hemispherical	0.48–0.5B	0.41–0.43B	17
Three-valve open	0.38–0.4B	0.41–0.43B	19
Four-valve pent-roof	0.38–0.41B	0.31–0.35B	20

* See Fig. 15.26.
† For fixed-valve-timing; values about 5% higher for variable valve timing.

Typical valve timing, valve-lift profiles, and valve open areas for a four-stroke cycle fixed-valve-timing spark-ignition engine are shown in Fig. 6.16. There is no universally accepted criterion for defining valve timing points. Some are based upon a specific lift criterion. For example, the Society of Automotive Engineers (SAE) defines valve timing events based on reference valve-lift points: [10]


TC
IVO
EVC
16
12
$\left(\frac{dV}{d\theta}\right)_{max}$ = 73 BTC
$\left(\frac{dV}{d\theta}\right)_{max}$ = 73 ATC
Exh ℄ = 110
Inl ℄ = 106
48
52
IVC
EVO
BC

(*a*)


1
2
3
L_v
D
L_v
D
D_s
D_p
$\frac{L_v}{D} \leq 0.123$
$\frac{L_v}{D} \geq 0.125$
$4DL_v \geq D_p^2 - D_s^2$

(*b*)

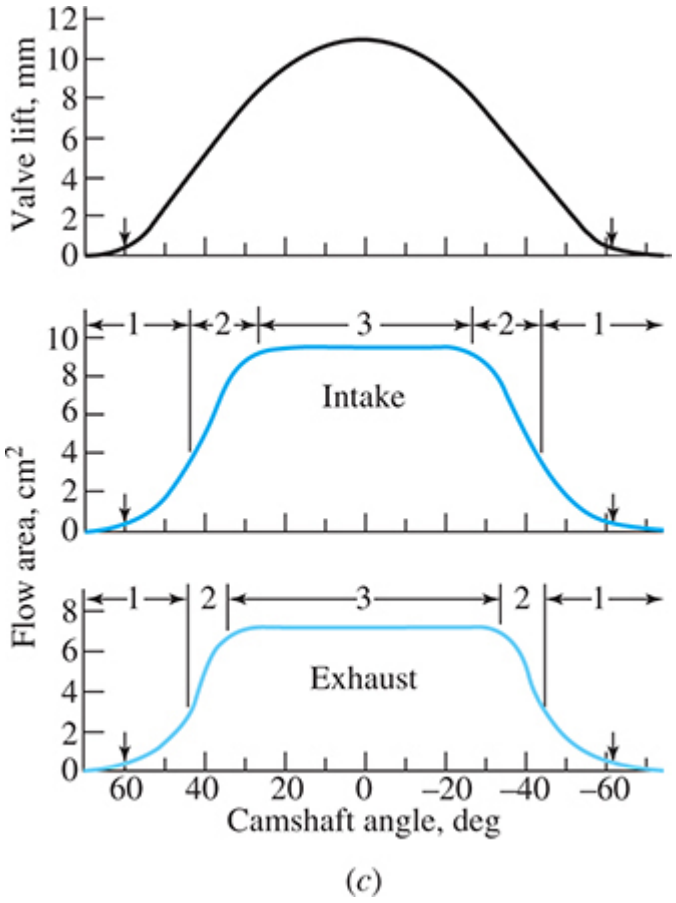


Figure 6.16 (*a*) Typical valve timing diagram for high-speed 2.2-dm^3 four-cylinder fixed-valve-timing spark-ignition engine; (*b*) schematic showing three stages of valve lift; (*c*) valve-lift curve and corresponding minimum intake and exhaust valve open areas as a function of camshaft angle. Intake and exhaust valve diameters are 3.6 and 3.1 cm, respectively. [11]

1. Hydraulic lifters: Opening and closing positions are the 0.15-mm (0.006-in.) valve-lift points.
2. Mechanical lifters: Valve opening and closing positions are the points of 0.15-mm (0.006-in.) lift plus the specified lash.

Alternatively, valve timing events can be defined based on angular criteria along the lift curve. [11] What is important is when significant gas flow through the valve open area either starts or ceases.

The instantaneous valve flow area depends on valve lift and the geometric details of the valve head, seat, and stem. There are three separate stages to

the flow area development as valve lift increases, as shown in Fig. 6.16*b*. [12] For low valve lifts, the minimum flow area corresponds to a frustum of a right circular cone where the conical face between the valve and the seat, which is perpendicular to the seat, defines the flow area. For this stage:

$$\frac{w}{\sin\beta\cos\beta} > L_v > 0$$

and the minimum area is

$$A_{\min} = \pi L_v \cos\beta\, D_v - 2w + \frac{L_v}{2}\sin 2\beta \tag{6.9}$$

where β is the valve seat angle (usually 45° though other angles are used), L_v is the valve lift, D_v is the valve head diameter (the outer diameter of the seat), and w is the seat width (difference between the inner and outer seat radii).

For the second stage, the minimum area is still the slant surface of a frustum of a right circular cone, but this surface is no longer perpendicular to the valve seat. The base angle of the cone increases from $(90 - \beta)°$ toward that of a cylinder, 90°. For this stage:

$$\left[\left(\frac{D_p^2 - D_s^2}{4D_m}\right) - w^2\right]^{1/2} + w\tan\beta \geq L_v \frac{w}{\sin\beta\cos\beta}$$

and

$$A_{\min} = \pi D_m \left[\left(L_v - w\tan\beta\right)^2 + w^2\right]^{1/2} \tag{6.10}$$

where D_p is the port diameter, D_s is the valve stem diameter, and D_m is the mean seat diameter ($D_v - w$).

Finally, when the valve lift is sufficiently large, the minimum flow area is no longer between the valve head and seat; it is the port flow area minus the sectional area of the valve stem. Thus, for

$$L_v > \left[\left(\frac{D_p^2 - D_s^2}{4D_m}\right)^2 - w^2\right]^{1/2} + w\tan\beta$$

then

$$A_{\min} = \frac{\pi}{4}(D_p^2 - D_s^2) \tag{6.11}$$

Intake and exhaust valve open areas corresponding to a typical valve-lift profile are plotted versus camshaft angle in Fig. 6.17*c*. These three different flow regimes are indicated. The maximum valve lift is normally about 12% of the cylinder bore.

Inlet valve opening (IVO) typically occurs 10 to 25° BTC. Engine performance is relatively insensitive to this timing point. It should occur sufficiently before TC so that cylinder pressure does not drop early in the intake stroke. Intake valve closing usually falls in the range 30 to 70° after BC. Later closing provides more time for cylinder filling under conditions where cylinder pressure is below the intake manifold pressure at BC (at higher speeds). IVC is one of the principal factors that determine high-speed volumetric efficiency. It also affects low-speed volumetric efficiency due to backflow into the intake (see Sec. 6.2.4).

Variable valve control allows IVC to be optimized in both these speed ranges. EVO occurs 50 to 60° before BC, well before the end of the expansion stroke, so that the blowdown process can assist in expelling the exhaust gases at higher speeds. The goal here is to reduce the cylinder pressure to close to the exhaust manifold pressure as soon as possible after BC over the full engine speed range. Note that the timing of EVO affects the cycle efficiency since it determines the effective burned gas expansion ratio. Exhaust valve closing (EVC) ends the exhaust process and determines the duration of the valve overlap period. EVC typically falls in the range 8 to 20° after TC. At idle and light load, in spark-ignition engines (which are throttled), it therefore regulates the quantity of exhaust gases that flow back into the combustion chamber through the exhaust valve under the influence of intake manifold vacuum. At high engine speeds and loads, it regulates how much of the cylinder burned gases are exhausted. EVC timing should occur sufficiently far after TC that the cylinder pressure does not rise near the end

of the exhaust stroke. Late EVC favors high power at the expense of low-speed torque and idle combustion quality. Note from the timing diagram (Fig. 6.17*a*) that the points of maximum valve lift and of maximum piston velocity (Fig. 2.2) do not coincide.

The effect of valve geometry and timing on air flow can be illustrated conceptually by dividing the rate of change of cylinder volume by the instantaneous minimum valve flow area to obtain a *pseudo flow velocity* for each valve: [11]

$$v_{ps} = \frac{1}{A_{min} d\theta} = \frac{\pi \beta^2}{4 A_{min}} \frac{ds}{d\theta} \tag{6.12}$$

where V is the cylinder volume [Eq. (2.4)], B is the cylinder bore, s is the distance between the wrist pin and crank axis [see Fig. 2.1 and Eq. (2.5)], and A_{min} is the valve area given by Eqs. (6.9) , (6.10), or (6.11).

Instantaneous pseudo-flow-velocity profiles for the exhaust and intake strokes of a four-stroke four-cylinder engine are shown in Fig. 6.17. Note the appearance of two peaks in the pseudo flow velocity for both the exhaust and intake strokes. The broad peaks occurring at maximum piston velocity reflect the fact that valve flow area is constant at this point. The peaks close to TC result from the exhaust valve closing and intake valve opening profiles. The peak at the end of the exhaust stroke is important since it indicates a significant pressure drop across the valve at this point, which will result in higher trapped residual mass. The magnitude of this exhaust stroke pseudo-velocity peak depends strongly on the timing of EVC. The pseudo-velocity peak at the start of the intake stroke is much less important. That the pseudo velocities early in the exhaust stroke and late in the intake stroke are low indicates that phenomena other than quasi-steady flow govern the flow rate. These are the periods when exhaust blowdown, and ram and tuning effects in the intake, are most important.

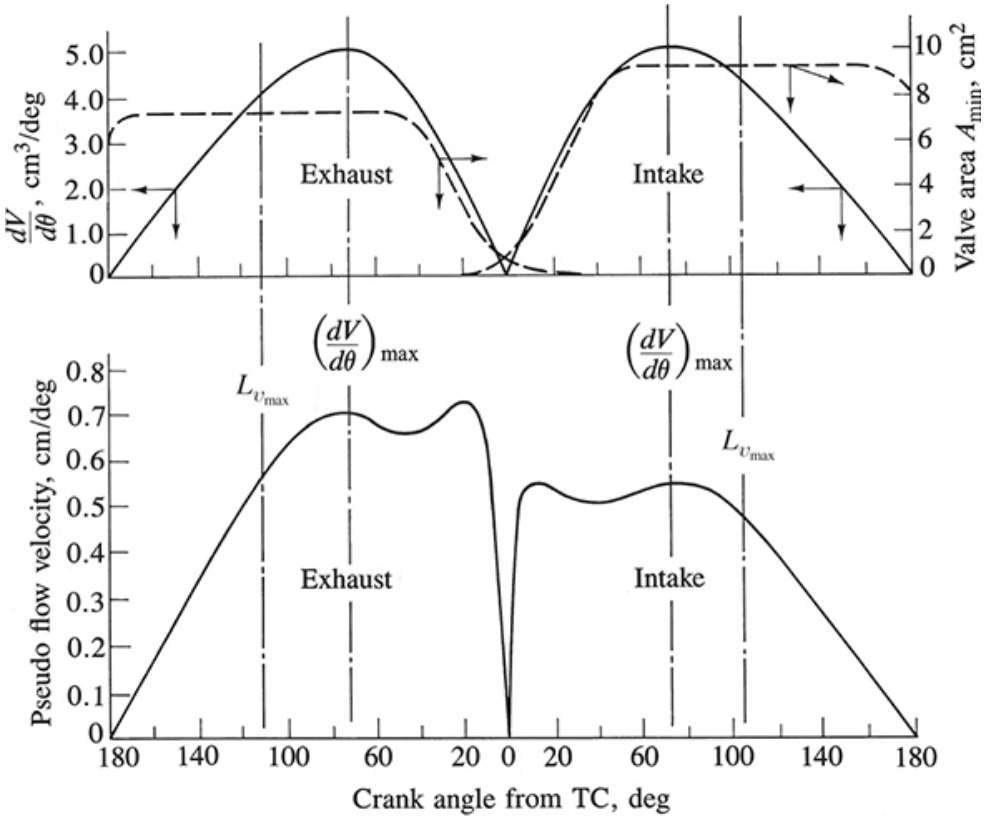


Figure 6.17 Rate of change of cylinder volume $dV/d\theta$, valve minimum flow area A_{min}, and pseudo flow velocity as functions of crank angle for exhaust and intake valves of Fig. 6.16.[11]

6.3.2 Flow Rates and Discharge Coefficients

The mass flow rate through a poppet valve is usually described by the equation for compressible flow through a flow restriction [Eq. (C.8) or (C.9) in App. C]. This equation is derived from one-dimensional isentropic flow analysis, and real gas flow effects are included by means of an experimentally determined discharge coefficient C_D. The airflow rate is related to the upstream stagnation pressure p_0 and stagnation temperature T_0, static pressure just downstream of the flow restriction (assumed equal to the pressure at the restriction, p_T), and a reference area A_R characteristic of the valve design:

$$\dot{m} = \frac{C_D A_R p_0}{RT_0^{1/2}} \left(\frac{p_T}{p_0}\right)^{1/\gamma} \left\{\frac{2\gamma}{\gamma - 1}\left[1 - \left(\frac{p_T}{p_0}\right)^{(\gamma - 1/\gamma)}\right]\right\}^{1/2} \tag{6.13}$$

When the flow is choked, that is, $p_T / p_0 \leq [2/(\gamma + 1)]^{\gamma/(\gamma - 1)}$ the appropriate equation is

$$\dot{m} = \frac{C_D A_R p_0}{(RT_0)^{1/2}} \gamma^{1/2} \left(\frac{2}{\gamma + 1}\right)^{(\gamma+1)/2(\gamma-1)} \tag{6.14}$$

For flow into the cylinder through an intake valve, p_0 is the intake system pressure p_i, *and* p_T is the cylinder pressure. For flow out of the cylinder through an exhaust valve, p_0 is the cylinder pressure and p_T is the exhaust system pressure.

The value of C_D and the choice of reference area are linked together: their product, $C_D A_R$, is the effective flow area of the valve assembly, A_E. Several different reference areas have been used. These include the valve head area $\pi D_v^2/4$, [13] the port area at the valve seat $\pi D_p^2/4$, [14] the geometric minimum flow area [Eqs. (6.9) , (6.10), and (6.11)], and the curtain area $\pi D_v L_v$ [15] where L_v is the valve lift. The choice is arbitrary, though some of these choices allow easier interpretation than others. As has been shown above, the geometric minimum flow area is a complex function of valve and valve seat dimensions. The most convenient reference area in practice is the so-called valve curtain area:

$$A_C = \pi D_v L_v \tag{6.15}$$

since it varies linearly with valve lift and is simple to determine.

Intake Valves

Figure 6.18 shows the results of steady-flow tests on a typical inlet valve configuration with a sharp-cornered valve seat.[15] The discharge coefficient based on valve curtain area is a discontinuous function of the valve-lift/diameter ratio. The three segments shown correspond to different flow regimes as indicated. At low lifts, the flow remains attached to the valve

head and seat, giving high values for the discharge coefficient. At intermediate lifts, the flow separates from the valve head at the inner edge of the valve seat as shown. An abrupt decrease in discharge coefficient occurs at this point. The discharge coefficient then increases with increasing lift since the size of the separated region remains approximately constant while the minimum flow area is increasing. At high lifts, the flow separates from the inner edge of the valve seat as well. [12] Typical maximum values of L_v/D_v are 0.25.

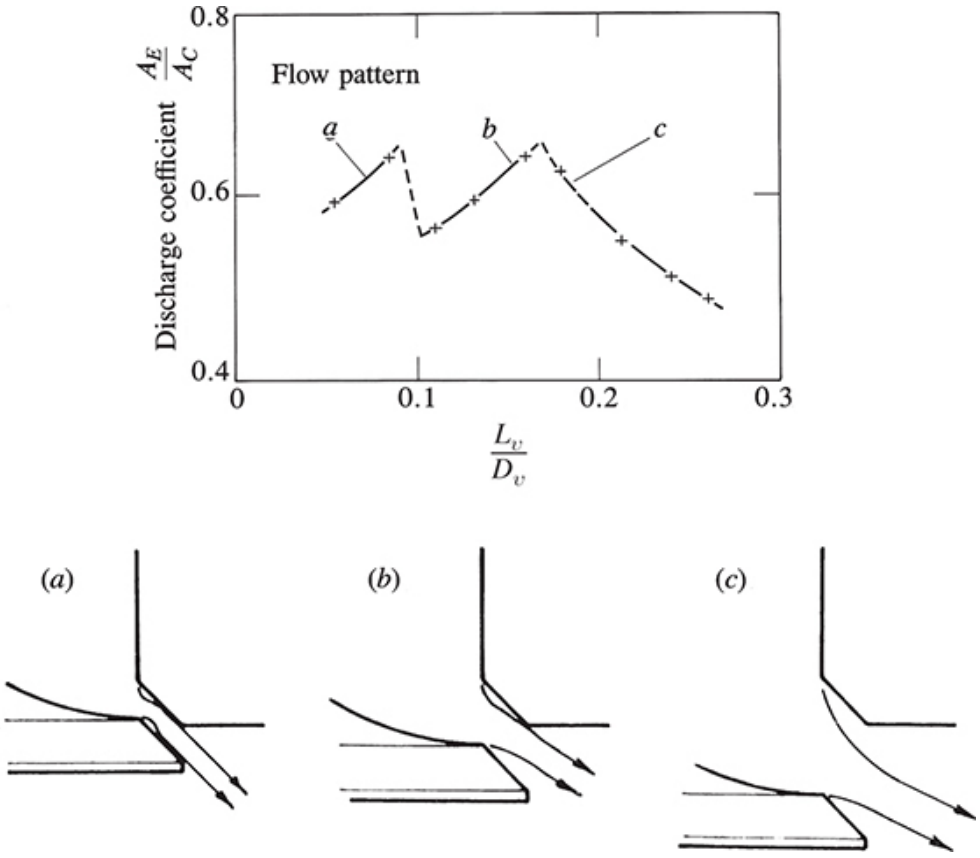


Figure 6.18 Discharge coefficient of typical intake poppet valve (effective flow area/valve curtain area) as a function of valve lift. Different segments correspond to the flow regimes indicated. [15]

An important question is whether these steady-flow data are representative of the dynamic flow behavior of the valve in an operating engine. There is some evidence that the "change points" between different flow regimes shown in Fig. 6.18 occur at slightly different valve lifts under

dynamic operation than under steady-flow operation. Also, as shown in Fig. 6.10, the pressure upstream of the valve varies significantly during the intake process. However, it has been shown that over the normal engine speed range, steady-flow discharge-coefficient results can be used to predict dynamic performance with reasonable precision. [12, 16]

In addition to valve lift, the performance of the intake valve assembly is influenced by the following factors: valve seat width, valve seat angle, rounding of the seat corners, port design, and cylinder head shape. In many engine designs the port and valve assembly are used to generate a rotational motion (*swirl*) inside the engine about the cylinder axis during the induction process: for example, the cylinder head can be shaped to restrict the flow through one side of the valve open area and thereby generate swirl. Swirl production is discussed later, in Sec. 8.3 . Swirl generation significantly reduces the valve (and port) flow coefficient. The intake ports can also be shaped to guide the airflow preferentially through the upper portion of the intake valve open area (between the valve and the center of the cylinder head), and across the cylinder. This generates a *tumbling* flow—a rotating motion in the plane including the cylinder axis—as the piston travels down the intake stroke. Swirl and tumble are discussed later, in Sec. 8.3 .

Changes in seat width affect the L_v/D_v at which the shifts in flow regimes illustrated in Fig. 6.18 occur. C_D increases as seat width decreases. The seat angle β affects the discharge coefficient in the low-lift regime in Fig. 6.18. Rounding the upstream corner of the valve seat reduces the tendency of the flow to break away thus increasing C_D at higher lifts. At low valve lifts, when the flow remains attached, increasing the Reynolds number decreases the discharge coefficient. Once the flow breaks away from the wall, there is no Reynolds number dependence of C_D. [15]

For well-designed ports (e.g., Fig. 6.15) the discharge coefficient of the port and valve assembly need be no lower than that of the isolated valve (except when the port is used to generate swirl or tumble). However, if the cross-sectional area of the port is not sufficient or the radius of the surface at the inside of the bend is too small, a significant reduction in C_D for the assembly can result. [15]

At high engine speeds, unless the intake valve is of sufficient size, the inlet flow during part of the induction process can become choked (i.e., reach

sonic velocity at the minimum valve flow area; see Sec. 6.2.5). Choking substantially reduces volumetric efficiency. Various definitions of inlet Mach number have been used to identify the onset of choking. Taylor and coworkers [13] correlated volumetric efficiencies measured on a range of engine and intake valve designs with an *inlet Mach index Z* formed from an average gas velocity through the intake valve:

$$Z = \frac{A_p \bar{S}_p}{C_i A_i a} \tag{6.16}$$

where A_i is the nominal inlet valve area ($\pi D_v^2/4$), C_i is a mean valve discharge coefficient based on the area A_i, and a is the sound speed. From the method used to determine C_i, it is apparent that $C_i A_i$ is the average effective open area of the valve (it is the average value of $C_D \pi D_v L_v$): Z corresponds closely, therefore, to the mean Mach number in the intake valve throat. Taylor's correlations show that η_v decreases rapidly for $Z \geq 0.5$.

An alternative equivalent approach to this problem has been developed, based on the average flow velocity through the valve during the period the valve is open. [17] A *mean inlet Mach number* was defined:

$$\bar{M}_i = \frac{\bar{v}_i}{a} \tag{6.17}$$

where $\bar{v}_i$ is the mean intake flow velocity during the valve open period. $\bar{M}_i$ is related to Z via

$$\bar{M}_i = \frac{Z\left(\eta_v/100\right)180}{\theta_{\mathrm{IVC}} - \theta_{\mathrm{IVO}}} \tag{6.18}$$

This mean intake Mach number correlates volumetric efficiency characteristics better than the Mach index. For a series of small modern four-cylinder engines, when M_i approaches 0.5 the volumetric efficiency decreases significantly more rapidly. This is due to the flow becoming choked during part of the intake process. This relationship can be used to

size the intake valve/port combination for the optimum volumetric efficiency at maximum engine speed.

Exhaust Valves

In studies of the flow from the cylinder through an exhaust poppet valve, different flow regimes at low and high lift occur, as shown in Fig. 6.19. Values of C_D based on the valve curtain area, for several different exhaust valve and port combinations, are given in Fig. 6.20. A sharp-cornered isolated poppet valve (i.e., straight pipe downstream, no port) gives the best performance.

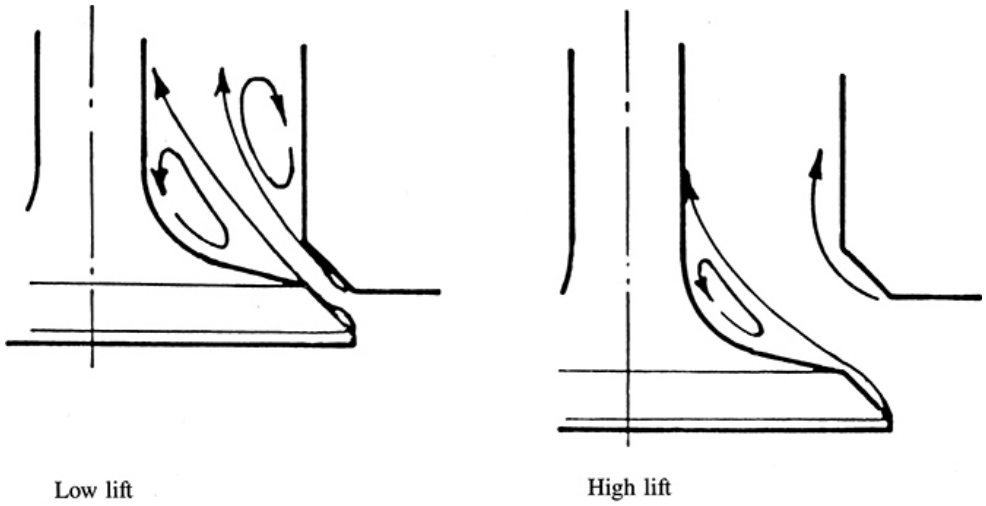


Figure 6.19 Flow pattern through exhaust valve at low and high lift.[15]

At high lifts, $L_v/D_v \geq 0.2$ the breakaway of the flow reduces the discharge coefficient. (At $L_v/D_v = 0.25$ the effective area is about 90% of the minimum geometric area. For $L_v/D_v < 0.2$ it is about 95%. [15]) The port design significantly affects C_D at higher valve lifts, as indicated by the data from the four port designs in Fig. 6.20. Good designs can approach the performance of isolated valves, however. Exhaust valves operate over a wide range of pressure ratios (1 to 5). For pressure ratios greater than about 2 the flow will be choked, but the effect of pressure ratio on discharge coefficient is small and confined to higher lifts (e.g., ±5% at $L_v/D_v = 0.3$). [14]

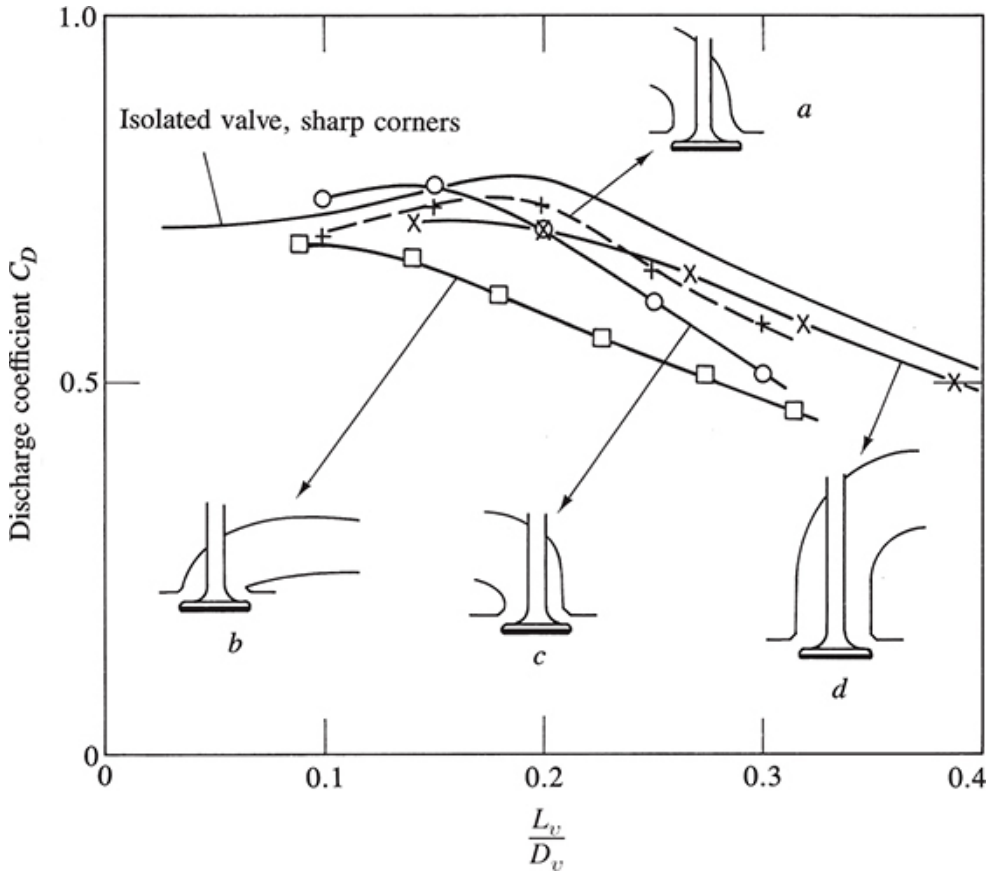


Figure 6.20 Discharge coefficient as function of valve lift for several exhaust valve and port designs:[15] *a*,[18] *b*,[17] *c*,[18] *d*.[19]

6.3.3 Variable Valve Timing and Control

Benefits

Increasingly, variable control of valve operation as a function of engine load and speed is being used instead of fixedvalve timing and operation, which used to be the standard approach. The simplest approach—rotation of the camshafts relative to the crankshaft—varies the phasing of the intake and exhaust camshafts via a cam phaser, to improve the engine's airflow capability at high speed (thus allowing the maximum speed to be increased), and increase the torque at low speed. More sophisticated (variable valve control—VVC) valve-actuation mechanisms are also used which can vary valve timing, open duration, and lift. While greater benefits result from controlling the intake valves, there are additional benefits from varying the exhaust valve behavior: increasing the effective expansion ratio of the burned

gases before EVO (Miller and Atkinson cycles; see Sec. 5.6); controlling the in-cylinder residual burned gas fraction to improve idle and light-load combustion stability and reduce NO_x and HC emissions. Variable valve control also has the potential for reducing the engine's pumping work at lighter loads: see Sec. 13.7 . As already mentioned in Sec. 6.2 , and shown in Fig. 6.21, later IVC (and higher valve lift) increase the air charging of the cylinder at high speed and allow the piston speed to be increased before airflow choking limits are reached (see Sec. 6.2.5). Early IVC at low speed decreases the backflow of air (plus residual) into the intake during the early part of the compression stroke. The engine full load torque benefits are apparent in Fig. 6.9. Note that airflow gains at high speed (and increasing maximum engine speed) increase the maximum brake power of the engine. At fixed engine-in-vehicle performance, a downsized engine can then be used, which at part-load is less throttled, has lower friction, and thus has a higher brake fuel-conversion efficiency.

Valve Timing Characteristics for Performance Objectives

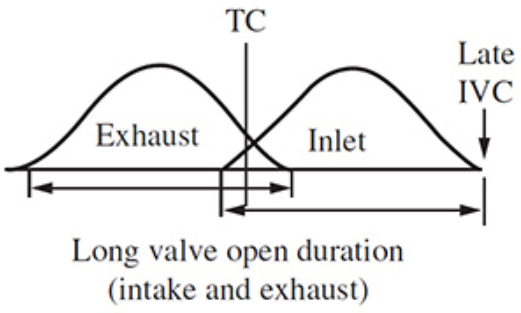
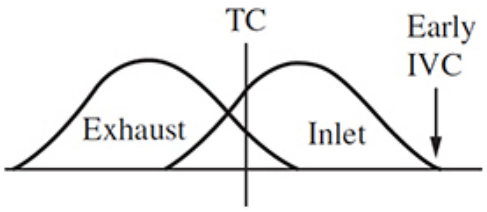
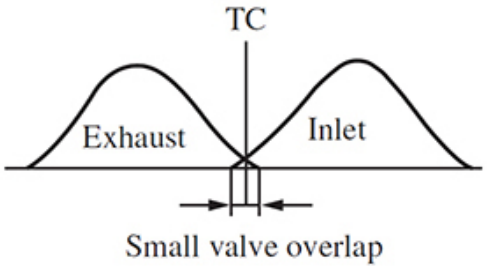
Engine performance objectives	Required valve timing characteristics
Greater maximum power	
Greater low-speed torque	
Better idle combustion stability	

Figure 6.21 Desired exhaust and intake valve timings as a function of the three listed performance objectives.

Mechanical Approaches

Over the years, many valve control mechanisms have been developed, and a number of these are now used in production automobile and truck engines. One simple approach in multi-cylinder gasoline engines is cylinder cut out at lighter engine loads. The valves (intake and exhaust) are deactivated in some of the engine's cylinders (e.g., 4-cylinders out of 8; 2-cylinders out of 4). The displaced volume of the firing cylinders is reduced: thus at the same output

torque each of these (reduced number of cylinders) is operating more highly loaded with a higher intake manifold pressure and reduced pumping work. While the mechanical friction of the non-firing cylinders is still present, a net average-driving fuel-consumption benefit of some 5% results from deactivating half the engine's cylinders. [20]

Figure 6.22 provides a description and assessment of the various mechanistic approaches to variable valve control that are used in practice. These different approaches are illustrated by the changes in intake valve lift profile each mechanism employs. The figure also provides a rating on an excellent to poor scale of their normalized power, torque, fuel consumption, and emissions characteristics, and number of parts (or their complexity).

Type	Description	Power	Torque	Fuel consumption	Emissions	Number of parts
Varying cam phasing (continuous)		◎	◎	○	◎	◎
Switching between different cam profiles		◎	○	△	△	△
3-D cam profiles (and lateral shift)		◎	◎	○	○	×
Fully variable lift and timing (with camshaft)		○	◎	◎	○	×
Electromagnetic valve drive		○	◎	◎	◎	×

◎ Excellent ○ Good △ Medium × Poor

Figure 6.22 Schematic of intake valve timing and lift profiles for five different valve control approaches/mechanisms, and their rankings on power, torque, fuel consumption, emissions, number of parts.[21]

Cam Phasing

Continuously adjusting the phasing of the intake cam and the exhaust cam relative to the crankshaft is the commonest approach. An intake valve closing adjustment range of about 40 degrees (± 20 degrees about 50°ABC) would be typical. A similar range is used for exhaust valve phasing. It is low cost and relatively effective at increasing low-speed torque (10%), high-speed torque (5%), and maximum engine speed (up to 5%), relative to fixed-valve-timing levels. At idle, the camshafts are set for later intake and exhaust valve opening and closing, respectively, to reduce the valve overlap period, and thus the exhaust backflow and in-cylinder residual, to produce more stable idle combustion, allow lower idle speeds, and reduced idle fuel consumption. At high loads and low-to-mid speeds, the intake cam is rotated to close the intake valve earlier to reduce the amount of inducted air pushed back from the cylinder into the intake during the first part of the compression stroke. At mid speeds, normal IVC angles are used: at high speeds, the cam is rotated to retard IVC to take advantage of the extra air inducted due to intake manifold ram charging (see Sec. 6.2.4).

Cam Switching

With cam profile switching, the intake valve motion is driven by two (or even three) cams: a lower-lift shorter-duration intake cam profile is used at lower speeds; at higher speed, a higher-lift longer-duration cam profile is used. This can be achieved, for example, with an additional rocker arm that rides freely on the high-speed cam at low speeds when the low-speed cam is controlling, which is then connected/pinned to the low-speed rocker arm (and thus overrides it) at high speeds since the high-speed cam is larger (has higher lift and longer open duration). Cam shifting can provide a larger increase in maximum power (as a result of the larger high-speed intake-valve cam profile), and can also more readily give low valve overlap at low speed and thus good torque along with excellent idle combustion quality.

Fully Variable Valve Control

A valve control system that permits variations in lift, open duration, and timing is termed *fully variable*. One approach is to use cams with curved three-dimensional profiles and lateral shift capability which provide substantial degrees of freedom. This type of camshaft control enables fully variable adjustment of intake valve lift, opening angle, and open duration. An

alternative is the valvetronic system [22] where an electric motor rotates an eccentric shaft that varies the interaction between the intake cams and the cam roller followers that actuate the valves. Low intake valve lifts are used to throttle the airflow, and the conventional throttle valve can be removed. Such approaches, in addition to improving full load airflow, can be used to increase the in-cylinder flow velocities and turbulence levels and thus enhance the combustion rate and stability. Figure 6.23 illustrates various intake valve lift profiles, where valve lift is varied to both control (throttle) the airflow into the cylinder and generate in-cylinder flows that enhance combustion and improve emissions control (e.g., with different lift profiles for the two intake valves).

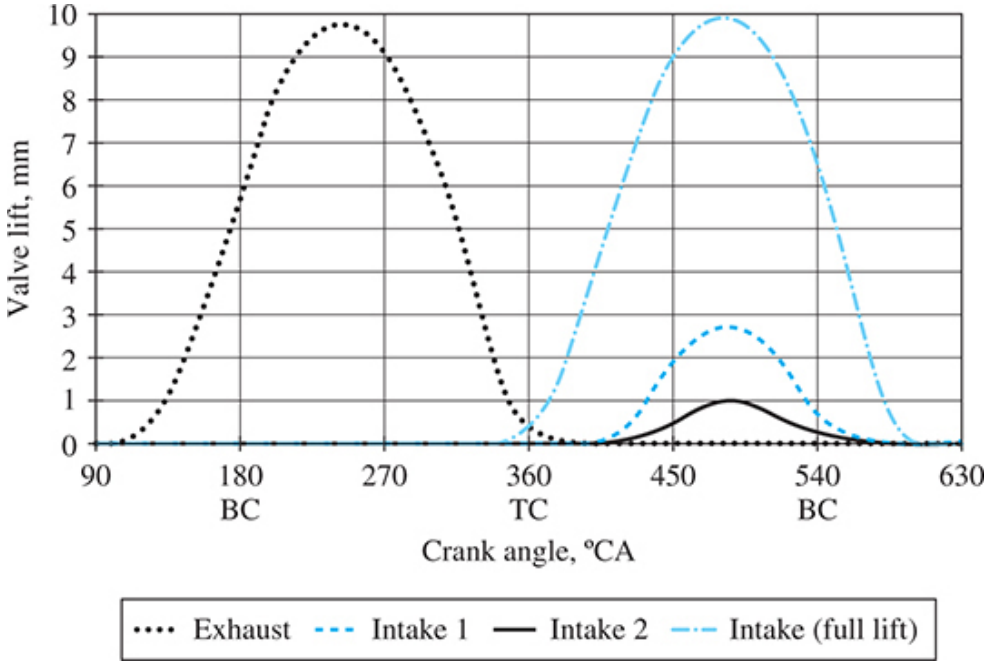


Figure 6.23 Exhaust and intake valve lift profiles for the BMW Valvetronic variable valve control mechanism. Changes in intake valve lift and duration variations are used to throttle the airflow as engine load is reduced. [22]

The final valve control mechanism shown in Fig. 6.22 is electrical rather than mechanical valve actuation, where valves are operated electromagnetically with solenoids or by electro-hydraulic actuators. Such systems, without camshafts, can provide fully flexible valve control and thus have significant potential for improving torque, fuel consumption, and

emissions. The challenges to deploying this type of technology are the substantial forces required (which increase with the square of engine speed), noise, drive energy requirements to overcome valvetrain friction, packaging, and cost.

Pumping Work Reduction

In spark-ignition engines at part-load, the intake airflow is throttled to below atmospheric pressure. Thus, during the exhaust and intake strokes of the standard four-stroke cycle, work is transferred *to* the piston. In naturally-aspirated engines this *pumping work* is counted as a part of engine friction (see Secs. 2.5 and 13.7). Variable valve control can reduce this pumping work loss, which in the ideal four-stroke-cycle model (Sec. 5.2) is given by $(p_e - p_i)V_d$. At a given (light-load) torque, the mass of (stoichiometric) fuel-air mixture burned in the cylinder is proportional to that torque. The mass trapped in the cylinder is given by the ideal gas law, applied at the intake-valve-closing (IVC) location:

$$p_{\text{cyl}} V_{\text{IVC}} = m_{\text{cyl}} R T_{\text{cyl}} \tag{6.19}$$

About the same mass of fuel and air is required to produce this torque as the IVC angle changes (the change in gross-indicated fuel-conversion efficiency is modest). Also, T_{cyl} changes little. Thus as V_{IVC} becomes smaller, p_{cyl} during the intake process must be increased, and the pumping work is therefore reduced. Figure 6.24 shows the fuel consumption gains that result as the cylinder volume at IVC is reduced either by *later* IVC during the compression stroke, or much *earlier* IVC during the intake stroke. A benefit in *net-indicated* specific fuel consumption of about 8% relative to typical fixed IVC operation results. Because variable valve control can change the cylinder pressure-volume relationship during the compression and expansion strokes, defining pumping work for these modified cycles requires a more sophisticated approach. See Sec. 13.7 where this topic is discussed in more detail.

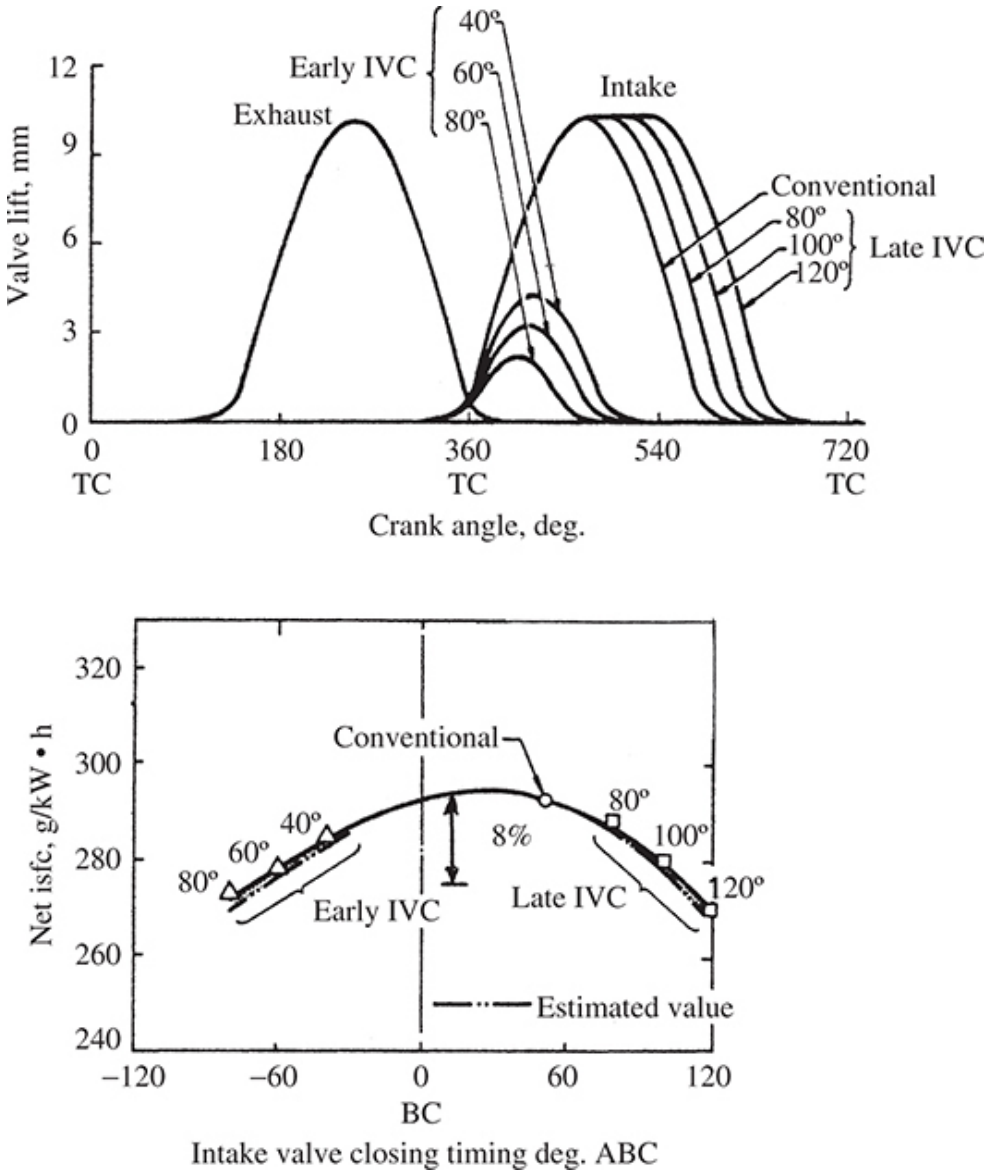


Figure 6.24 Illustration of how variations of intake valve timing and lift (with both early and late IVC) improve net isfc by reducing the engine's pumping friction work.[23] Engine conditions: 1400 rev/min, torque 29 N·m, *A/F* 14.5, MBT timing, no EGR.

6.4 RESIDUAL GAS FRACTION

The residual gas fraction in the cylinder during compression is determined by the details of the exhaust and intake processes. Its magnitude affects volumetric efficiency and engine performance directly, and efficiency and

emissions through its effect on working-fluid thermodynamic properties and on combustion. The residual gas fraction is primarily a function of intake and exhaust pressures, engine speed, compression ratio, valve timing, and exhaust system dynamics.

The residual gas mass fraction x_r (or burned gas fraction in the in-cylinder unburned mixture when EGR is used) is usually determined by measuring the CO_2 concentration in a sample of gas extracted from the cylinder during the compression stroke. Then

$$x_r = (\tilde{x}_{CO_2})_c / (\tilde{x}_{CO_2})_e \tag{6.17}$$

where the subscripts c and e denote compression and exhaust and $\tilde{x}_{CO_2}$ are mole fractions in the wet gas. Usually CO_2 volume or mole fractions are measured in dry gas streams (see Sec. 4.9). A correction factor K,

$$K = \frac{(\tilde{x}_i)_{wet}}{(\tilde{x}_i)_{dry}} = \frac{1}{1 + 0.5\left[y\left(\tilde{x}^*_{CO_2} + \tilde{x}^*_{CO}\right) - 0.74\tilde{x}^*_{CO}\right]} \tag{6.18}$$

where y is the molar hydrogen/carbon ratio of the fuel, and $\tilde{x}^*_{CO_2}$, $\tilde{x}^*_{CO}$ are dry mole fractions, can be used to convert the dry mole fraction measurements to wet mole fractions.

The residual gas mass in the cylinder is the result of two (additive) processes: (1) the gas trapped in the cylinder just before IVO when the valve overlap flow commences, m_{cyl}; (2) the backflow of burned gas from the exhaust port into the cylinder during the valve overlap period (from IVO to EVC), m_{exh}; see Figs. 6.1 and 6.16. Thus, m_r is

$$m_r = m_{cyl,IVO} + \int_{\theta_{IVO}}^{\theta_{EVC}} \dot{m}_{exh}\, dt \tag{6.19}$$

All the burned gas that flows from the cylinder into the intake port during the pressure equalization process (cylinder pressure drops to the intake manifold pressure as this backflow occurs) is assumed to flow back into the cylinder during the subsequent intake process. [24] The residual mass fraction x_r, as determined from ideal gas cycle analysis and the in-cylinder mass at

intake valve opening (at TC), is given by Eq. (5.35) . Its temperature is given by Eq. (5.36) .

The residual gas fraction component due the valve overlap backflow is given by:

$$x_{r,\text{backflow}} = \int \frac{\dot{m}_{\text{exh}}}{\rho_a V_d \left[\left(r_c+1\right)/r_c\right]} \frac{1}{6N} dt \tag{6.20}$$

and the total residual gas fraction trapped in the cylinder (after IV closing) is

$$x_r = x_{r,\text{IVO}} + x_{r,\text{backflow}} \tag{6.21}$$

Several models of the combination of these two processes have been developed and correlated against measured data. Figure 6.25 shows the backflow residual fraction, trapped fraction, and total residual as a function of intake manifold pressure (absolute). An intake pressure of about 0.45 bar is typical of a part-load operating condition for a light-duty vehicle with a naturally-aspirated spark-ignition engine. For a turbocharged SI engine, in a similar vehicle, the engine's intake manifold pressure would be about twice that, and the residual fraction significantly lower.

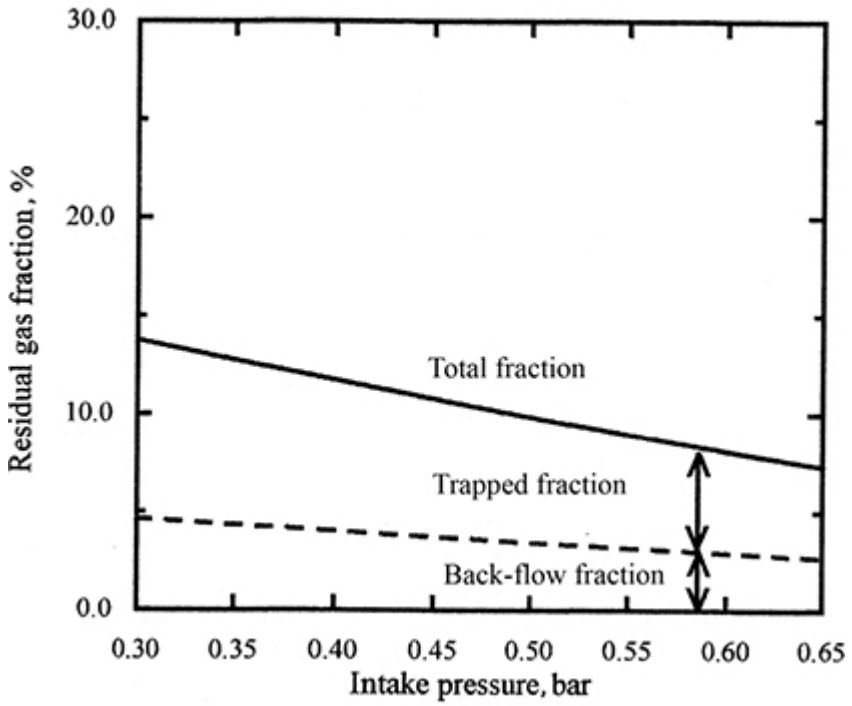


Figure 6.25 Residual gas fraction as a function of intake pressure (bar, absolute) for a naturally-aspirated gasoline engine showing the backflow fraction, the trapped fraction, and the total residual fraction.[24]

Residual gas measurements in a spark-ignition engine are given in Fig. 6.26, which shows the effect of changes in speed, valve overlap, compression ratio, and air/fuel ratio for a range of inlet manifold pressures for a 2-liter, 88.5-mm bore, four-cylinder engine. [25] The effect of variations in spark timing was negligible. Inlet pressure, speed, and valve overlap are the most important variables, though the exhaust pressure also affects the residual fraction. Normal settings for IVO (about 15° before TC) and EVC (about 12° after TC) provide sufficient overlap for good scavenging, but avoid excessive backflow from the exhaust port into the cylinder.

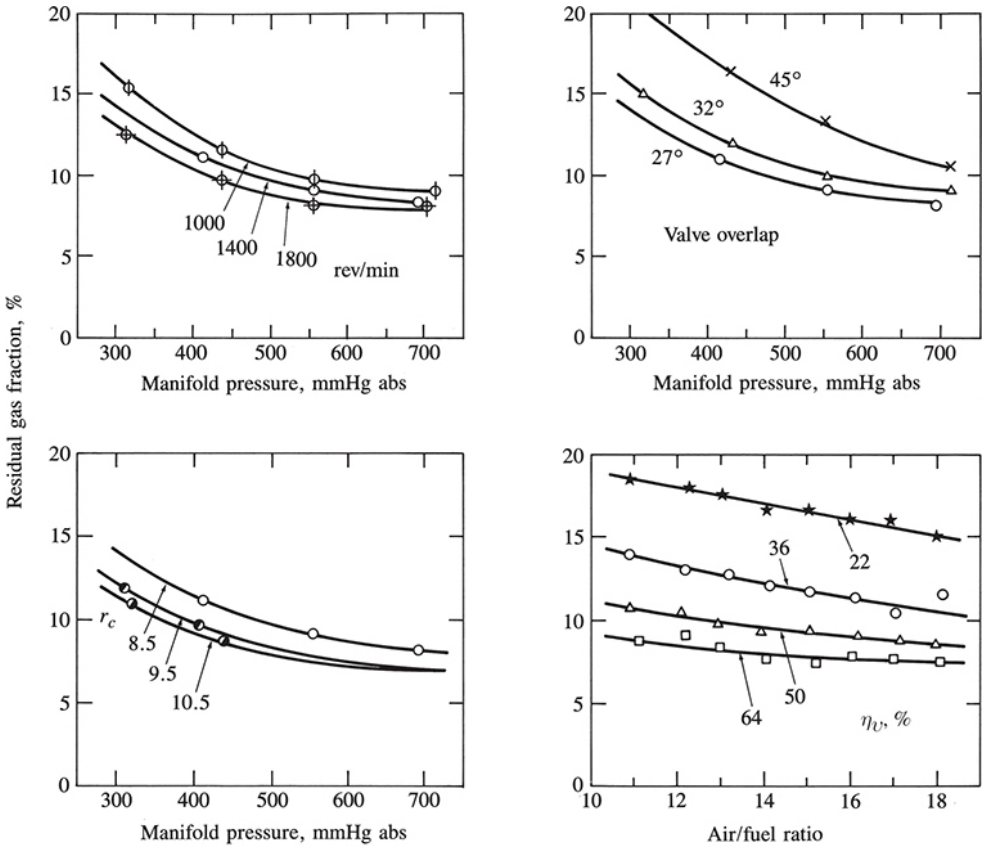


Figure 6.26 Residual gas fraction for a 2-liter four-cylinder spark-ignition engine as a function of intake manifold pressure for a range of speeds, compression ratios, and valve overlaps: also as a function of air/fuel ratios

for a range of volumetric efficiencies. Operating conditions, unless noted: speed = 1400 rev/min, A/F = 14.5, spark timing set to give 0.95 maximum torque, compression ratio = 8.5. [25]

Residual gas fractions in diesel engines are substantially lower than in SI engines because inlet and exhaust pressures are comparable in magnitude and the compression ratio is almost twice as large. Also, as consequence of the diesel's lean operation, a substantial fraction of the residual gas is air.

6.5 EXHAUST GAS FLOW RATE AND TEMPERATURE VARIATION

The exhaust gas mass flow rate and the properties of the exhaust gas vary significantly during the exhaust process. The origin of this variation for an ideal exhaust process is evident from Fig. 5.3. The thermodynamic state (pressure, temperature, etc.) of the gas in the cylinder varies continuously during the exhaust blowdown phase, until the in-cylinder pressure closely approaches the exhaust manifold pressure. In the real exhaust process, the exhaust valve restricts the flow out of the cylinder, the valve lift varies with time, and the cylinder volume changes during the blowdown process, but the principles remain the same.

Measurements have been made of the variation in mass flow rate through the exhaust valve and the gas temperature at the exhaust port exit during the exhaust process of a spark-ignition engine. [26] Figure 6.27 shows examples of the instantaneous mass flow rate data at three different engine speeds, at wide-open throttle. The *blowdown* and *displacement* phases of the exhaust process are evident. Simple quasi-steady models of these phases give good agreement with the data at lower engine speeds. The blowdown model shown applies orifice flow equations to the flow across the opening exhaust valve using the measured cylinder pressure and estimated gas temperature for upstream stagnation conditions. Equation (C.9) is used when the pressure ratio across the valve exceeds the critical value. Equation (C.8) is used when the pressure ratio is less than the critical value. The displacement model assumes the gas in the cylinder is incompressible as the piston moves through

the exhaust stroke. As engine speed increases, the crank angle duration of the blowdown phase lengthens. There is evidence of dynamic effects occurring at the transition between the two phases. The peak mass flow rate during blowdown does not vary substantially with speed since the flow is choked. The mass flow rate at the time of maximum piston speed during the displacement phase scales approximately with piston speed. As the inlet manifold pressure is reduced below the wide-open-throttle value, the proportion of the charge which exits the cylinder during the blowdown phase decreases but the mass flow rate during the displacement phase remains essentially constant (at a given engine speed). As speed increases, the displacement-phase exhaust flow rate increases in proportion to the piston speed, since the piston motion drives this phase of the exhaust process.

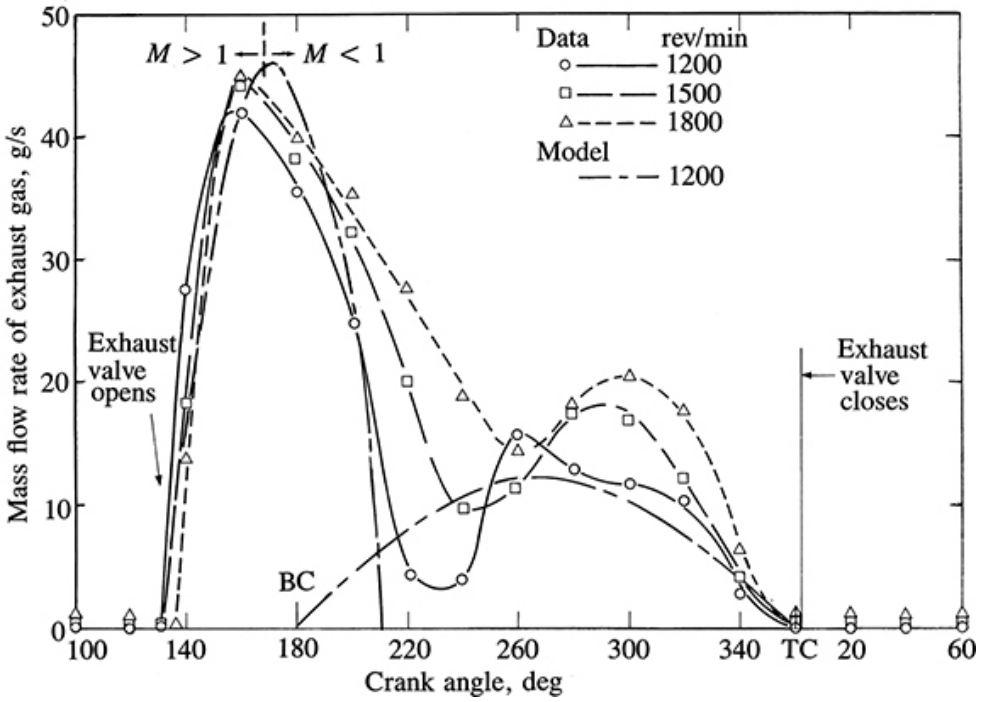


Figure 6.27 Instantaneous mass flow rate of exhaust gas through the valve versus crank angle: equivalence ratio = 1.2, wide-open throttle, compression ratio = 7. Dash-dot line is one-dimensional compressible flow model for blowdown and incompressible displacement model for the exhaust stroke. [26]

The exhaust gas temperature varies substantially through the exhaust process: it decreases due to gas expansion and to heat loss as the gas flows past the exhaust valve and through the exhaust system.

Figure 6.28 shows the measured cylinder pressure, calculated cylinder gas temperature and exhaust mass flow rate, and measured gas temperature at the exhaust port exit for a single-cylinder spark-ignition engine at mid-load and low speed.[27] The average cylinder-gas temperature falls rapidly during blowdown, and continues to fall during the exhaust stroke due to heat transfer to the cylinder walls. The gas temperature at the port exit at the start of the exhaust flow pulse is a mixture of hotter gas which has just left the cylinder and cooler gas which left the cylinder at the end of the previous exhaust process and has been stationary in the exhaust port while the valve has been closed. The port exit temperature has a maximum about where the transition from blowdown flow to displacement flow occurs, as the gas comes momentarily to rest and loses a fraction of its thermal energy to the exhaust port walls.

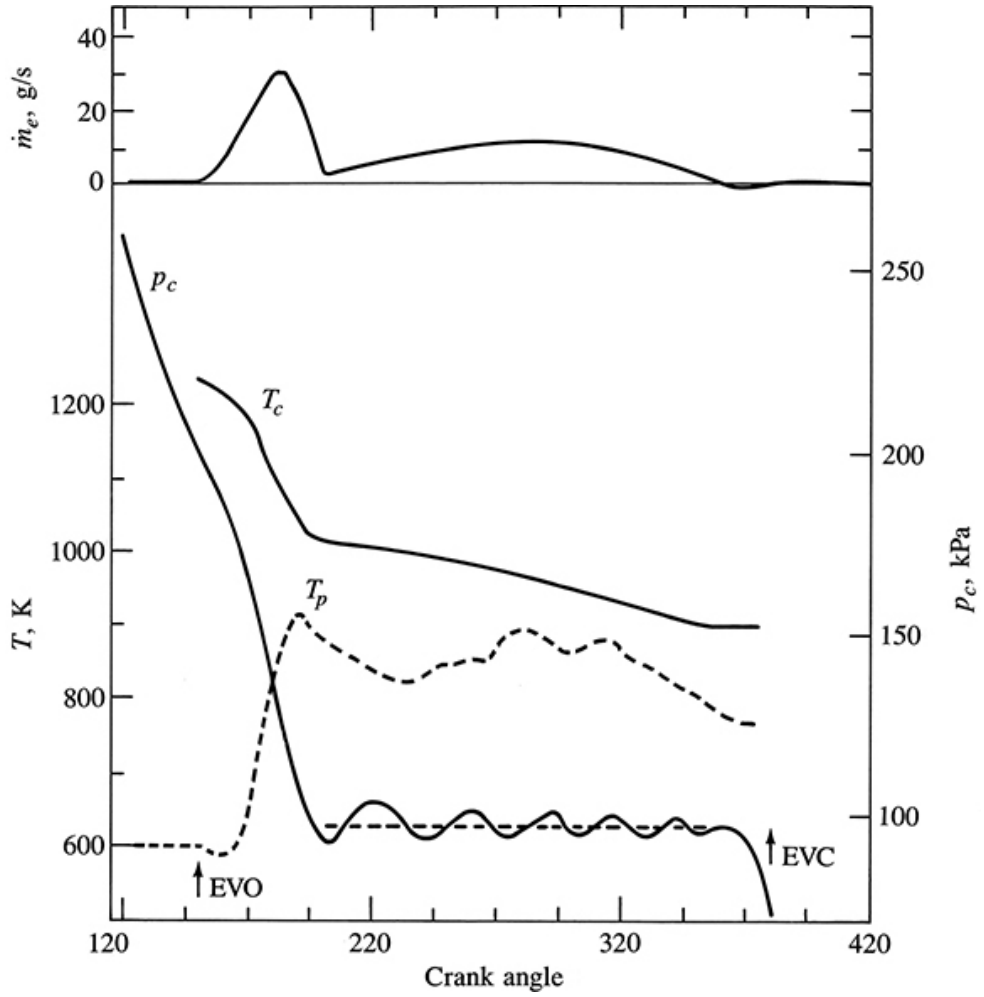


Figure 6.28 Measured cylinder pressure p_c, calculated cylinder-gas temperature T_c, exhaust mass flow rate $\dot{m}_e$, and measured gas temperature at exhaust port exit T_p, for single-cylinder spark-ignition engine. Speed = 1000 rev/min, imep = 414 kPa, equivalence ratio = 1.2, spark timing = 10° BTCq, $r_c = 7.2$. [27]

Variations in engine load and speed impact exhaust gas temperatures. Increasing load increases the mass and temperature in the blowdown pulse. Increasing speed raises the gas temperature throughout the exhaust process. These effects are the result of variations in the relative importance of heat transfer in the cylinder during the expansion stroke, and heat transfer to the exhaust valve and port as gas is exhausted. The time available for heat transfer, which depends on engine speed and exhaust gas flow rate, is the most critical factor. The exhaust temperature variation with equivalence ratio follows from the variation in expansion stroke temperatures, with maximum values at $\phi = 1.0$ and lower values for leaner and richer mixtures. Diesel engine exhaust temperatures are significantly lower than spark-ignition engine exhaust temperatures because of their lean overall equivalence ratio (which increases from very low at idle to about 0.7 at high loads) and their higher expansion ratio during the power stroke.

The average exhaust gas temperature is an important quantity for determining the performance of turbochargers, catalytic converters, and particulate traps. The time-averaged exhaust temperature does not correspond to the average energy of the exhaust gas because the flow rate varies substantially. An enthalpy-averaged temperature

$$\bar{T}_h = \left(\int_{\text{EVO}}^{\text{EVC}} \dot{m} c_p T_g \, d\theta \right) \Big/ \left(\int_{\text{EVO}}^{\text{EVC}} \dot{m} c_p \, d\theta \right) \tag{6.22}$$

is the best indicator of exhaust thermal energy. Exhaust gas temperatures are usually measured with a thermocouple. Thermocouple-averaged temperatures are close to time-averaged temperatures. Mass-averaged exhaust temperatures (which are close to $\bar{T}_h$ if c_p variations are small) for a spark-ignition engine at the exhaust port exit are about 100 K higher than time-averaged thermocouple determined temperatures. Mass-average

temperatures in the cylinder during the exhaust process are about 200 to 300 K higher than mass-averaged port temperatures. [28]

Exhaust gas temperatures increase with increasing engine speed, load, and spark retard. Relative air/fuel ratio (or equivalence ratio) is especially important, and thus exhaust gas temperatures in gasoline SI engines are higher than exhaust gas temperatures in diesel engines (which vary significantly with load since load and equivalence ratio scale with each other). Gasoline engines with stoichiometric operation have exhaust gas temperatures of about 650°C (925 K). When operated lean, exhaust gas temperatures decrease (e.g., at $\lambda = 1.4$, by about 50 deg. C). On the rich side of stoichiometric, exhaust gas temperatures rise slightly, peak at about $\lambda = 0.95$ and then decrease. In diesels, where load is controlled by varying the fuel/air equivalence ratio, at light load exhaust gas temperatures are low (200 to 300°C), and at high load ($\phi = 0.7$) are about 450°C. Small high-speed DI diesels have somewhat lower exhaust gas temperatures than do heavy-duty truck DI engines. [29] Note exhaust gas temperature values are usually time-average measurements made close to the exhaust port exit.

6.6 SCAVENGING IN TWO-STROKE CYCLE ENGINES

6.6.1 Two-Stroke Engine Configurations

In two-stroke cycle engines, every outward (or downward) stroke of the piston is a power stroke. To achieve this behavior, the burned gases have to be replaced by fresh charge while the piston moves through its bottom center (BC) position. The period during which the products of combustion are exhausted from the cylinder and the fresh charge enters, the gas exchange period or scavenging process, begins at the time that the exhaust port is first exposed (see Fig. 1.3) or the exhaust valve opens (Fig. 1.36). It is completed at the time when both the exhaust ports (or valves) and scavenge (or transfer) ports are closed. It is useful and convenient to divide the total gas exchange process into two periods: the *exhaust blowdown period*, from exhaust opening to scavenge port opening; and the *scavenging period*, which

is the remainder. The fresh charge must be supplied to the engine cylinder at a high-enough pressure to displace the in-cylinder burned gases from the previous cycle. Raising the pressure of the intake mixture is done with a pump or blower or compressor. Air capacity is just as important as in the four-stroke cycle. However, in the two-stroke cycle scavenging process, a fraction of the air is inevitably lost—it passes straight through the cylinder. Thus a greater air mass flow rate must be achieved to obtain the same output power. This loss of air during scavenging makes tracking the gas exchange process more complicated.

The different categories of two-stroke cycle scavenging flows and the port (and valve) arrangements that produce them are illustrated in Figs. 6.29 and 6.30. Scavenging arrangements are broadly classified into: (*a*) *cross-scavenged*, (*b*) *loop-scavenged*, and (*c*) *uniflow-scavenged configurations*. The location and orientation of the scavenging ports control the scavenging process, and the most common arrangements are indicated. Cross- and loop-scavenging systems use exhaust and ports in the cylinder wall, uncovered by the piston as it approaches BC. The uniflow system may use transfer ports with exhaust valves in the cylinder head, or transfer and exhaust ports with opposed pistons. Despite the different flow patterns obtained with each cylinder geometry, the general operating principles are similar. Air in a diesel, or fuel-air mixture in a spark-ignition engine, must be supplied to the transfer ports at a pressure higher than the exhaust system pressure.

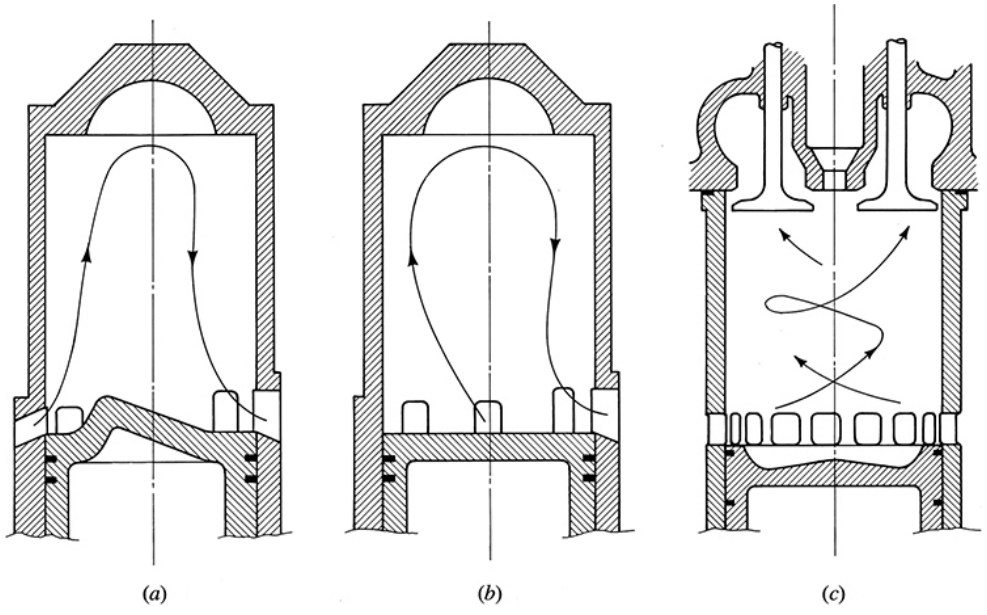


Figure 6.29 (*a*) Cross-scavenged, (*b*) loop-scavenged, and (*c*) uniflow-scavenged two-stroke cycle flow configurations.

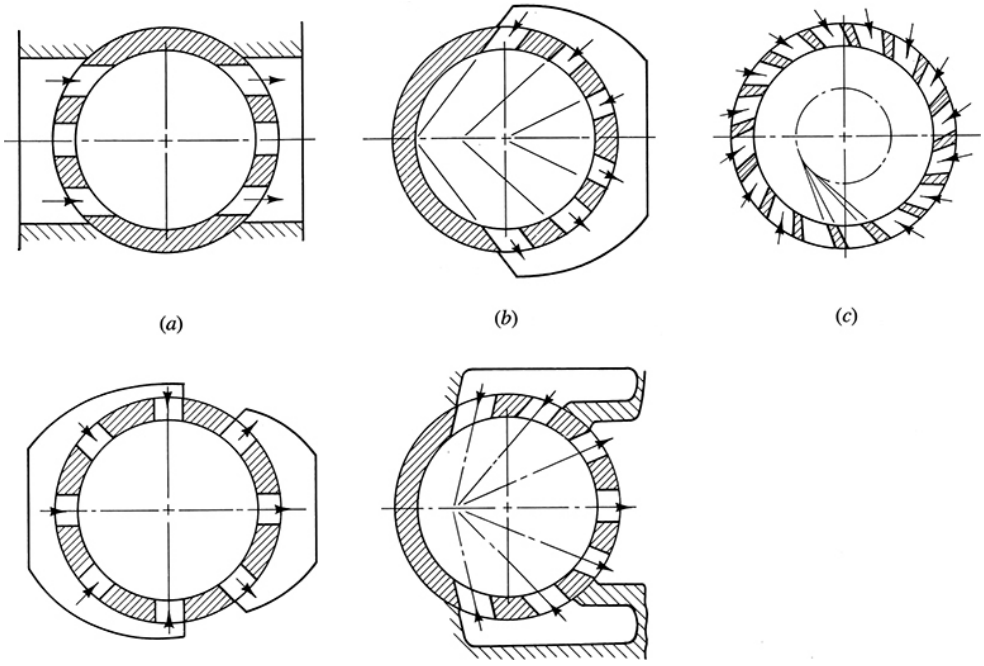


Figure 6.30 Common porting arrangements that go with (*a*) cross-scavenged, (*b*) loop-scavenged, and (*c*) uniflow-scavenged configurations.

Figure 6.31 illustrates the principles of the scavenging process for a uniflow engine design. Between 100 and 110° after TC, the exhaust valve opens and a blowdown discharge process commences. Initially, the pressure ratio across the exhaust valve exceeds the critical value (see App. C) and the flow at the valve will be sonic: as the cylinder pressure decreases, the pressure ratio drops below the critical value. Following this blowdown (or free exhaust) period, the scavenging ports open between 60 and 40° before BC when the cylinder pressure slightly exceeds the scavenging pump pressure. Because the burned gas flow is toward the exhaust valves, which now have a large open area, the exhaust flow continues and no backflow occurs. When the cylinder pressure falls below the transfer system supply pressure, air enters the cylinder and the scavenging process starts. This fresh charge flow continues as long as the transfer ports are open and the total supply pressure exceeds the pressure in the cylinder. As the cylinder pressure rises above the exhaust pressure, the fresh charge flowing into the

cylinder displaces the burned gases; however, some of the fresh charge mixes with the burned gases and is expelled with them. The exhaust valves usually close after the inlet ports close. Since the flow in the cylinder is toward the exhaust valve, additional scavenging is obtained. Proper flow patterns for the fresh charge are extremely important for good scavenging and charging of the cylinder.

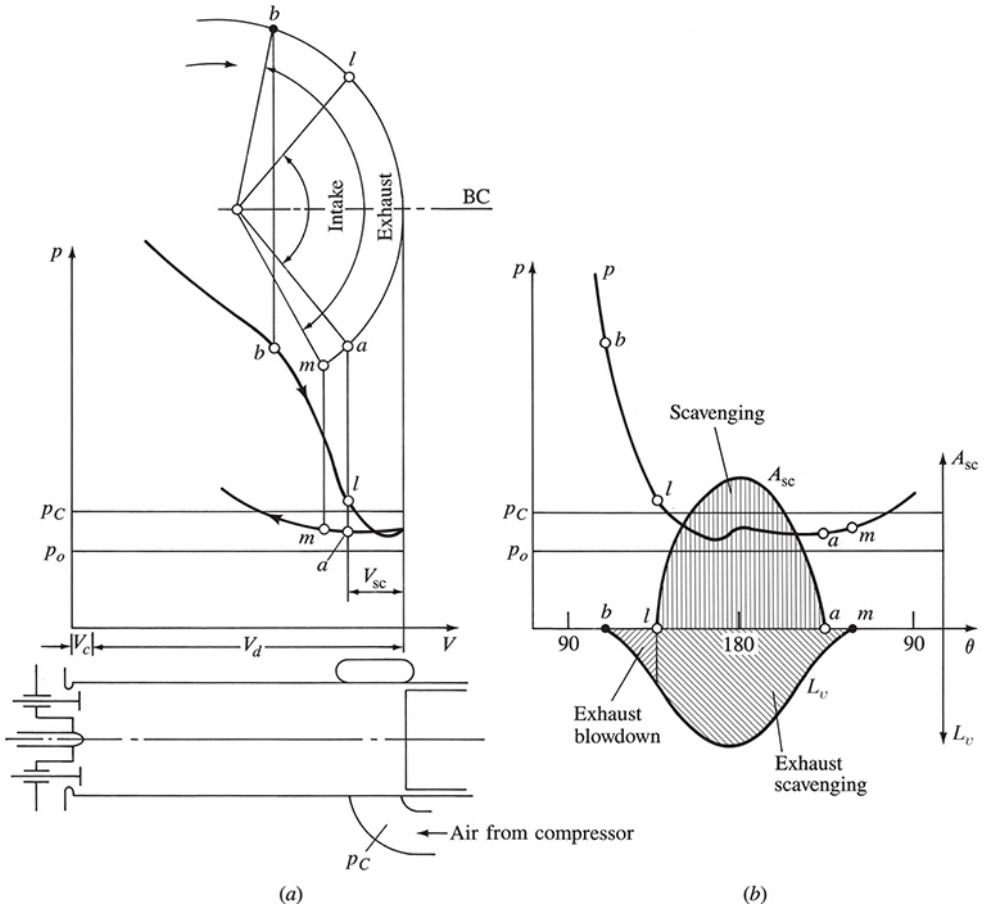


Figure 6.31 Gas exchange process in two-stroke cycle uniflow-scavenged diesel engine: (*a*) valve and port timing and pressure-volume diagram; (*b*) pressure, scavenging port open area A_{sc}, and exhaust valve lift L_U as functions of crank angle.[1]

Common methods for supercharging or pressurizing the fresh charge are shown in Fig. 6.32. In large two-stroke cycle engines, more complex combinations of these approaches are often used, as shown in Fig. 1.24. Substantial additional detail on two-stroke cycle engines and their

scavenging characteristics can be found in texts by Blair [30] and Heywood and Sher. [31]

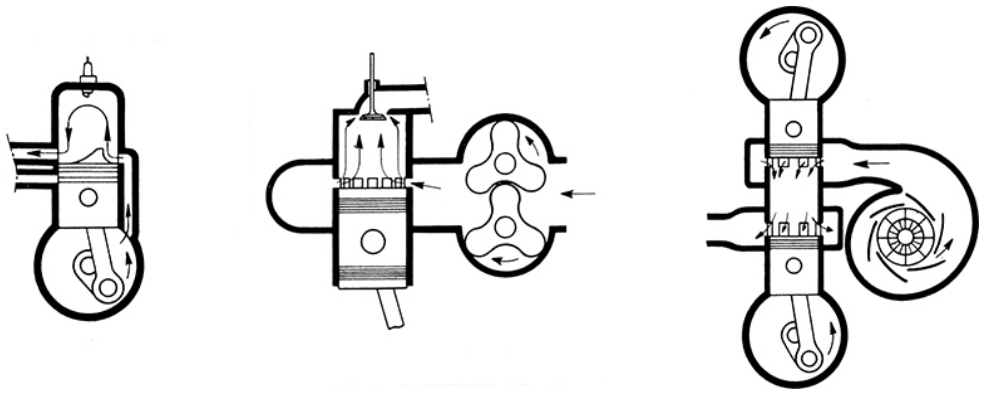


Figure 6.32 Common methods of pressurizing the fresh charge in two-stroke cycle engines: left, crankcase compression; center, roots blower; right, centrifugal compressor.[13]

6.6.2 Scavenging Parameters and Models

The following parameters are used to quantify the scavenging process:

The *delivery ratio* Λ:

$$\Lambda = \frac{\text{mass of delivered air (or mixture) per cycle}}{\text{reference mass}} \tag{6.23}$$

compares the actual scavenging air mass (or mixture mass) to that required in an ideal charging process. [g] The reference mass is defined as displaced volume × ambient air (or mixture) density. Ambient air (or mixture) density is determined at atmospheric conditions or at intake conditions. This definition is useful for experimental purposes. For analytical work, it is often convenient to use the trapped cylinder mass m_{tr} as the reference mass.

The *trapping efficiency* η_{tr}:

$$\eta_{tr} = \frac{\text{mass of delivered air (or mixture) retained}}{\text{mass of delivered air (or mixture)}} \tag{6.24}$$

indicates what fraction of the air (or mixture) supplied to the cylinder is retained in the cylinder.

The *scavenging efficiency* η_{sc}:

$$\eta_{sc} = \frac{\text{mass of delieverd air (or mixture) retained}}{\text{mass of trapped cylinder charge}} \tag{6.25}$$

indicates to what extent the residual gases in the cylinder have been replaced with fresh air.

The *purity* of the charge, β:

$$\beta = \frac{\text{mass of air trapped cylinder charge}}{\text{mass of trapped cyclinder charge}} \tag{6.26}$$

indicates the degree of dilution, with burned gases, of the unburned mixture in the cylinder.

The *charging efficiency* n_{ch}:

$$\eta_{ch} = \frac{\text{mass of delivered air (or mixture) retained}}{\text{displaced volume} \times \text{ambient density}} \tag{6.27}$$

indicates how effectively the cylinder volume has been filled with fresh air (or mixture). Charging efficiency, trapping efficiency, and delivery ratio are related by

$$\eta_{ch} = \Lambda \eta_{tr} \tag{6.28}$$

When the reference mass in the definition of delivery ratio is the trapped cylinder mass m_{tr} (or closely approximated by it) then

$$\eta_{sc} = \Lambda \eta_{tr} \tag{6.29}$$

In real scavenging processes as the fresh charge pushes the burned gas out of the cylinder, mixing occurs between the fresh charge and the burned gases and some of the fresh charge may is exhausted. Two limiting ideal models of this process are: (1) perfect displacement and (2) complete mixing. Perfect displacement or scavenging would occur if the burned gases were pushed out

by the fresh gases without any mixing. Complete mixing occurs if entering fresh mixture mixes instantaneously and uniformly with the cylinder contents. *h*

For *perfect displacement* (with m_{tr} as the reference mass in the delivery ratio),

$$\begin{aligned} \eta_{sc} &= \Lambda \quad \text{and} \quad \eta_{tr} = 1 \quad \text{for } \Lambda \leq 1 \\ \eta_{sc} &= 1 \quad \text{and} \quad \eta_{tr} = \Lambda^{-1} \quad \text{for } \Lambda > 1 \end{aligned} \tag{6.30}$$

For the *complete mixing* limit, consider the scavenging process as a quasi-steady flow process. Between time t and $t + dt$, a mass element dm_{ad} of air is delivered to the cylinder and is uniformly mixed throughout the cylinder volume. An equal amount of fluid, with the same proportions of air and burned gas as the cylinder contents at time t, leaves the cylinder during this time interval. Thus the mass of air delivered between t and $t + dt$ which is retained, dm_{ar}, is given by

$$dm_{ar} = dm_{ad}\left(1 - \frac{m_{ar}}{m_{tr}}\right)$$

Assuming m_{tr} is constant, this integrates over the duration of the scavenging process to give

$$\frac{m_{ar}}{m_{tr}} = 1 - \exp\left(\frac{m_{ad}}{m_{tr}}\right) \tag{6.31}$$

Thus, for *complete mixing*, with the above definitions,

$$\begin{aligned} \eta_{sc} &= 1 - e^{-\Lambda} \\ \eta_{tr} &= \frac{1}{\Lambda}(1 - e^{-\Lambda}) \end{aligned} \tag{6.32}$$

Figure 6.33 shows η_{sc} and η_{tr} for the perfect displacement and complete mixing assumptions as a function of Λ, the delivery ratio.

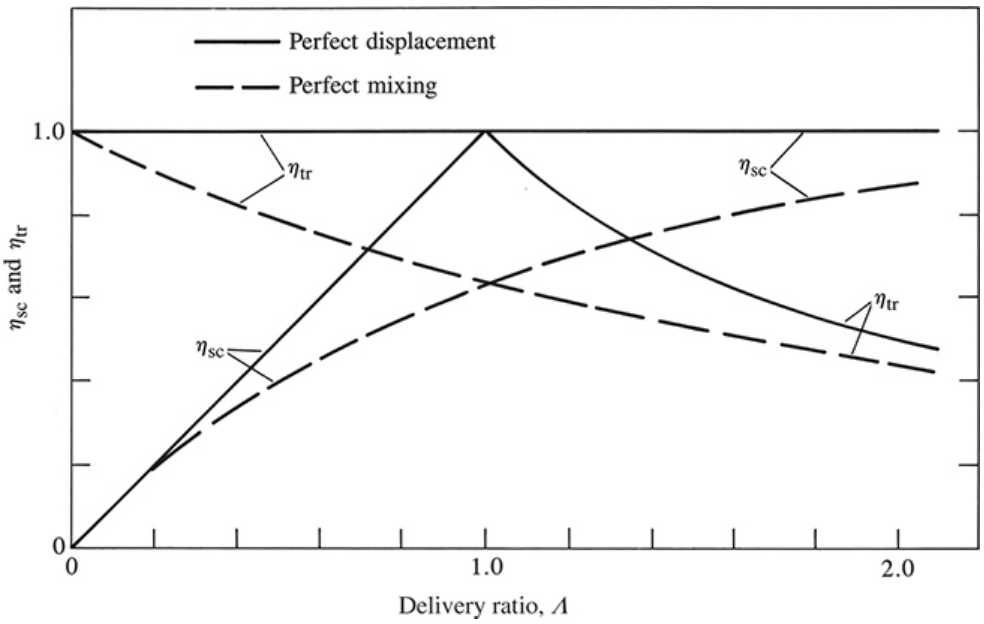


Figure 6.33 Scavenging efficiency η_{sc} and trapping efficiency η_{tr} versus delivery ratio A for (isothermal) perfect displacement and complete mixing models.

An additional possibility is the direct flow of fresh mixture through the cylinder into the exhaust without entraining burned gases. This is called *short-circuiting*; it is obviously undesirable since some fresh air or mixture is wasted. There is no simple model for this process. When short-circuiting occurs, lower scavenging efficiencies result even though the volume occupied by the short-circuiting flow through the cylinder does displace an equal volume of the burned gases. Another phenomenon which reduces scavenging efficiency is the formation of *pockets* or *dead zones* in the cylinder volume where burned gases can become trapped and escape displacement or entrainment by the fresh scavenging flow. These unscavenged zones are most likely to occur in regions of the cylinder that remain secluded from the main fresh mixture flow path.

6.6.3 Actual Scavenging Processes

Several methods have been developed for determining what occurs in actual cylinder scavenging processes. Accurate measurement of scavenging efficiency is difficult due to the problem of measuring the trapped air mass.

Estimation of η_{sc} from indicated mean effective pressure, and from gas sampling, are the most reliable methods. [13] Flow visualization experiments in liquid analogs of the cylinder and flow velocity mapping techniques have proved useful in providing a qualitative picture of the scavenging flow field and identifying problems such as short-circuiting and dead volumes.

Flow visualization studies indicate the key features of the scavenging process. Figure 6.34 shows a sequence of frames from a movie of one liquid scavenging another in a model of a large two-stroke cycle loop-scavenged diesel. The physical variables were scaled to maintain the same values of the appropriate dimensionless numbers for the liquid analog flow and the real engine flow. The density of the liquid representing air (which is dark) was twice the density of the liquid representing burned gas (which is clear). Early in the scavenging process, the fresh air jets penetrate into the burned gas and displace it first toward the cylinder head and then toward the exhaust ports (the schematic gives the location of the ports). During this initial phase, the outflowing gas contains no air; pure displacement of the burned gas from the cylinder is being achieved. Then short-circuiting losses start to occur, due to the damming-up or buildup of fresh air on the cylinder wall opposite the exhaust ports. The short-circuiting fluid flows directly between the scavenge ports and the exhaust ports above them. Since this damming-up of the inflowing fresh air back toward the exhaust ports continues, short-circuiting losses will also continue. While the scavenging front remains distinct as it traverses the cylinder, its turbulent character indicates that mixing between burned gas and air across the front is taking place. For both these reasons (short-circuiting and short-range mixing), the outflowing gas, once the displacement phase is over, contains an increasing amount of fresh air.

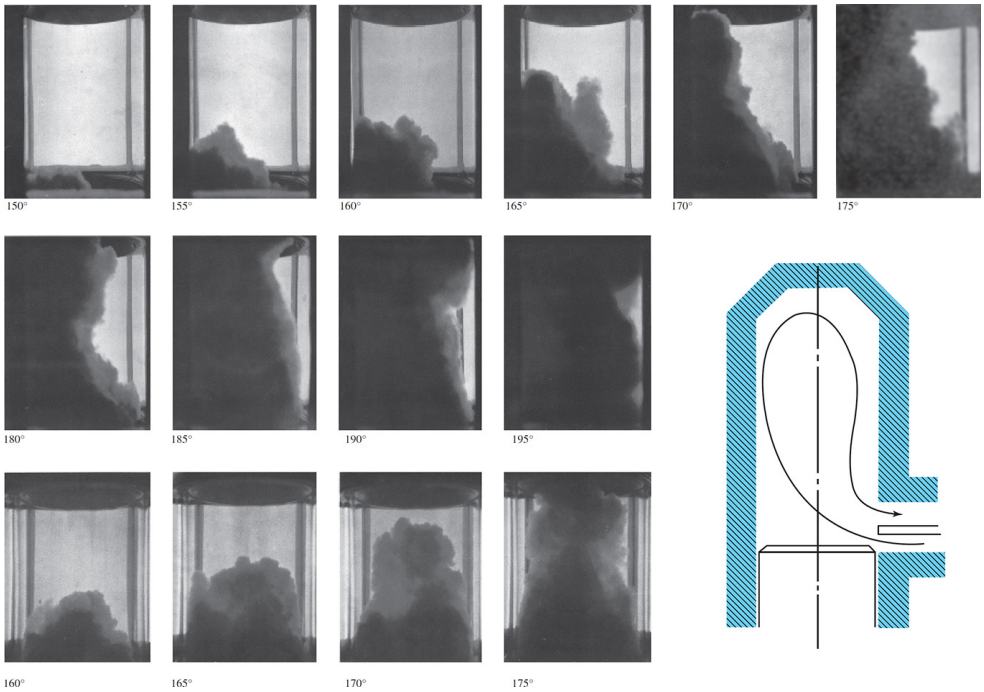


Figure 6.34 Photos of one fluid scavenging another in liquid analog experiments in model loop-scavenged engine cylinder. Top two rows: view perpendicular to scavenging loop. Bottom row: orthogonal views. Dark denser fluid displacing light less-dense fluid. [32]

Outflowing fluid composition measurements from this model study of a Sulzer two-stroke loop-scavenged diesel engine confirm this sequence of events. At 24 crank angle degrees after the onset of scavenging, fresh air due to short-circuiting was detected in the exhaust. At the time the displacement front reached the exhaust port (65° after the onset of scavenging), loss of fresh air due to scavenging amounted to 13% of the scavenge airflow. The actual plot of degree of purity (or η_{sc}) versus delivery ratio ($\mathcal{A}$) closely followed the perfect displacement line for $\mathcal{A} < 0.4$. For $\mathcal{A} > 0.4$, the shape of the actual curve was similar in shape to the complete mixing curve.

Engine tests confirm these results from model studies. Initially, the exhausted gas contains no fresh air or mixture; only burned gas is being displaced from the cylinder. However, within the cylinder both displacement *and* mixing at the interface between burned gas and fresh gas are occurring. The departure from perfect scavenging behavior is evident when fresh mixture first appears in the exhaust. For loop-scavenged engines this is typically when $\mathcal{A} \approx 0.4$. For uniflow scavenging this perfect scavenging

phase lasts somewhat longer; for cross-scavenging it is over sooner (because the short-circuiting path is shorter).

A three-zone model developed by Benson, [33] based on observations and measurements such as those described above, provides a useful framework for the two-stroke loop- and cross-scavenging processes: see Fig. 6.35. He suggested that the gas exchange process takes place in three places. In phase I, the fresh charge jets enter the cylinder and displace some of burnt in-cylinder gases. Mixing occurs at the boundaries of the jets. In phase II, some of the fresh charge entering the cylinder passes straight through and exits. In this phase only fresh charge is passing through the exhaust ports; mixing, however, continues to take place at the boundaries of the (fresh charge) jets. In phase III, a mixture of fresh charge and burnt gases leaves the cylinder; the composition of this mixture changes with time. In summary, three processes occur: (1) Fresh air displaces burned gases from the cylinder (desirable), at lower values of the delivery ratio, A; then (2) some of the fresh air (mixture) flows straight through the cylinder—short-circuiting occurs (undesirable); and throughout (3), short-range mixing occurs between the fresh air and burned gases: later in the scavenging process these mixed gases exit the cylinder (undesirable). Benson suggested these phase durations:

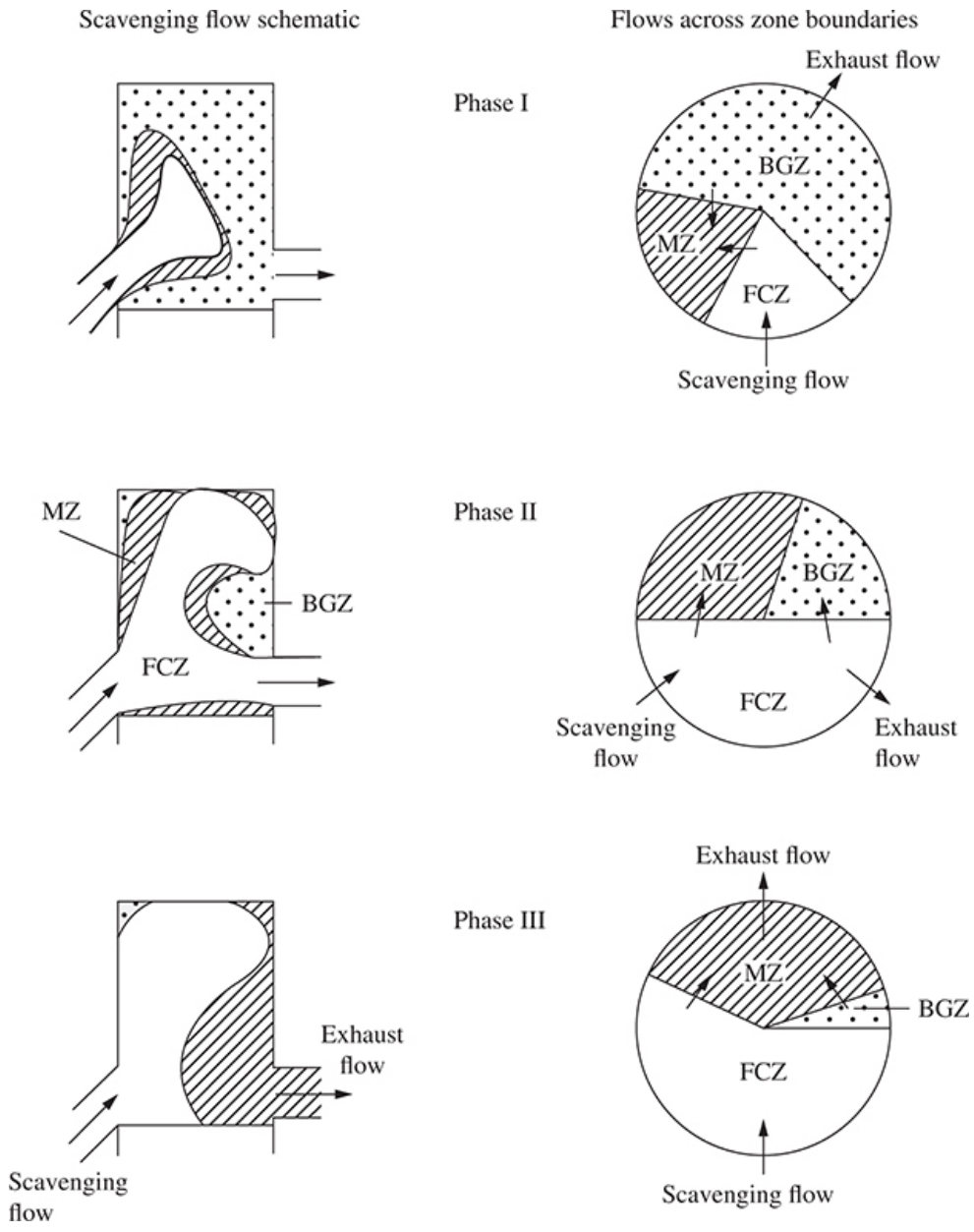


Figure 6.35 Three-zone two-stroke cycle engine scavenging model, showing early (I), mid (II), and later (III) phases of the process. BGZ, burned gas zone; FCZ, fresh charge zone; MZ, mixing zone. [33]

Phase I: $95° < \theta < 155°$ ATC

Phase II: $155° < \theta < 200°$ ATC

Phase III: $200° < \theta < 265°$ ATC

The value of this model is the conceptual framework it proposes: that the scavenging process proceeds in sequential phases where different flow behavior occurs.

The mixing that occurs is short-range mixing, not mixing throughout the cylinder volume. The jets of scavenging mixture, on entering the cylinder, mix readily with gases in the immediate neighborhood of the jet efflux. More efficient scavenging—that is, less mixing—is obtained by reducing the size of the inlet ports while increasing their number. It is important that the jets from the inlet ports slow down significantly once they enter the cylinder. Otherwise the scavenging front will reach the exhaust ports or valves too quickly. The jets are frequently directed to impinge on each other or against the cylinder wall. Swirl in uniflow-scavenged systems may be used to obtain an equivalent result.

The desirable loop-scavenging flow is illustrated in Fig. 6.36. The scavenging jets enter symmetrically with sufficient velocity to fill up about half the cylinder cross-section, and thereafter flow at lower velocity along the cylinder wall toward the cylinder head. By proper direction of the scavenging jets, it is possible to achieve almost no outflow in the direction of the exhaust from the cross-hatched stagnation zone on the opposite cylinder wall. In fact, measurement of the velocity profile in this region is a good indicator of the effectiveness of the scavenging flow. If the flow along the cylinder wall toward the head is stable, that is, if its maximum velocity occurs near the wall and the velocity is near zero on the plane perpendicular to the axis of symmetry of the ports (which passes through the cylinder axis), the scavenging flow will follow the desired path. If there are “tongues” of scavenging flow toward the exhaust port, either in the center of the cylinder or along the walls, then significant short-circuiting will occur.

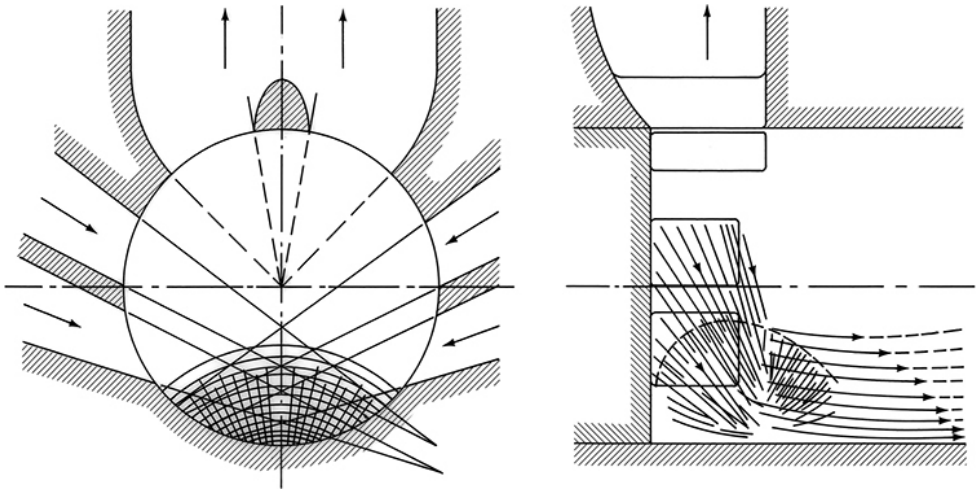
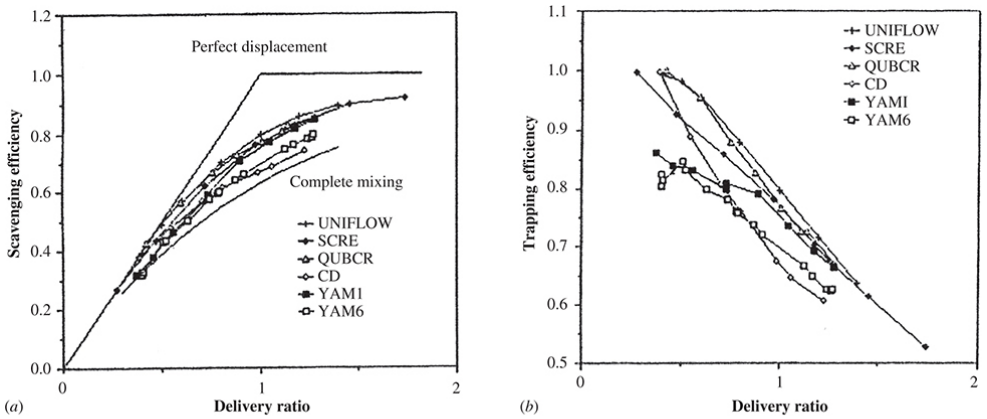


Figure 6.36 Desirable airflow in loop-scavenged engine: air from the entering jets impinges on far cylinder wall and flows toward the cylinder head.[34]

In uniflow-scavenged configurations, the inlet ports are evenly spaced around the full circumference of the cylinder and are usually directed so that the scavenging jets create a swirling flow within the cylinder (see Fig. 6.30). Results of measurements of scavenging front location in rig flow tests of a uniflow-scavenged two-stroke diesel cylinder (with an exhaust valve), as the inlet port angle was varied to give a wide range of swirl, showed that inlet jets directed tangentially to a circle of half the cylinder radius gave the most stable scavenging front profile over a wide range of conditions.

Though the scavenging processes in spark-ignition and diesel two-stroke engines are similar, these two types of engine operate with quite different delivery ratios. In mixture-scavenged spark-ignition engines, any significant expulsion of fresh fluid with the burned gas results in a significant loss of fuel and causes high hydrocarbon emissions as well as loss of the energy expended in pumping the flow which passes straight through the cylinder. In diesels the scavenging medium is air, so only the pumping work is lost. One consequence of this is that two-stroke spark-ignition engines are usually crankcase pumped. This approach provides the maximum pressure and thus also the maximum velocity in the scavenging medium at the start of the scavenging process just after the cylinder pressure has blown down; as the crankcase pressure falls during the scavenging process, the motion of the scavenging front within the cylinder also slows down. Figure 6.37 shows the

trapping and scavenging efficiencies as a function of delivery ratio, of several different designs of crankcase-scavenged spark-ignition engines. These quantities depend significantly on intake and exhaust port design and open period, and the exhaust system configuration. For two-stroke cycle spark-ignition engines, which use crankcase pumping, delivery ratios range from about 0.5 to close to 1.0.



Name	Type	Comments
UNIFLOW	Ported uniflow	Very good scavenging
SCRE	3 port loop	Good scavenging
YAM1	5 port loop	Quite good scavenging
YAM6	5 port loop	Bad loop scavenging
CD	Classic cross	Normal cross type
QUBCR	QUB cross	QUB radial ports

Figure 6.37 (*a*) Trapping efficiency η_{tr} and (*b*) scavenging efficiency η_{sc}, of several experimental scavenging system designs of two-stroke cycle spark-ignition engines, as a function of delivery ratio; these two parameters characterize scavenging effectiveness. Perfect displacement and perfect mixing model scavenging efficiencies are also shown. [30]

Figure 6.38 shows scavenging data typical of large two-stroke diesels. The purity (mass of air in trapped cylinder charge/mass of trapped cylinder charge) is shown as a function of the delivery ratio. The different scavenging configurations have different degrees of effectiveness, with uniflow scavenging being the most efficient. These diesel engines normally operate with delivery ratios in the range 1.2 to 1.4. Their large size (and slower engine speeds) results in more effective scavenging than in smaller engine loop, and especially cross-scavenged designs.

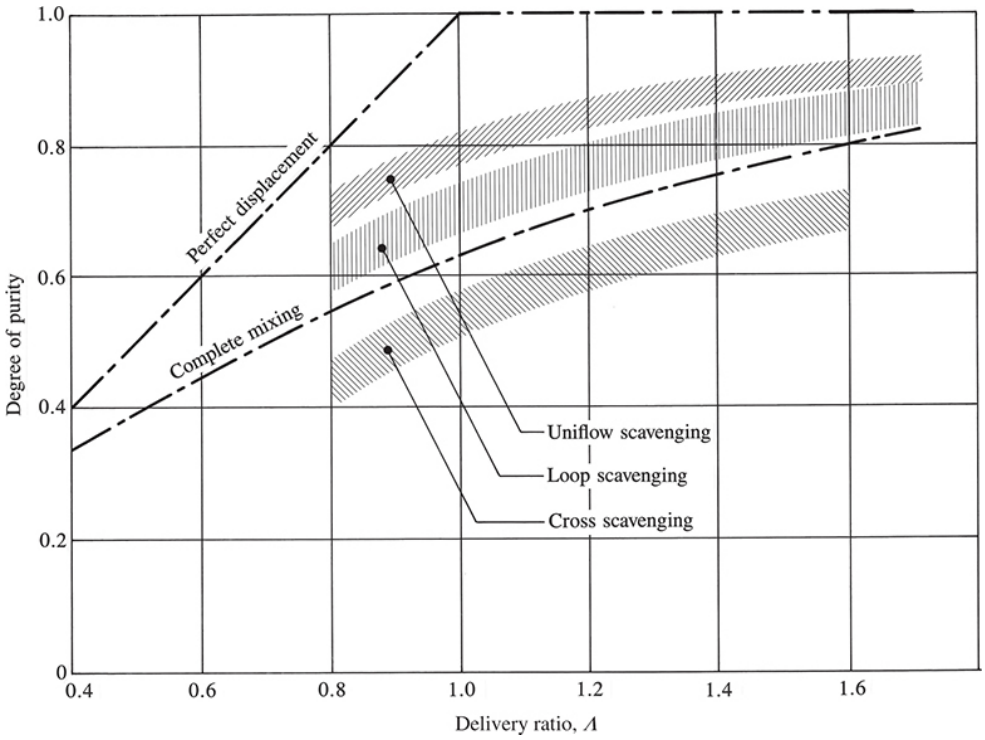


Figure 6.38 Purity as a function of delivery ratio A for different types of large marine two-stroke diesel engines.[35]

6.7 FLOW THROUGH TWO-STROKE ENGINE PORTS

The importance of the scavenging and exhaust ports to the proper functioning of the two-stroke cycle scavenging process is clear from the above discussion. The crank angle at which the ports open; the size, number, geometry, and location of the ports around the cylinder circumference; and the direction and velocity of the jets issuing from the ports into the cylinder, all affect the scavenging flow. A summary of the information available on flow through piston-controlled ports can be found in Annand and Roe. [15] Both the flow resistance of the intake (or transfer) and exhaust port configurations, as well as the details of the flow pattern produced by the port system inside the cylinder during scavenging, are important. Figure 6.39 defines the essential geometrical characteristics of such ports. Rectangular

ports make best use of the cylinder wall area and give precise timing control. Ports can be tapered, and may have axial and tangential inclination as shown.

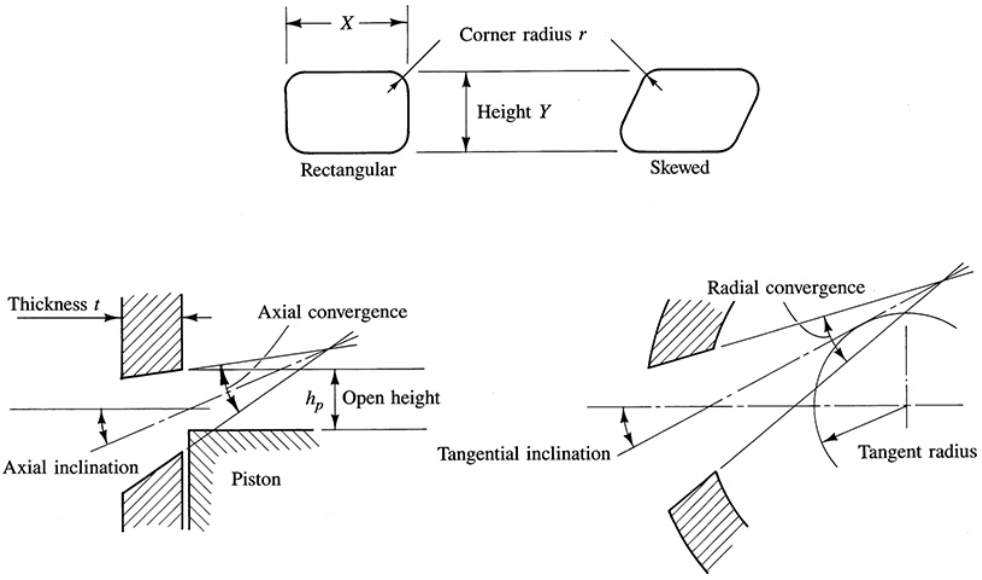


Figure 6.39 Parameters which define the geometry of scavenging or transfer ports.[15]

Scavenging Ports

Figure 6.40 illustrates the flow patterns expected downstream of piston-controlled inlet ports. For small openings, the flow remains attached to the port walls. For fully-open ports with sharp corners the flow detaches at the upstream corners. Both a rounded entry and converging taper to the port help prevent flow detachment within the port. Discharge coefficients for ports have been measured as a function of the open fraction of the port, the pressure ratio across the port, and port geometry and inclination (see Ref. 12 for a detailed summary). The most appropriate reference area for evaluating the discharge coefficient is the open area of the port (see Fig. 6.39). When the open height h_p is less than ($Y - r$) but greater than r, this is

$$A_R = Xh_p - 0.43\,r^2 \tag{6.33}$$

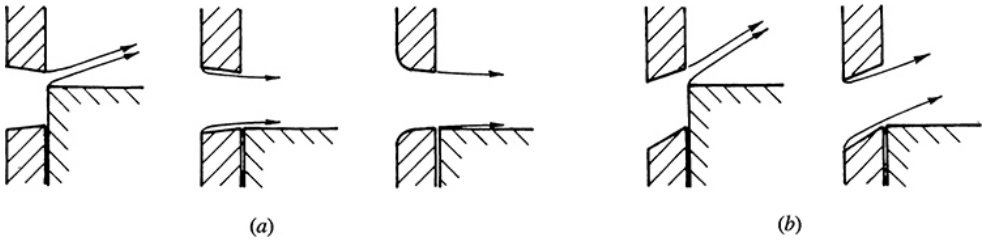


Figure 6.40 Flow pattern through piston-controlled scavenging ports. (*a*) Port axis perpendicular to wall; small opening and large opening with sharp and rounded entry. (*b*) Port axis inclined. [15]

where Y is the port height, X the port width, and r the corner radius. For $h_p = Y$, the reference area is

$$A_R = XY - 0.86\,r^2 \tag{6.34}$$

The effect of variations in geometry and operating conditions on the discharge coefficient C_D can usually be interpreted by reference to the flow patterns illustrated in Fig. 6.40. The effects of inlet port open fraction and port geometry on C_D are shown in Fig. 6.41: geometry effects are most significant at small and large open fractions. C_D varies with pressure ratio, increasing as the pressure ratio increases. Empirical relations that predict this variation with pressure ratio have been developed. [36]

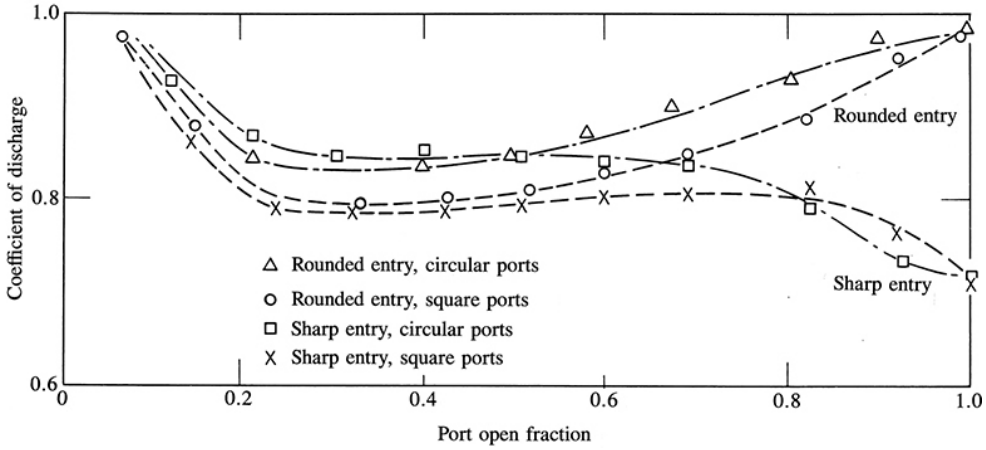


Figure 6.41 Steady-state discharge coefficients as a function of port open fraction (uncovered height/port height) for different scavenge port designs.

Pressure ratio across port = 2.35. [18]

Tangentially inclined scavenge ports are used when an in-cylinder swirling motion is desired to improve scavenging or when jet focusing or impingement within the cylinder off the cylinder axis is required (see Sec. 6.6.3). The discharge coefficient decreases as the jet tangential inclination increases. The jet angle and the port angle can deviate significantly from each other depending on the details of the port design and the open fraction.

Exhaust Ports

In piston-controlled exhaust ports, the angle of the jet from a thin-walled exhaust port increases as indicated in Fig. 6.42. With thick ports, the walls are usually tapered to allow the outward flow to diffuse. The pressure ratio across the exhaust ports varies substantially during the exhaust process. The pressure ratio has a significant effect on the exhaust port discharge coefficient, as shown in Fig. 6.43. The changes in exit jet angle and separation point explain the effects of increasing open fraction and pressure ratio. The discharge coefficient also increases modestly with increasing gas temperature.

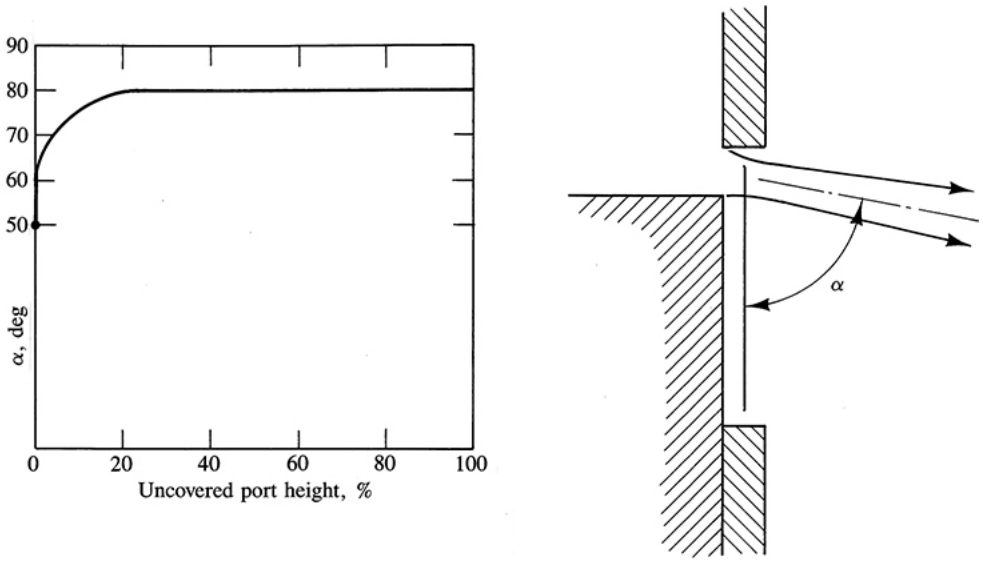


Figure 6.42 Angle of jet exiting the exhaust port as a function of open port height.[34]

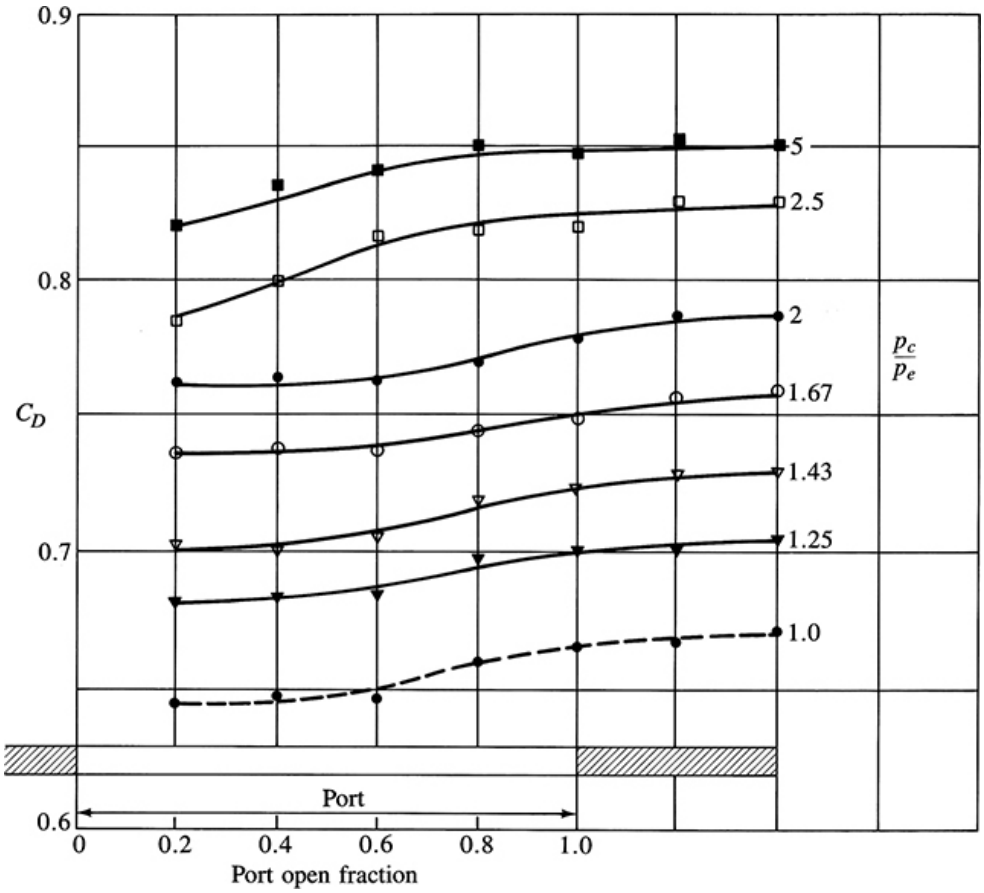


Figure 6.43 Discharge coefficient of a single rectangular exhaust port (7.6-mm wide × 12.7-mm high) in the wall of a 51-mm bore cylinder as a function of open fraction and pressure ratio. Steady-flow rig tests at 21°C. p_c = cylinder pressure, p_e = exhaust system pressure. [36, 37]

Piston Deflectors

With simple cross-scavenged two-stroke (gasoline) engines, a deflector on the piston crown is often used to force the scavenging flow toward the cylinder head and away from the exhaust port. Figure 1.3 shows a typical deflector shape. This type of piston, with cross-scavenging, has good scavenging characteristics at low power when the engine is throttled, and at low engine speed. At higher throttle openings the scavenging efficiency is not especially good. Also, the impact of the deflector on combustion chamber shape, and the engine's knock and surface ignition limits is detrimental. Usually cross-scavenged engines use a lower compression ratio than loop-scavenged engines. However, the cross-scavenged port design simplifies the

block casting and machining significantly, providing manufacturing and packaging advantages. Also, narrow ports can be used, which permits the piston rings to operate without pinning, which evens out and reduces ring wear.

Figure 6.44 shows the major geometric features of a piston deflector. Blair [30] recommends the following geometric proportions. The deflector open or flow area A_d, the area between the liner and vertical deflector surface, should be slightly larger than the total scavenging port open area [given by $n_p A_p$, where A_p (= A_R) is given by Eq. (6.34)]. Thus the entering flow will be turned smoothly through 90° and flow axially up the cylinder. A deflection ratio R_d can be defined:

$$R_d = A_d/(n_p A_p) \tag{6.35}$$

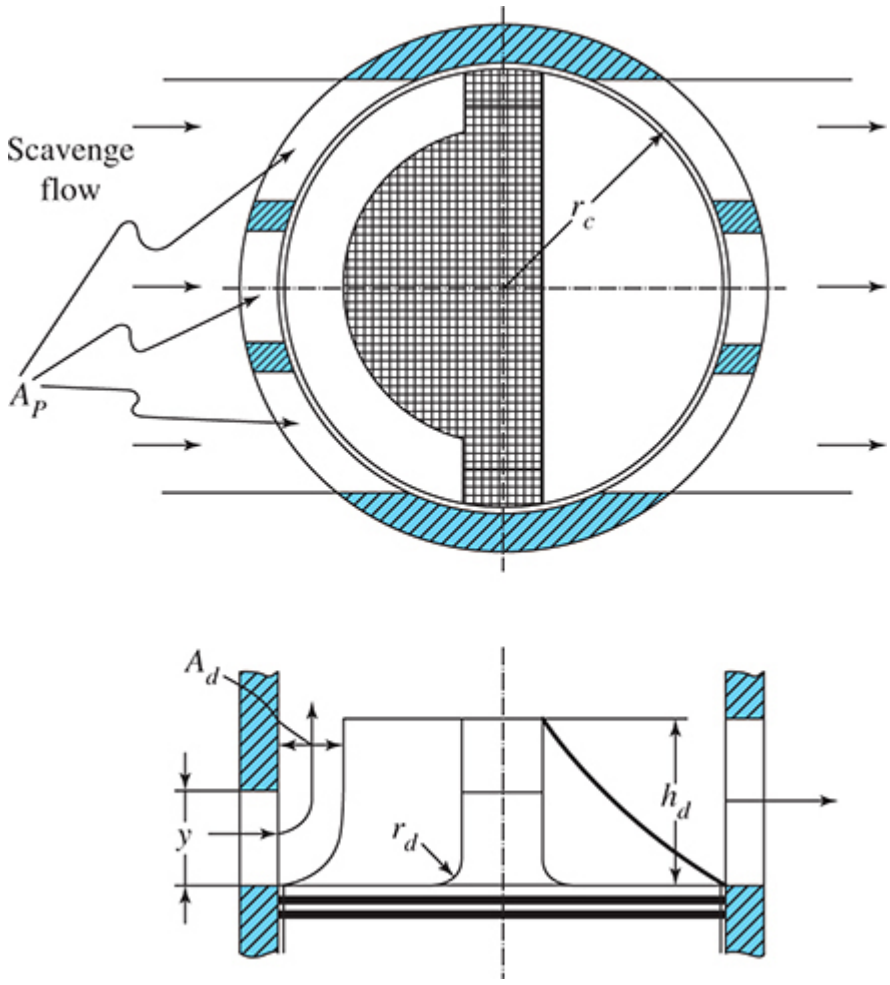


Figure 6.44 Geometry of scavenge flow and exhaust flow ports, and piston deflector.[30, 31]

where A_d is defined in Fig. 6.44. R_d should be between 1.1 and 1.2. The deflector height is the other critical parameter. It should be higher than the port by about 50%, but no higher than necessary. The deflector wall is placed about midway between the exhaust and scavenge ports. Its sides are usually sloped at 45°, or curved, so it will fit the combustion chamber at TC with adequate clearance. The radius of the deflector as it joins the piston is usually about one-third of the height. Increasing the deflection ration, *Rd* in Eq. (6.35) , steadily decreases the trapping efficiency as the delivery ratio increases. Smaller deflector flow areas ($R_d = 1.1 – 1.2$) deflect the flow better, reduce short-circuiting, and hence improve charge trapping.

6.8 SUPERCHARGING AND TURBOCHARGING

6.8.1 Methods of Power Boosting

The maximum power a given engine can deliver is limited by the amount of fuel that can be burned efficiently inside the engine cylinder. This is limited by the amount of air that is inducted into each cylinder each cycle. If this air is compressed to a higher density than ambient, prior to entry into the cylinder, the maximum power an engine of fixed dimensions can deliver will be increased. This is the primary purpose of supercharging; Eqs. (2.39) to (2.41) show how power, torque, and mean effective pressure are proportional to inlet air density. The term *supercharging* refers to increasing the air (or mixture) density by increasing its pressure prior to entering the engine cylinder. Three basic methods are used to accomplish this. The first is *mechanical supercharging* where a separate pump or blower or compressor, usually driven by power taken from the engine, provides the compressed air. The second method is *turbocharging*, where a turbocharger—a compressor and turbine on a single shaft—is used to boost the inlet air (or mixture) density. Energy available in the engine's exhaust stream is used to drive the

turbocharger turbine which drives the turbocharger compressor which raises the inlet fluid density prior to entry to each engine cylinder. The third method — *pressure wave supercharging*—uses wave action in the intake and exhaust systems to compress the intake mixture. The use of intake and exhaust manifold tuning to increase volumetric efficiency (see Sec. 6.2.2) is one example of this method of increasing air density. An example of a pressure wave supercharging device is the Comprex, which uses the pressure available in the exhaust gas stream to compress the inlet mixture stream by direct contact between the two fluids in narrow flow channels (see Sec. 6.8.5). Figure 6.45 shows typical arrangements of the different supercharging and turbocharging systems. The most common arrangements use a mechanical supercharger (Fig. 6.45*a*) or turbocharger (Fig. 6.45*b*). Combinations of an engine-driven compressor and a turbocharger (Fig. 6.45*c*) are also used (e.g., in large marine engines; Fig. 1.36). Two-stage turbocharging (Fig. 6.45*d*) is one viable approach for providing very high boost pressures (4 to 7 bar) to obtain even higher engine brake mean effective pressures. Turbocompounding, which uses a second turbine in the exhaust directly geared to the engine drive shaft to extract additional exhaust energy, Fig. 6.45, is an alternative method of increasing engine power and efficiency. Charge cooling with a heat exchanger (often called an aftercooler or intercooler), after compression, prior to gas entry to the cylinder, is commonly used to decrease the temperature of the air and thus further increase the air or mixture density, as shown in Fig. 6.45*f*.

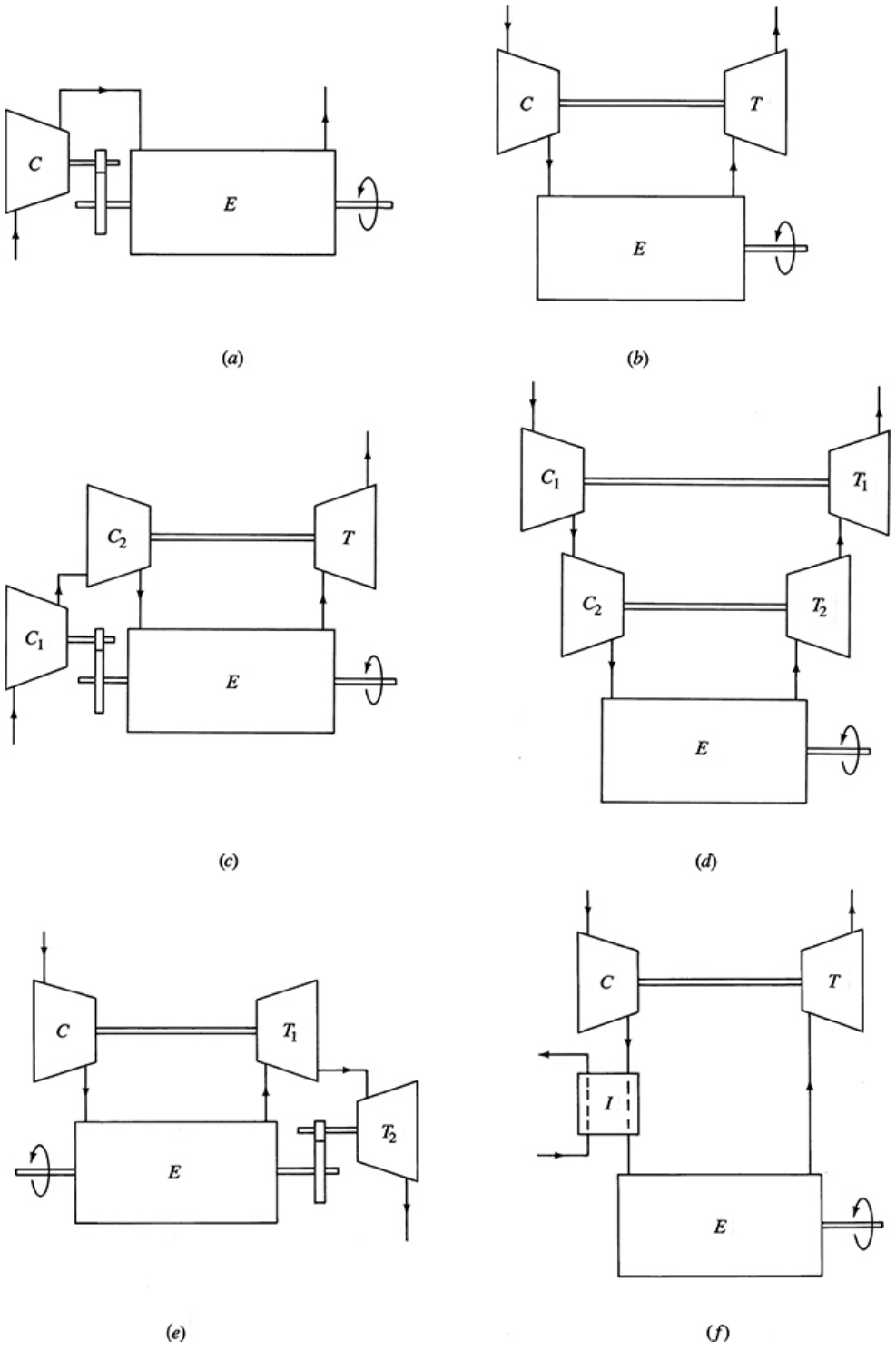


Figure 6.45 Supercharging and turbocharging configurations: (*a*) mechanical supercharger; (*b*) turbocharger; (*c*) engine-driven compressor, and turbocharger; (*d*) two-stage turbocharging; (*e*) turbocharging with

turbocompounding; (*f*) turbocharger with intercooler. *C* compressor, *E* engine, *I* inter-cooler, *T* turbine.

Supercharging is used in four-stroke cycle engines to boost the power per unit displaced volume. Some form of supercharging is necessary in two-stroke cycle engines to raise the fresh air (or mixture) pressure above the exhaust pressure so that the cylinders can be scavenged effectively. With additional boost in two-stroke cycle engines, the power density is also raised. This section reviews the operating characteristics of the blowers, compressors, turbines, and wave-compression devices used to increase inlet air or mixture density or convert exhaust-gas availability to work. The operating characteristics of supercharged and turbocharged engine systems are discussed in Chap. 15 .

Diesel engines, especially in automotive applications, are usually turbocharged. Diesels always operate fuel lean: thus, they do not fully use the air's oxygen in the cylinder for combustion. So forcing additional (i.e., compressed) air into the engine's cylinders and burning more fuel, thus increasing the engine's torque per unit volume, are especially attractive. High-performance gasoline-fueled spark-ignition engines have a history of being boosted with a mechanically driven supercharger to increase the power output of a given base engine. Turbocharged gasoline engines are replacing naturally-aspirated gasoline engines in the automotive sector to enable engine downsizing and down-speeding, and realize a significant fuel consumption benefit. The limiting abnormal combustion phenomenon of *knock* becomes more of a constraint, however, though the fuel-evaporation charge-cooling effect of the direct-injection fuel spray somewhat offsets that problem.

6.8.2 Basic Relationships

Expressions for the work required to drive a blower or compressor and the work produced by a turbine are obtained from the first and second laws of thermodynamics. The first law, in the form of the steady-flow energy equation, applied to a control volume around the turbomachinery component is

$$\dot{Q} - \dot{W} = \dot{m}\left[\left(h + \frac{C^2}{2} + gz\right)_{\text{out}} - \left(h + \frac{C_2}{2} + gz\right)_{\text{in}}\right] \tag{6.36}$$

where $\dot{Q}$ is the heat-transfer rate into the control volume, $\dot{W}$ is the shaft work-transfer rate out of the control volume, $\dot{m}$ is the mass flow rate, h is the fluid specific enthalpy, $C^2/2$ is the specific kinetic energy, and gz is the specific potential energy (which is not important and can be omitted).

A *stagnation* or *total enthalpy*, h_0, can be defined as

$$h_0 = h + \frac{C^2}{2} \tag{6.37}$$

For an ideal gas, with constant specific heats, a *stagnation* or *total temperature* follows from Eq. (6.33) :

$$T_0 = T + \frac{C^2}{2c_p} \tag{6.38}$$

A *stagnation* or *total pressure* is also defined: It is the pressure attained if the gas is isentropically brought to rest:

$$p_0 = p\left(\frac{T_0}{T}\right)^{\gamma/(\gamma-1)} \tag{6.39}$$

$\dot{Q}$ in Eq. (6.36) for pumps, blowers, compressors, and turbines is usually small enough to be neglected. Equation (6.36) then gives the work-transfer rate as

$$\dot{W} = \dot{m}(h_{0,\text{out}} - h_{0,\text{in}}) \tag{6.40}$$

A component efficiency is used to relate the actual work-transfer rate to the work-transfer rate required (or produced) by an equivalent reversible adiabatic device operating between the same pressures. The second law is then used to determine this reversible adiabatic work-transfer rate, which is that occurring in an isentropic process.

For a compressor, the *compressor isentropic efficiency* η_c is

$$\eta_c = \frac{\text{reversible power requirerment}}{\text{actual power requirement}} \tag{6.41}$$

Figure 6.46 shows the end states of the gas passing through a compressor on an h-s diagram. Both static (p_1, p_2) and stagnation (p_{01}, p_{02}) constant-pressure lines are shown. The total-to-total isentropic efficiency is, from Eq. (6.41) ,

$$\eta_{CTT} = \frac{h_{02s} - h_{01}}{h_{02} - h_{01}} \tag{6.42}$$

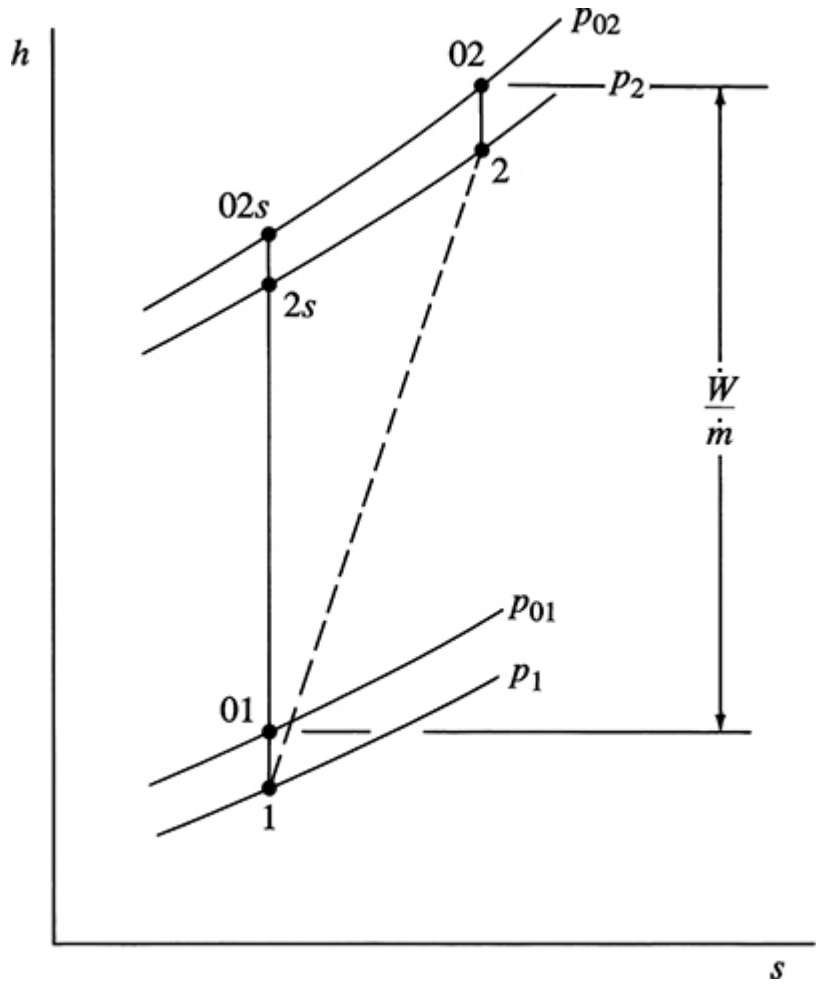


Figure 6.46 Enthalpy-entropy diagram for compressor. Inlet state 01, exit state 2; equivalent isentropic compressor exit state 2 s.

which, since c_p is essentially constant for air, or fuel-air mixture, becomes

$$\eta_{\mathrm{CTT}} = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}} \tag{6.43}$$

Since the process 01 to 02 *s* is isentropic,

$$T_{02s} = T_{01}\left(\frac{p_{02}}{p_{02}}\right)^{(\gamma-1)/\gamma}$$

Equation (6.43) becomes

$$\eta_{\mathrm{CTT}} = \frac{(p_{02}/p_{01})^{(\gamma-1)/\gamma} - 1}{(T_{02}/T_{01}) - 1} \tag{6.44}$$

In deriving Eq. (6.44) it has been tacitly assumed that the kinetic-energy pressure head $(p_{02} - p_2)$ can be recovered. In internal combustion engine applications the compressor feeds the engine via a large manifold, and much of this kinetic energy will be dissipated. The blower or compressor should be designed for effective recovery of this kinetic energy before the exit duct. Since the kinetic energy of the gas leaving the compressor is not usually recovered, a more realistic definition of efficiency is based on exit static conditions: [38]

$$\eta_{\mathrm{CTS}} = \frac{T_{2s} - T_{01}}{T_{02} - T_{01}} = \frac{(p_2/p_{01})^{(\gamma-1)/\gamma} - 1}{(T_{02}/T_{01}) - 1} \tag{6.45}$$

This is termed the total-to-static efficiency. The basis on which the efficiency is calculated should always be clearly stated.

The work-transfer rate or power required to drive the compressor is obtained by combining Eq. (6.40) , the ideal gas model, and Eq. (6.44) :

$$-\dot{W}_C = \dot{m}_i c_{p,i}(T_{02} - T_{01}) = \frac{\dot{m}_i c_{p,i} T_{01}}{\eta_{\mathrm{CTT}}}\left[\left(\frac{p_{02}}{p_{01}}\right)^{(\gamma-1)/\gamma} - 1\right] \tag{6.46}$$

where the subscript i denotes inlet mixture properties. If η_{CTS} is used to define the compressor performance, then p_2 replaces p_{02} in Eq. (6.46) . Equation (6.46) gives the thermodynamic power requirement. There will also be mechanical losses in the blower or compressor. Thus the power required to drive the device, $-\dot{W}_{C,D}$ will be

$$-\dot{W}_{C,D} = -\frac{\dot{W}_C}{\eta_m} \tag{6.47}$$

where η_m is the blower or compressor mechanical efficiency.

Figure 6.47 shows the gas states at inlet and exit to a *turbine* on an *h*-*s* diagram. State 03 is the inlet stagnation state; 4 and 04 are the exit static and stagnation states, respectively. States 4 *s* and 04 *s* define the static and stagnation exit states of the equivalent reversible adiabatic turbine. The *turbine isentropic efficiency* is defined as

$$\eta_T = \frac{\text{actual power output}}{\text{reversible power output}} \tag{6.48}$$

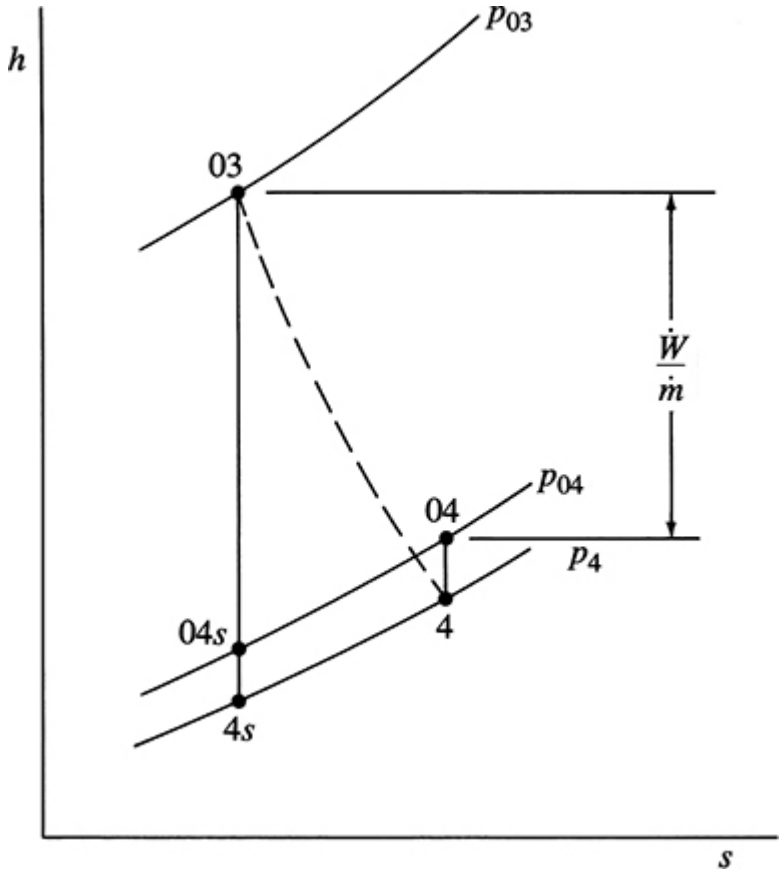


Figure 6.47 Enthalpy-entropy diagram for a turbine. Inlet state 03, exit state 4; equivalent isentropic turbine exit state 4 *s*.

Thus, the total-to-total turbine efficiency is

$$\eta_{TTT} = \frac{h_{03} - h_{04}}{h_{03} - h_{04s}} \tag{6.49}$$

If the exhaust gas is modeled as an ideal gas with constant specific heats, then Eq. (6.49) can be written

$$\eta_{TTT} = \frac{T_{03} - T_{04}}{T_{03} - T_{04s}} = \frac{1 - (T_{04}/T_{03})}{1 - (p_{04}/p_{03})^{(\gamma-1)/\gamma}} \tag{6.50}$$

Note that for exhaust gas over the temperature range of interest, c_p may vary significantly with temperature (see Figs. 4.10 and 4.17).

Since the kinetic energy at the exit of a turbocharger turbine is usually wasted, a total-to-static turbine isentropic efficiency, where the reversible adiabatic power output is that obtained between inlet stagnation conditions and the exit static pressure, is more realistic: [38]

$$\eta_{\mathrm{TTS}} = \frac{h_{03} - h_{04}}{h_{03} - h_{4s}} = \frac{T_{03} - T_{04}}{T_{03} - T_{4s}} = \frac{1 - (T_{04}/T_{03})}{1 - (p_4/p_{03})^{(\gamma-1)/\gamma}} \tag{6.51}$$

The power delivered by the turbine is given by [Eqs. (6.40) and (6.50)]:

$$\dot{W}_T = \dot{m}_e (h_{03} - h_{04}) = \dot{m}_e c_{p,e} (T_{03} - T_{04}) = \dot{m}_e c_{p,e} \eta_{\mathrm{TTT}} T_{03} \left[1 - \left(\frac{p_{04}}{p_{03}} \right)^{(\gamma_e - 1)/\gamma_e} \right] \tag{6.52}$$

where the subscript e denotes exhaust gas properties. If the total-to-static turbine efficiency (η_{TTS}) is used in the relation for $\dot{W}_T$, then p_4 replaces p_{04} in Eq. (6.52) . With a turbocharger, the turbine is mechanically linked to the compressor. Hence, at constant turbocharger speed,

$$\dot{W}_C = \eta_m \dot{W}_T \tag{6.53}$$

where η_m is the mechanical efficiency of the turbocharger. The mechanical losses are mainly bearing friction losses. The mechanical efficiency is usually combined with the turbine efficiency since these losses are difficult to separate out.

It is advantageous if the operating characteristics of blowers, compressors, and turbines can be expressed in a manner that allows easy comparison between different designs and sizes of devices. This can be done by describing the performance characteristics in terms of dimensionless numbers. [38] The most important dependent variables are: mass flow rate $\dot{m}$, component isentropic efficiency η, and temperature difference across the device ΔT_0. Each of these is a function of the independent variables: $p_{0,\mathrm{in}}$, $p_{0,\mathrm{out}}$ (or p_{out}), $T_{0,\mathrm{in}}$, N (speed), D (characteristic dimension), R (gas constant), γ (c_p/c_v), and μ(viscosity); i.e.,

$$\dot{m}, \eta, \Delta T_0 = f\left(p_{0,\text{in}}, p_{0,\text{out}}, T_{0,\text{in}}, N, D, R, \gamma, \mu\right) \tag{6.54}$$

By dimensional analysis, these eight independent variables can be reduced to four dimensionless groups:

$$\dot{m}\frac{\sqrt{RT_{0,\text{in}}}}{p_{0,\text{in}}D^2}, \eta, \frac{\Delta T_0}{T_{0,\text{in}}} = f\left(\frac{ND}{\sqrt{RT_{0,\text{in}}}}, \frac{p_{0,\text{out}}}{p_{0,\text{in}}}, \frac{\dot{m}}{\mu D}, \gamma\right) \tag{6.55}$$

The Reynolds number, $\dot{m}/(\mu D)$, has little effect on performance and γ is fixed by the gas. Therefore, these variables can be omitted and Eq. (6.51) becomes

$$\dot{m}\frac{\sqrt{RT_{0,\text{in}}}}{p_{0,\text{in}}D^2}, \eta, \frac{\Delta T_0}{T_{0,\text{in}}} = f\left(\frac{ND}{\sqrt{RT_{0,\text{in}}}}, \frac{p_{0,\text{out}}}{p_{0,\text{in}}}\right) \tag{6.56}$$

For a particular device, the dimensions are fixed and the value of R is fixed. So it has become the convention to plot

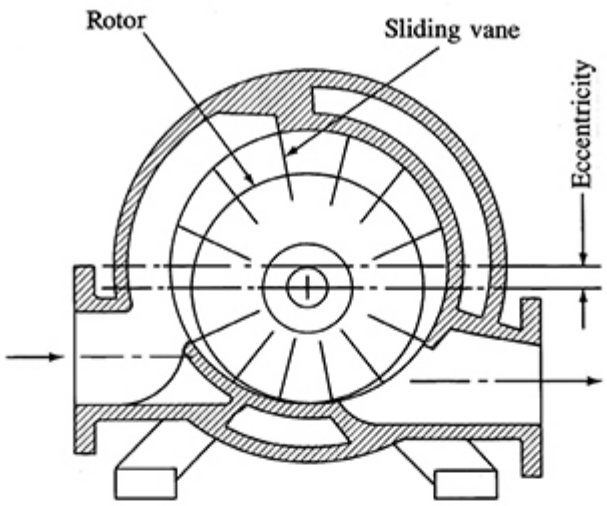
$$\dot{m}\frac{\sqrt{T_{0,\text{in}}}}{p_{0,\text{in}}}, \eta, \frac{\Delta T_0}{T_{0,\text{in}}} = f\left(\frac{N}{\sqrt{T_{0,\text{in}}}}, \frac{p_{0,\text{out}}}{p_{0,\text{in}}}\right) \tag{6.57}$$

$\dot{m}\sqrt{T_{0,\text{in}}}/p_{0,\text{in}}$ is referred to as the corrected mass flow; $N/\sqrt{T_{0,\text{in}}}$ is referred to as the corrected speed. The disadvantage of this convention of removing D and R is that the groups of variables are no longer dimensionless, and performance plots or maps relate to a specific machine.

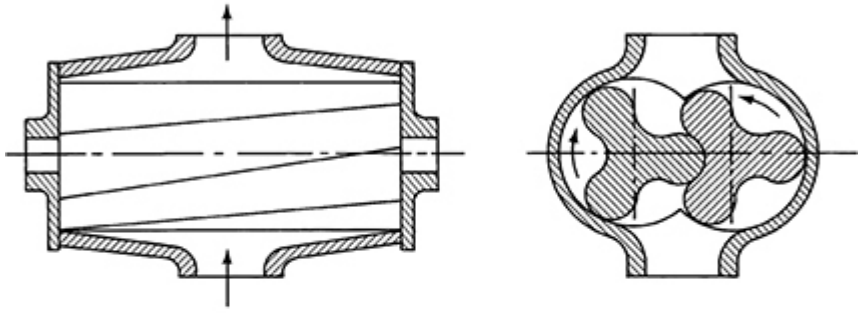
Compressor characteristics are usually plotted in terms of the pressure ratio (p_{02}/p_{01}) or (p_2/p_{01}) against the corrected mass flow $\dot{m}\sqrt{T_{0,\text{in}}}/p_{01}$) along lines of constant corrected speed ($N/\sqrt{T_{01}}$). Contours of constant efficiency are superposed. Similar plots are used for turbines: (p_{03}/p_4) against $\dot{m}\sqrt{T_{03}}/p_{03}$ along lines of constant $N/\sqrt{T_{03}}$. Since these occupy a narrow region of the turbine performance map, other plots are often used (see Sec. 6.8.4).

6.8.3 Compressors

Practical mechanical supercharging devices can be classified into: (1) sliding vane compressors; (2) rotary compressors; and (3) centrifugal compressors. The first two types are positive displacement compressors; the last type is an aerodynamic compressor. Four different examples of positive displacement compressors are illustrated in Fig. 6.48.

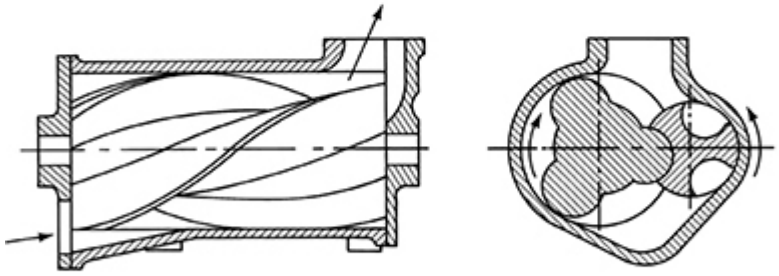


(*a*)



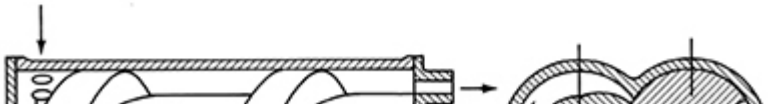
Roots blower

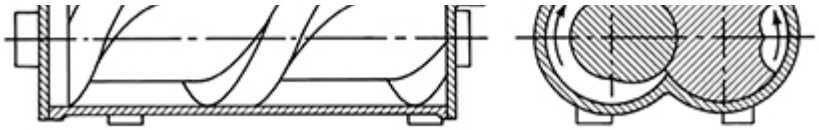
(*b*)



Lysholm compressor

(*c*)





Screw compressor

(*d*)

Figure 6.48 Positive displacement compressors: (*a*) sliding vane compressor; (*b*) roots blower; (*c*) Lysholm compressor; (*d*) screw compressor.[39]

In the *sliding vane compressor* (Fig. 6.48*a*), deep slots are cut into the rotor to accommodate thin vanes which are free to move radially. The rotor is mounted eccentrically in the housing. As the rotor rotates, the centrifugal forces acting on the vanes force them outward against the housing, thereby dividing the crescent-shaped space into several compartments. Ambient air is drawn through the intake port into each compartment as its volume increases to a maximum. The trapped air is then compressed as the compartment volume decreases, and discharged through the outlet port. The flow capacity of the sliding vane compressor depends on the maximum induction volume which is determined by the housing cylinder bore, rotor diameter and length, eccentricity, number of vanes, and dimensions of the inlet and outlet ports. The actual flow rate and pressure rise at constant speed will be reduced by leakage. Also, heat transfer from the moving vanes and rotor and stator surfaces will reduce compression efficiency unless cooling is used to remove the thermal energy generated by friction between the vanes, and the rotor and stator. The volumetric efficiency can vary between 0.6 and 0.9 depending on the size of the machine, the quality of the design, and the method of lubrication and cooling employed. The mass flow rate at constant speed depends on the pressure ratio only through its (weak) effect on volumetric efficiency. The isentropic efficiency is relatively low.

A more commonly used positive displacement supercharger is the *Roots blower* (Fig. 6.48*b*). The two rotors are connected by gears. The working principles are as follows. Air trapped in the recesses between the rotor lobes and the housing is carried toward the delivery port without significant change in volume. As these recesses open to the delivery line, since the suction side is closed, the trapped air is rapidly compressed by the backflow

from the higher-pressure delivery line. This intermittent delivery produces nonuniform torque on the rotor and pressure pulses in the delivery line. Roots blowers are most suitable for small pressure ratios (about 1.2); for example, in two-stroke cycle engines. The volumetric efficiency depends on the running clearances, rotor length, rotational speed, and pressure ratio. In the three-lobe machines shown (two lobes are sometimes used) the volume of each recess V_R is

$$V_R = 0.564R^2l$$

where R is the rotor radius and l the blower length. The mass flow parameter is

$$\frac{\dot{m}\sqrt{T_0/T_{std}}}{p_0/p_{std}} = \text{constant} \times \rho_i \eta_v N R^2 l \tag{6.58}$$

A performance map of a Roots blower is shown in Fig. 6.49. At constant speed, the flow rate depends on increasing pressure ratio only through the resulting decrease in volumetric efficiency [Eq. (6.56)].[39] One advantage of the Roots blower is that its performance range is not limited by surge and choking as is the centrifugal compressor (see below). Its disadvantages are its high noise level, poorer efficiency, and large size. [40]

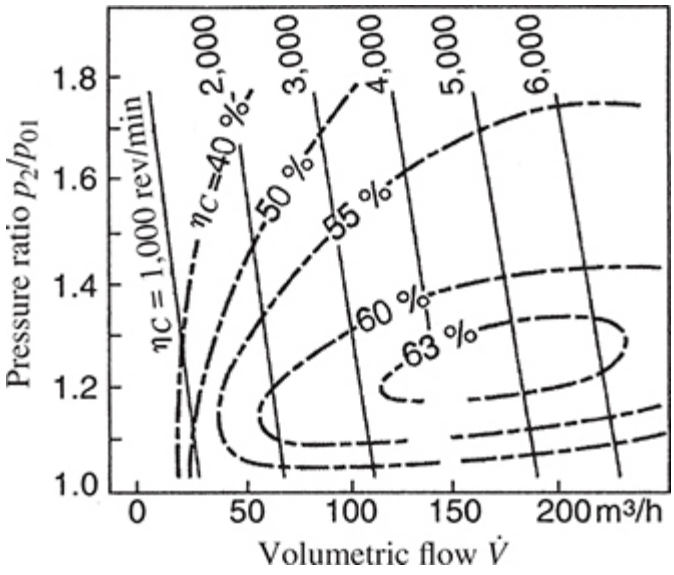


Figure 6.49 Performance map at standard inlet conditions for Roots blower: p_1, total pressure before compressor; p_2, total pressure after compressor; $_{\eta C}$, supercharger speed, rev/min; η_C, isentropic efficiency, $\dot{V}$ airflow rate, m^3/h^3. (*Courtesy Robert Bosch GmbH and SAE*.)

Screw compressors (Fig. 6.48*c* and *d*) must be precision machined to achieve close tolerances between rotating and stationary elements for satisfactory operation. They run at speeds between 3000 and 30,000 rev/min. It is usually necessary to cool the rotors internally. High values of volumetric and isentropic efficiency are claimed. [39]

The other compressors used with engines are aerodynamic devices. A *centrifugal compressor* is primarily used to boost inlet air or mixture density coupled with an exhaust-driven turbine in a turbocharger. It is a single-stage radial flow device, well suited to the high mass flow rates at the relatively low pressure ratios (up to about 3.5) required by the engine. To operate efficiently it must rotate at high angular speed. It is therefore much better suited to direct coupling with the exhaust-driven turbine of the turbocharger than to mechanical coupling through a gearbox to the engine for mechanical supercharging. It is widely used to boost diesel and gasoline engine outputs.

The centrifugal compressor consists of a stationary inlet casing, a rotating bladed impellor, a stationary diffuser (with or without vanes), and a collector or volute to bring the compressed air leaving the diffuser to the engine intake system (see Fig. 6.50). Figure 6.51 indicates, on an *h-s* diagram, how each component contributes to the overall pressure rise across the compressor. Air at stagnation state 0 is accelerated in the inlet to pressure p_1 and velocity C_1. The enthalpy change, 01 to 1, is $C_1^2/2$. Compression in the impeller flow passages increases the pressure to p_2 and velocity to C_2, corresponding to a stagnation state 02 if all the exit kinetic energy were recovered. The isentropic equivalent compression process has an exit static state 2 *s*. The diffuser, 2 to 3, converts as much as practical of the air kinetic energy at exit to the impeller ($C_2^2/2$) to a pressure rise ($p_3 - p_2$) by slowing down the gas in carefully shaped expanding passages. The final state, in the collector, has static pressure p_3, low kinetic energy $C_3^2/2$, and a stagnation pressure p_{03} which is less than p_{02} since the diffusion process is incomplete as well as irreversible.

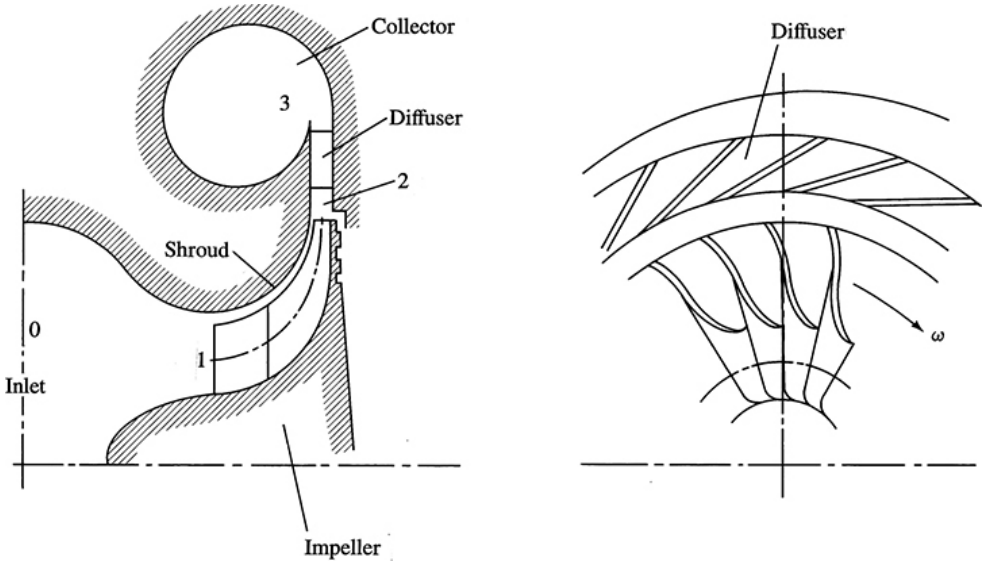


Figure 6.50 Schematic of centrifugal compressor.[38]

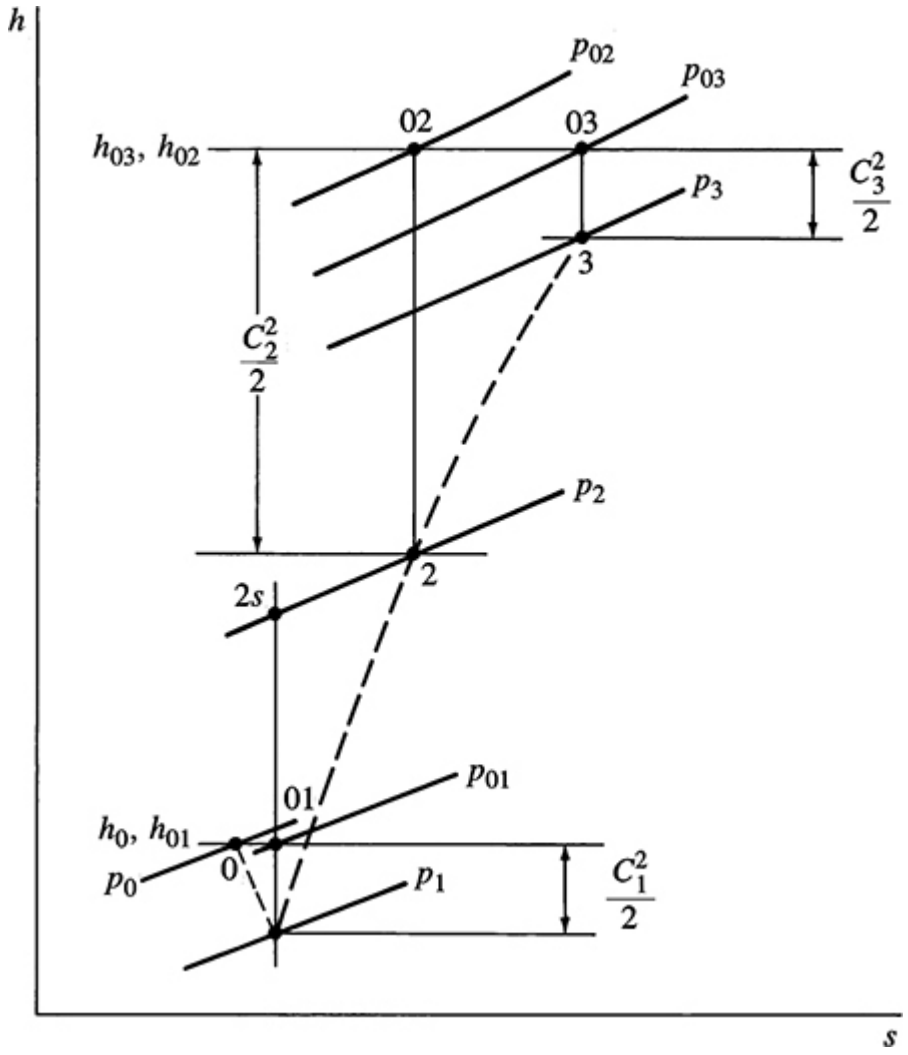


Figure 6.51 Enthalpy-entropy diagram for flow through centrifugal compressor.

The work transfer to the gas occurs in the impeller. It can be related to the change in gas angular momentum via the velocity components at the impeller entry and exit, which are shown in Fig. 6.52. Here C_1 and C_2 are the absolute gas velocities, U_1 and U_2 are the tangential blade velocities, and w_1 and w_2 are the gas velocities relative to the impeller, all at inlet (1) and exit (2), respectively. The torque T exerted on the gas by the impeller equals the rate of change of angular momentum:

$$T = \dot{m}(r_2 C_{\theta 2} - r_1 C_{\theta 1}) \tag{6.59}$$

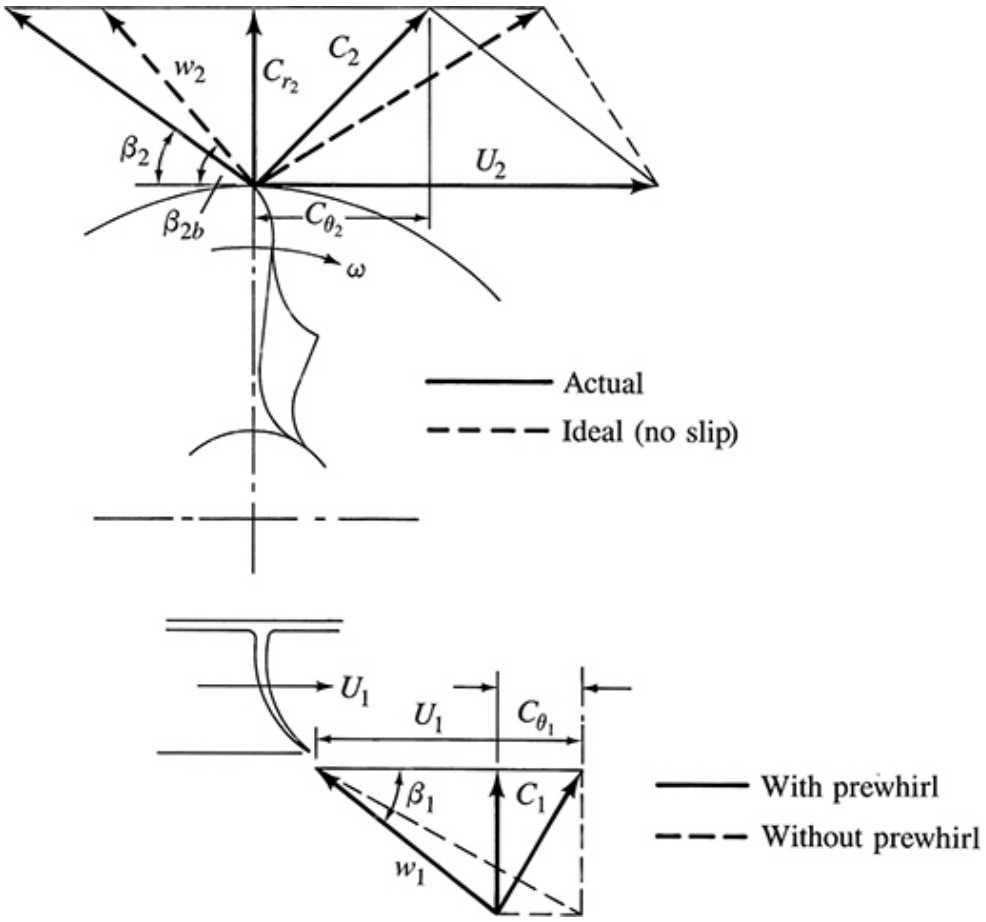


Figure 6.52 Velocity diagrams at inlet (1) and exit (2) to centrifugal compressor rotor or impeller.[38]

The rate of work transfer to the gas is given by

$$-\dot{W}_C = T\omega = \dot{m}\omega\left(r_2 C_{\theta 2} - r_1 C_{\theta 1}\right) = \dot{m}\left(U_2 C_{\theta 2} - U_1 C_{\theta 1}\right) \tag{6.60}$$

This is often called the Euler equation. Normally in compressors the inlet flow is axial so $C_{\theta 1} = 0$. Thus Eq. (6.60) can be written:

$$\frac{\dot{W}_C}{\dot{m}} = U_2 C_{\theta 2} = U_2\left(1 - \frac{C_{r2}}{U_2}\cot\beta_2\right) \tag{6.61}$$

where β_2 is the backsweep angle. In the ideal case with no slip, β_2 is the blade angle, β_{2b}. In practice, there is slip and β_2 is less than β_{2b}. Many

compressors have radial vanes (i.e., $\beta_{2b} = 90°$). A recent trend is backswept vanes ($\beta_{2b} < 90°$) which give higher efficiency. Since work transfer to the gas occurs only in the impeller, the work-transfer rate given by Eq. (6.61) equals the change in stagnation enthalpy $(h_{03} - h_{01})$ in Fig. 6.51 [see Eq. (6.40)].

The operating characteristics of a centrifugal compressor are usually described by a *performance map*. This shows lines of constant compressor efficiency η_C, and constant corrected speed $N/\sqrt{T_{0,\text{in}}}$ on a plot of pressure ratio $p_{0,\text{out}}/p_{0,\text{in}}$ against corrected mass flow $\dot{m}\sqrt{T_{0,\text{in}}}/p_{0,\text{in}}$ [see Eq. (6.57)]. Figure 6.53 indicates the form of such a map. The stable operating range in the center of the map is separated from an unstable region on the left by the *surge line*. When the mass flow is reduced at constant pressure ratio, local flow reversal eventually occurs in the boundary layer. Further reductions in mass flow cause the flow to reverse completely, causing a drop in pressure. This relieves the adverse pressure gradient. The flow reestablishes itself, builds up again, and the process repeats. Compressors should not be operated in this unstable regime. The stable operating regime is limited on the right by choking. The velocities increase as $\dot{m}$ increases, and eventually the flow becomes sonic in the limiting area of the machine. Extra mass flow through the compressor can only be obtained by higher speed. When the diffuser is choked, compressor speed may rise substantially with only a limited increase in the mass flow rate. [38]

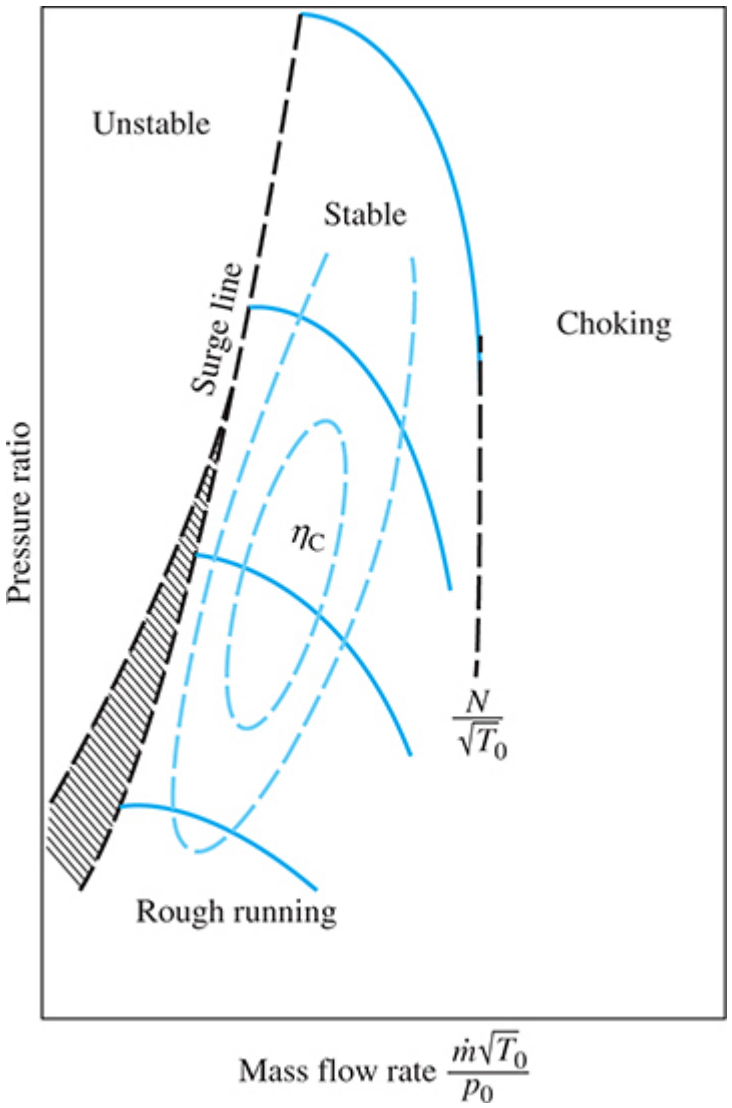


Figure 6.53 Schematic of compressor operating map showing stable operating range.[38]

Figure 6.54 shows an actual turbocharger compressor performance map. In practice, the map variables corrected speed and mass flow rate are usually defined as

$$N_{cor} = N\left(\frac{T_{ref}}{T_{0,in}}\right)^{1/2} \qquad \dot{m}_{cor} = \dot{m}\left(\frac{T_{0,in}}{T_{ref}}\right)^{1/2}\left(\frac{p_{ref}}{p_{0,in}}\right) \tag{6.62}$$

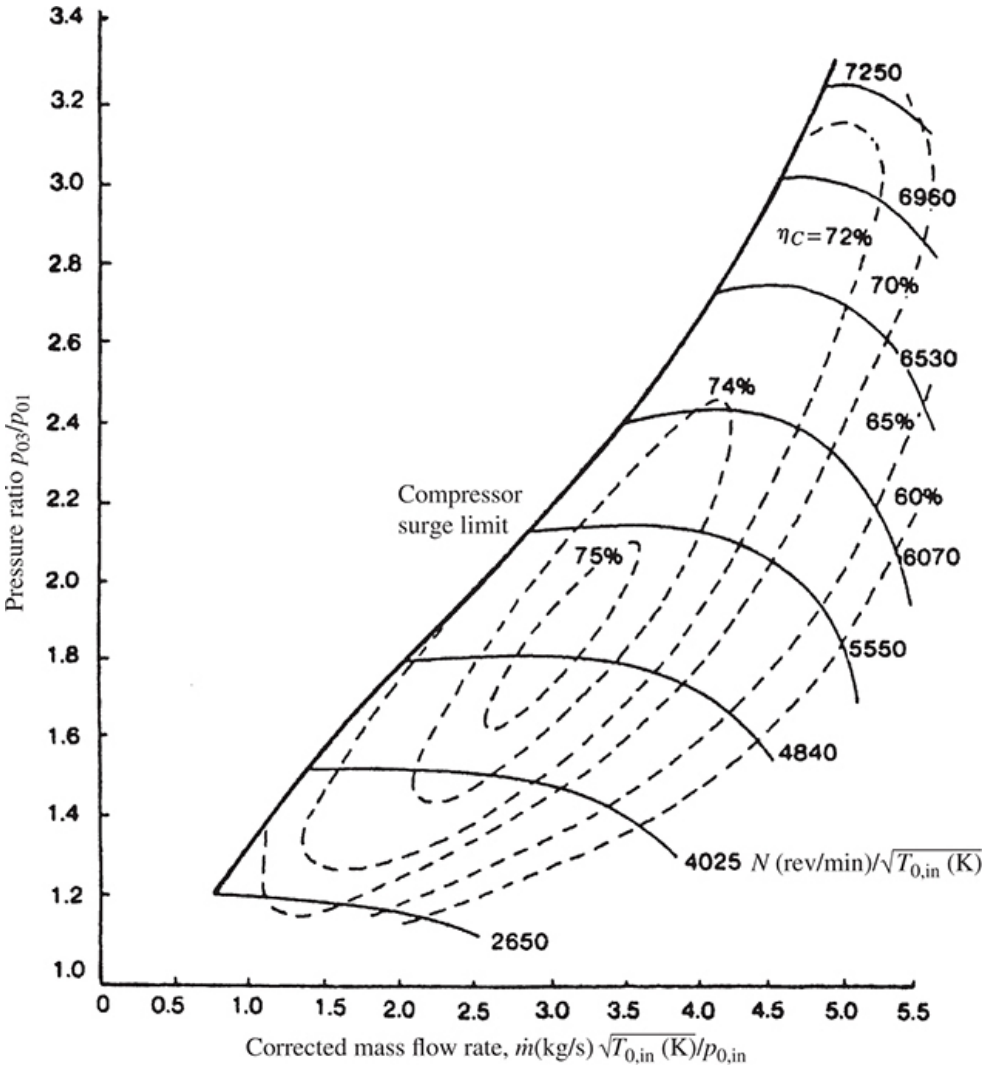


Figure 6.54 Centrifugal compressor operating map. Lines of constant corrected speed and compressor efficiency are plotted on a graph of pressure ratio against corrected mass flow. [41] T_{01} = inlet temperature (K), p_{01} = inlet pressure (bar), p_{03} = exit pressure (bar), N = speed (rev/min), m = mass flow rate (kg/s). Figure 6.51 defines the end states.

where T_{ref} and p_{ref} are standard atmospheric temperature and pressure, respectively. Though the details of different compressor maps vary, their general characteristics are similar. The high efficiency region runs parallel to the surge line (and close to it for vaneless diffusers). A wide *flow range* for the compressor (see Fig. 6.46) is important in turbochargers used for transportation applications.

6.8.4 Turbines

The turbocharger turbine is driven by the energy available in the engine exhaust. The ideal energy available is the blowdown work transfer produced by reversibly expanding the gas in the cylinder at exhaust valve opening to atmospheric pressure, and (for the four-stroke cycle engine) the work done by the piston displacing the gases remaining in the cylinder after blowdown.

The reciprocating internal combustion engine is inherently an unsteady pulsating flow device. Turbines can be designed to accept such an unsteady flow, but they operate more efficiently under steady-flow conditions. In practice, two approaches for recovering a fraction of the available exhaust energy are commonly used: constant-pressure turbocharging and pulse turbocharging. In *constant-pressure turbocharging*, an exhaust manifold of sufficiently large volume to damp out the mass flow and pressure pulses is used so that the flow to the turbine is essentially steady. The disadvantage of this approach is that it does not make full use of the high kinetic energy of the gases leaving the exhaust port; the losses inherent in the mixing of this high-velocity gas with a large volume of low-velocity gas cannot be recovered. With *pulse turbocharging*, short small-cross-section pipes connect each exhaust port to the turbine so that much of the kinetic energy associated with the exhaust blowdown can be utilized. By suitably grouping the different cylinder exhaust ports so that the exhaust pulses are sequential and have minimum overlap, the flow unsteadiness can be held to an acceptable level. The turbine must be specifically designed for this pulsating flow to achieve adequate efficiencies. The combination of increased energy available at the turbine, with reasonable turbine efficiencies, results in the pulse system being more commonly used in engines with mostly transient operation since it provides faster turbocharger response (primarily automotive engines). Constant-pressure turbocharging is used in larger, highly charged, slow-speed diesel engines in stationary use where loads are largely steady. [42]

Two types of turbines are used in turbochargers: radial and axial flow turbines. The radial flow turbine is similar in appearance to the centrifugal compressor: however, the flow is radially inward not outward. Radial flow turbines are normally used in automotive or truck applications. Larger engines—locomotive, stationary, or marine—may use axial flow turbines.

A drawing of a radial flow turbine is shown in Fig. 6.55. It consists of an inlet casing or scroll, a set of inlet nozzles (often omitted with small

turbines), and the turbine rotor or wheel. The function of each component is evident from the *h-s* diagram and velocity triangles in Fig. 6.56. The nozzles (01-2) accelerate the flow, with modest loss in stagnation pressure. The drop in stagnation enthalpy, and hence the work transfer, occurs solely in the rotor passages, 2-3: hence, the rotor is designed for minimum kinetic energy $C_3^2/2$ at exit. The velocity triangles at inlet and exit relate the work transfer to the change in angular momentum via the Euler equation:

$$\dot{W}_T = T\omega = \dot{m}\omega(r_2 C_{\theta 2} - r_3 C_{\theta 3}) = \dot{m}(U_2 C_{\theta 2} - U_3 C_{\theta 3}) \tag{6.63}$$

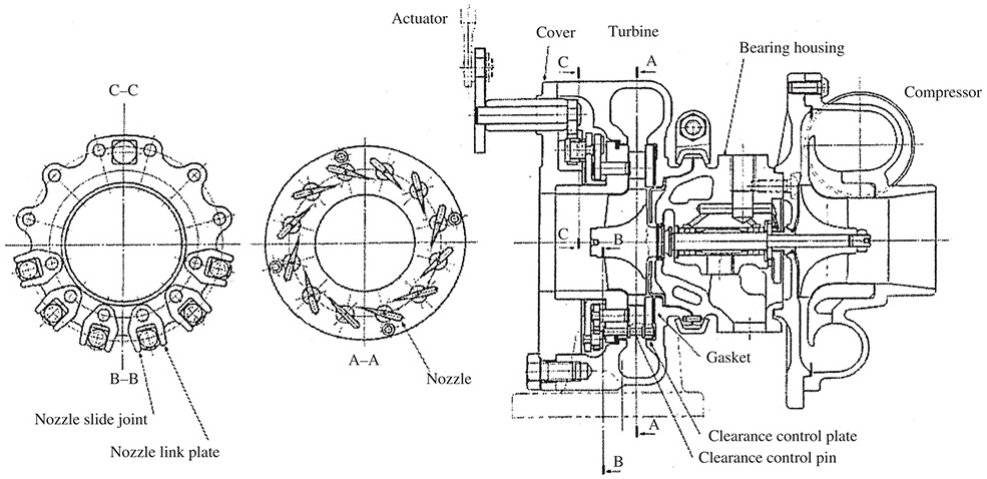


Figure 6.55 Cross-section and layout drawings of a radial flow turbine for a passenger car diesel engine with variable geometry nozzle inlet vanes.[43]

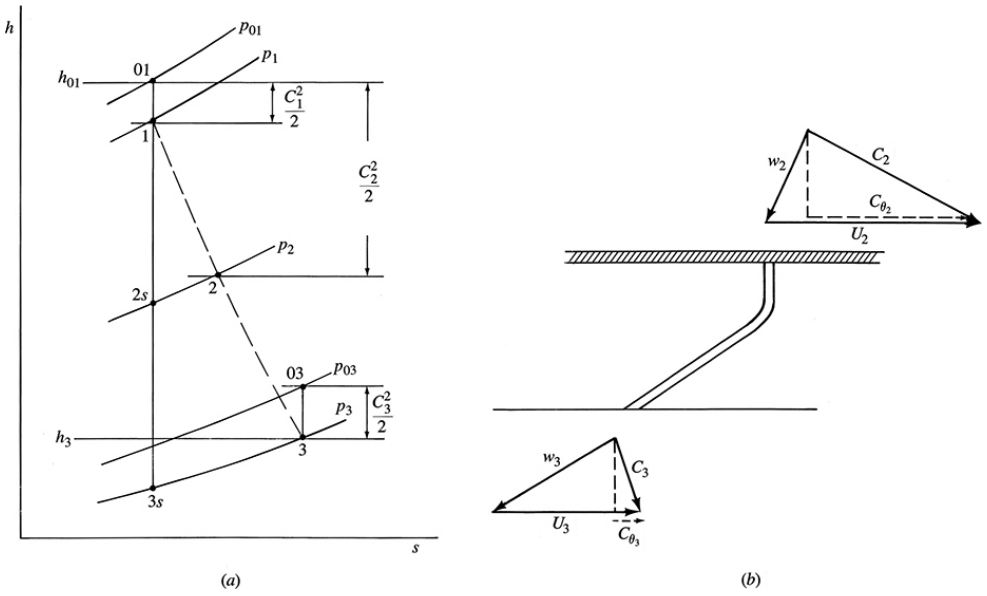


Figure 6.56 (*a*) Enthalpy-entropy diagram for radial turbine; (*b*) velocity diagrams at turbine rotor inlet (2) and exit (3).

where T is the torque and ω the rotor angular speed. Variable geometry nozzles can better direct the flow entering the rotor to be tangential to the vanes. For maximum work transfer the exit velocity should be axial. The work-transfer rate relates to the change in stagnation enthalpy via

$$\dot{W}_T = \dot{m}(h_{02} - h_{03}) = \dot{m}(h_{01} - h_{03}) \tag{6.64}$$

The turbine isentropic efficiency is given by Eqs. (6.44) to (6.47).

Many different types of plots have been used to define radial flow turbine characteristics. Figure 6.57 shows lines of constant corrected speed and efficiency on a plot of corrected mass flow rate versus pressure ratio, with the actual turbine efficiency. As flow rate increases at a given speed, it approaches a limit corresponding to the flow becoming choked in the nozzle blades or the rotor. For turbines, efficiency is often presented separately because the turbine operating regime is narrow.

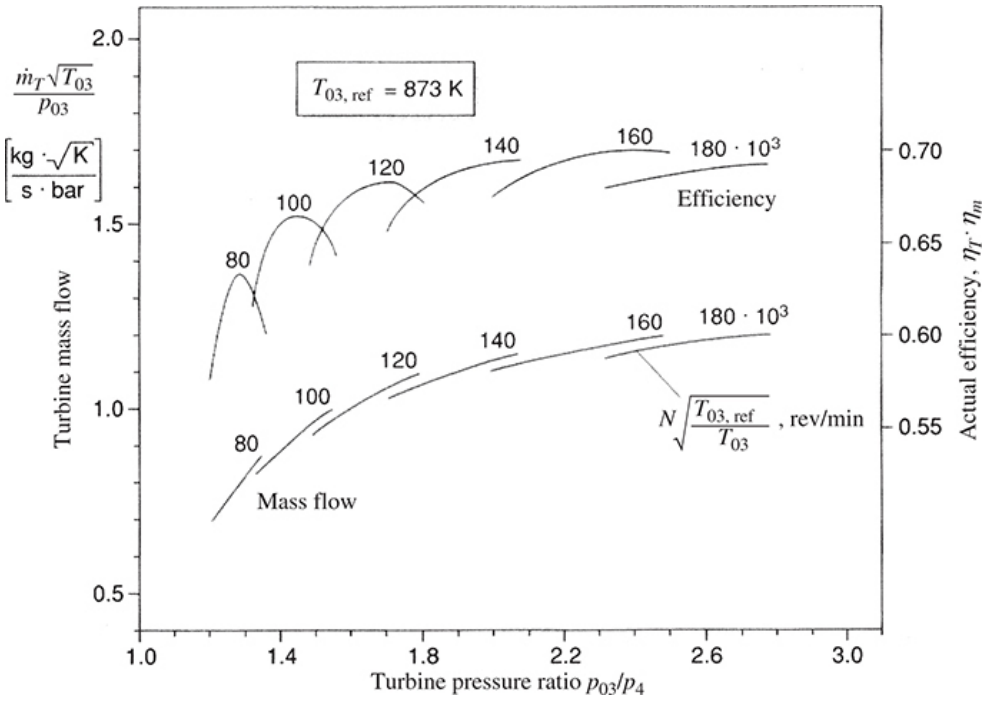


Figure 6.57 Radial turbine performance map showing lines of constant corrected speed on plots of corrected mass flow rate and actual efficiency

versus turbine pressure ratio. T_{01} = turbine inlet temperature (K), p_{01} = turbine inlet pressure (bar), p_3 = turbine exit pressure (bar), p_{03} = mass flow rate (kg/s), N = speed (rev/min). See Figs. 6.49 and 6.50 for nomenclature. (*Source: BorgWarner Turbo Systems*. [41])

A schematic of a turbocharger axial-flow turbine is shown in Fig. 6.58. Usually a single stage is sufficient to expand the exhaust gas efficiently through the pressure ratios associated with engine turbocharging. This turbine consists of an annular flow passage, a single row of nozzles or stator blades, and a rotating blade ring. The changes in gas state across each component are similar to those of the radial turbine shown in the *h-s* diagram of Fig. 6.56*a*. The velocity triangles at entry and exit to the rotor, shown in Fig. 6.59, relate the work transfer from the gas to the rotor to the change in angular momentum:

$$\dot{W}_T = \omega T = \dot{m}\omega(r_2 C_{\theta 2} + r_3 C_{\theta 3})$$

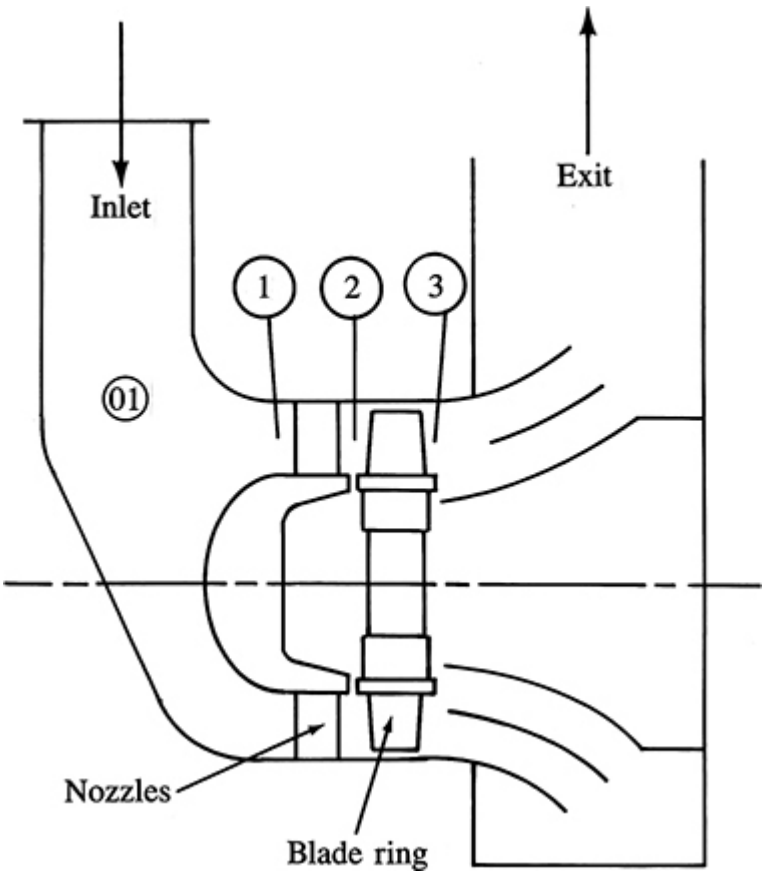


Figure 6.58 Schematic of single-stage axial flow turbine.

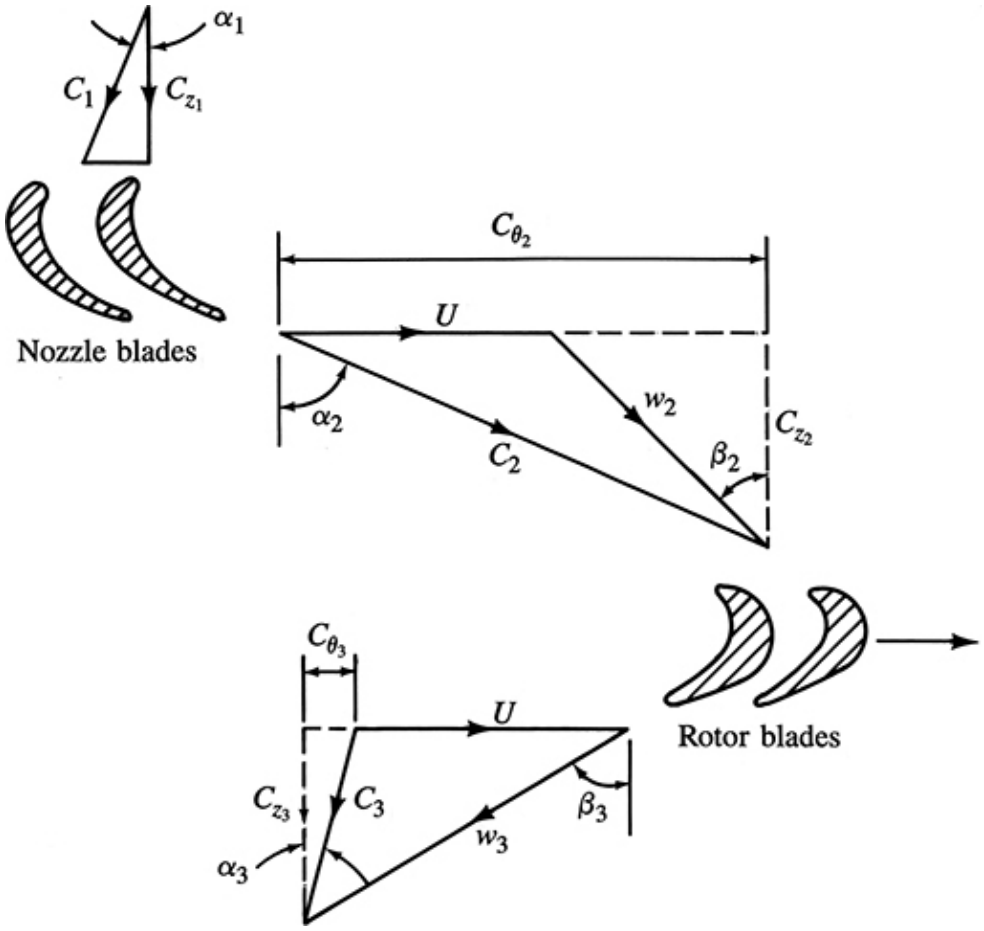


Figure 6.59 Velocity diagrams at entry (2) and exit (3) to axial-flow turbine blade ring.[38]

Since the mid-radius r_2 usually equals the mid-radius r_3,

$$\dot{W}_T = \dot{m}U(C_{\theta 2} + C_{\theta 3}) = \dot{m}U(C_2 \sin\alpha_2 + C_3 \sin\alpha_3)$$
$$= \dot{m}U(C_{z2} \tan\beta_2 + C_{z3} \tan\beta_3) \qquad (6.65)$$

Equation (6.64) relates the work-transfer rate to the stagnation enthalpy change as in the radial turbine.

Figure 6.60 shows axial turbine performance characteristics on the standard dimensionless plot of pressure ratio versus corrected mass flow rate. Here the constant speed lines converge to a single choked flow limit as

the mass flow is increased. In the radial turbine, the variation in centrifugal effects with speed cause a noticeable spread in the constant speed lines (Fig. 6.57).

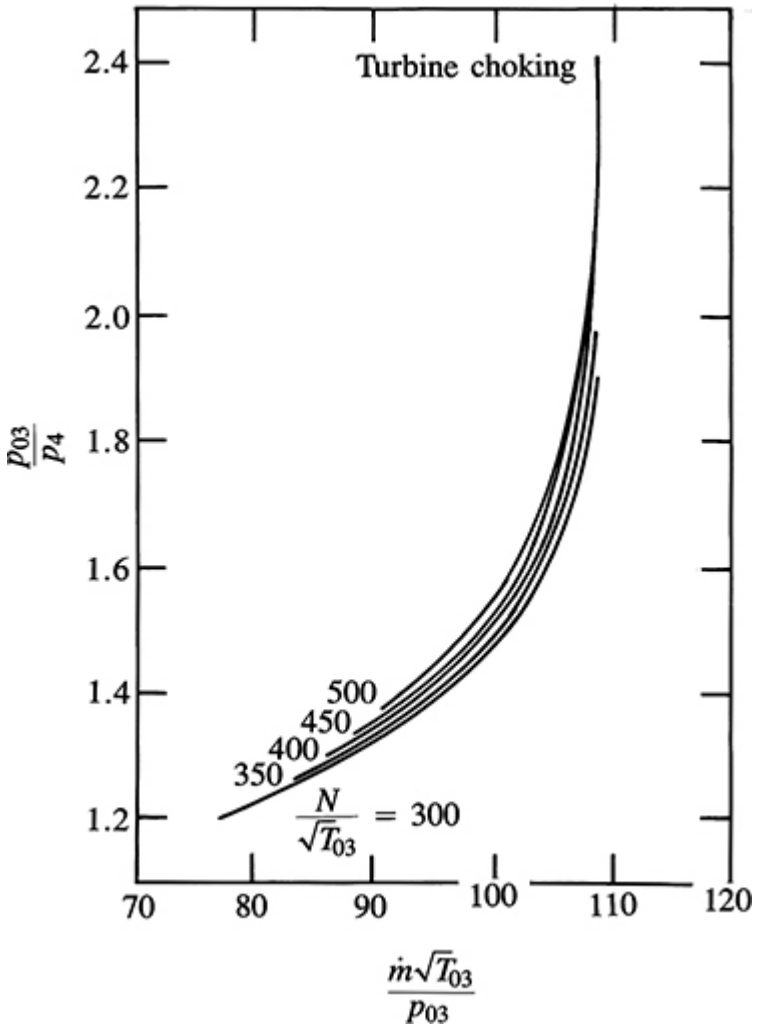


Figure 6.60 Axial flow turbine performance map: pressure ratio is plotted against corrected mass flow rate. T_{03} = turbine inlet temperature (K), p_{03} = turbine inlet pressure (bar), p_4 = turbine exit pressure (bar), p_3 = mass flow rate (kg/s), N = speed (rev/min).[38]

An alternative performance plot for turbines is efficiency versus *blade speed ratio*. This ratio is the blade speed U (at its mean height for an axial flow turbine or the wheel tip speed for a radial flow turbine), divided by the

velocity equivalent of the isentropic enthalpy drop across the turbine stage, C_s: that is,

$$\text{Blade speed ratio} = \frac{U}{C_s}$$

where

$$C_s = [2(h_{03} - h_{4s})]^{1/2} \tag{6.66}$$

This method of displaying performance is useful for matching compressor and turbine wheel size for operation of the turbine at optimum efficiency. Figure 6.61 shows such plots for an axial and radial flow turbine. The peak efficiency can occur for *U/Cs* between 0.5 and 0.8, depending on turbine design and application. [38]

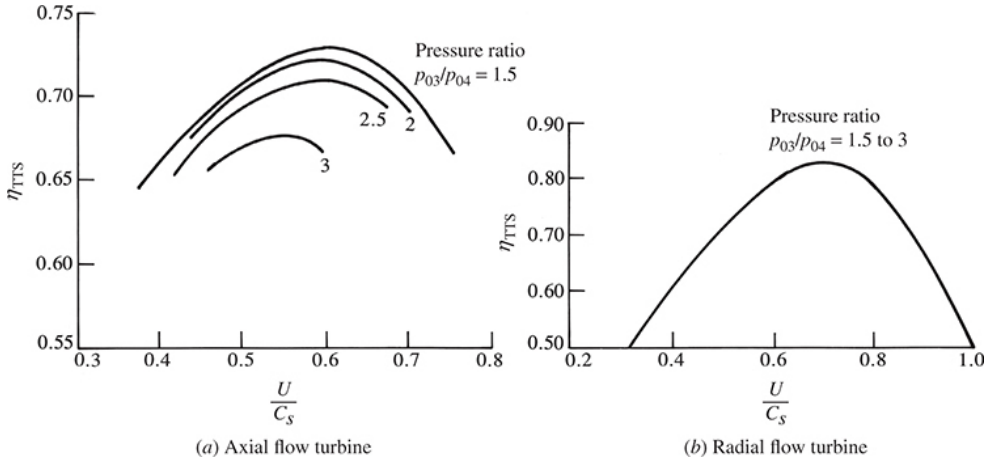


Figure 6.61 Plot of turbine total-to-static efficiency versus blade speed ratio *U/Cs* for (*a*) axial-flow and (*b*) radial-flow turbines.[38]

6.8.5 Compressor, Engine, Turbine Matching

The mass flows through the compressor, engine, turbine, and wastegate in a turbocharged engine must be consistent. The turbine inlet temperature depends on how the engine is operating: fuel and air flow rates, engine speed and power, are key variables. The turbine supplies the power that drives the compressor, and turbine and compressor operate at the same speed. The sum

of the turbine mass flow rate plus that through the wastegate equals the engine exhaust flow. Further, the relative geometric sizing of the turbocharger and the engine affect their joint performance (especially as a function of engine speed). Figure 6.62 illustrates the connections between these components, and the matching logic.

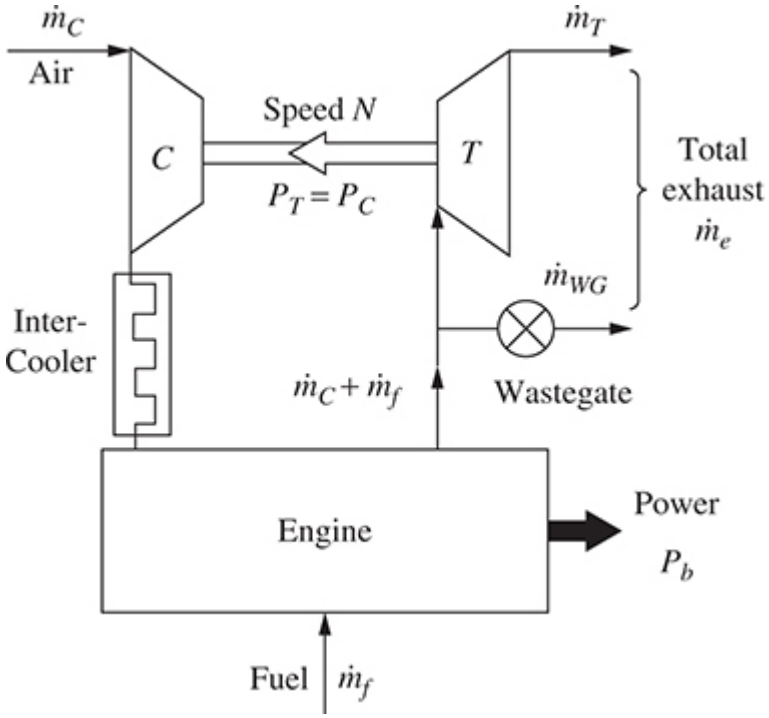


Figure 6.62 Schematic illustrating compressor, engine, turbine, matching.

Note that for the turbocharger to be providing significant boost, the engine must be operating at mid to higher load. Under lightly loaded conditions, especially at lower engine speeds, there is insufficient engine exhaust flow to generate enough turbine power to provide significant air compression. Under these conditions, the turbocharger effectively "idles." Accordingly, a *boost threshold* is defined: It is the engine speed (rev/min), at wide- open throttle, at which turbocharger air compression starts to occur.

For a given turbocharger, the compressor and turbine characteristics are linked with the engine as indicated in Fig. 6.62. Since the compressor and turbine are on a common shaft with speed N, their respective corrected speeds ($N/\sqrt{T_{01}}$ and $N/\sqrt{T_{03}}$) are linked by:

$$\frac{N}{\sqrt{T_{01}}} = \frac{N}{\sqrt{T_{03}}}\left(\frac{T_{03}}{T_{01}}\right)^{1/2} \tag{6.67}$$

For $\dot{m}_C = \dot{m}_T = \dot{m}$ their corrected mass flow rates are linked by:

$$\frac{\dot{m}\sqrt{T_{01}}}{p_{01}} = \frac{\dot{m}\sqrt{T_{03}}}{p_{03}}\left(\frac{p_{03}}{p_{01}}\right)\left(\frac{T_{01}}{T_{03}}\right)^{1/2} \tag{6.68}$$

When $\dot{m}_C\left[1+(F/A)\right] = \dot{m}_T$, the equation is easily modified.

Since the compressor and turbine powers are equal in magnitude, when $\dot{m}_C$ equals $\dot{m}_T$,

$$h_{02} - h_{01} = \eta_m (h_{03} - h_{04}) \tag{6.69}$$

or, with an ideal gas model,

$$c_{p,C}(T_{02} - T_{01}) = \eta_m c_{p,T}(T_{03} - T_{04}) \tag{6.70}$$

Equation (6.70) , with Eqs. (6.44) and (6.50), gives

$$\left(\frac{p_{02}}{p_{01}}\right)^{(\gamma_C - 1)/\gamma_C} - 1 = \eta_C \eta_T \eta_m \frac{c_{p,T}}{c_{p,C}}\left[1 - \left(\frac{p_4}{p_{04}}\right)^{(\gamma_T - 1)/\gamma_T}\right]\frac{T_{03}}{T_{01}} \tag{6.71}$$

Assuming that the turbine exit pressure p_4 equals atmospheric pressure p_{01}, the equilibrium or steady-state running lines for constant values of T_{03}/T_{01} can be determined.

Figure 6.63 shows an example of such a set of matched turbocharger characteristics, plotted on a turbocharger compressor map for a radial turbine with characteristics similar to Fig. 6.57.

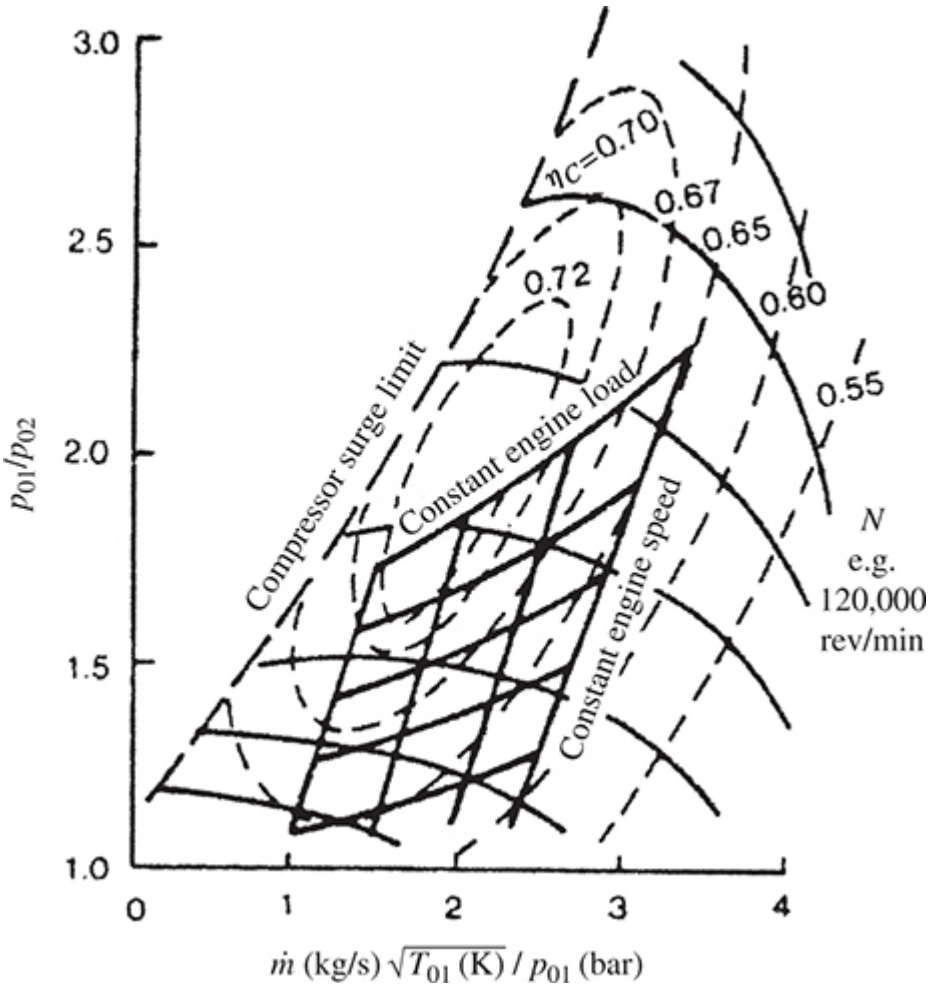


Figure 6.63 Illustration of steady-state turbocharger operating characteristics: compressor operating map (such as that in Fig. 6.54), with the airflow requirements of a four-stroke truck diesel engine superposed.[41] Mass flows through the compressor, engine, and turbine have to be consistent. See Fig. 6.62.

The problem of overspeeding the turbocharger and generating high boost levels, and very high cylinder pressures often requires that some of the exhaust be bypassed around the turbine. A bypass valve or *wastegate* (as shown in Fig. 6.64) is usually built into the turbocharger casing. It consists of a spring-loaded valve acting in response to the intake manifold pressure acting on a controlling diaphragm. When the wastegate is open, only a portion of the exhaust gases will flow through the turbine and generate power; the remainder passes directly into the exhaust system downstream of the turbine.

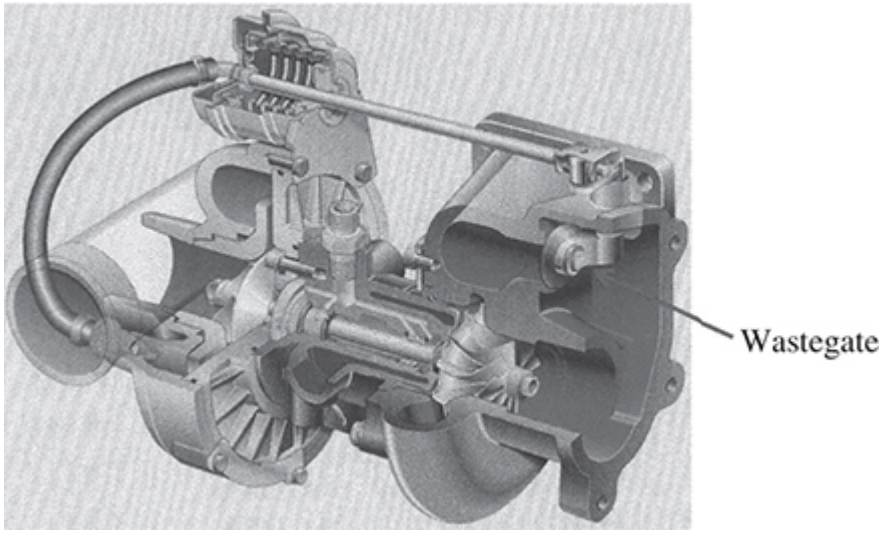


Figure 6.64 Cutaway drawing of turbocharger with a wastegate. (*Courtesy: BorgWarner Turbo Systems*.)

The relative sizing of the turbocharger and the engine significantly influences their combined behavior. A relatively smaller turbocharger provides higher compressor boost levels in the lower engine speed range, and less boost at higher engine speeds. A larger turbocharger has the inverse behavior: lower boost at lower speeds and higher boost at higher speeds. Higher boost at lower speeds is effective at offsetting "turbo lag": a phenomenon in automotive applications where, following a rapid opening of the throttle valve, the rotational inertia of the turbocharger delays its speeding up which delays the production of the desired increase in boost pressure (and thus airflow and torque).

In the commonly used turbocharger radial turbine, the *A/R* ratio is used as a measure of the above turbocharger characteristics (see Fig. 6.65*a*). *A* is the cross-sectional area of the turbine housing throat, and *R* is the distance between the center of the throat and the turbocharger shaft axis. There is interest for gasoline engine applications in using a variable flow turbocharger which uses a control valve located in the turbine intake port (see Fig. 6.65*b*) to reduce the turbine intake cross-sectional area at low speeds and flows and to increase that flow area at high speeds. The turbine has two scrolls, inner and outer circumferences, separated by a fixed vane. With low intake airflow, the outer scroll is closed off and higher torque at

low engine speed is achieved. With high intake airflow, both scrolls are open achieving high torque at high speed. [43]

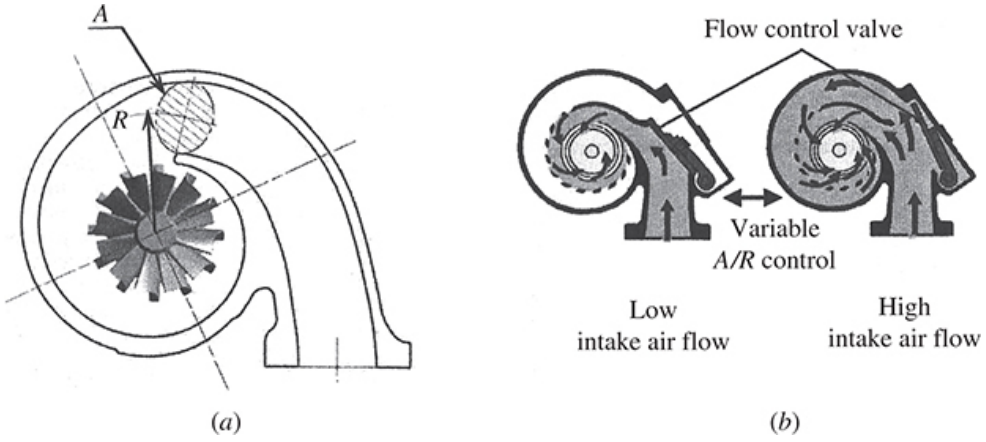


Figure 6.65 (*a*) Definition of *A/R* for a conventional turbine and (*b*) schematic of flow control valve.[43]

6.8.6 Wave-Compression Devices

Pressure wave superchargers make use of the fact that if two fluids having different pressures are brought into direct contact in long narrow channels, equalization of pressure occurs faster than mixing. One such device, the Comprex, has been developed for internal combustion engine supercharging which operates using this principle. [44] It is shown schematically in Fig. 6.66. The working channels of the Comprex are arranged on a rotor or cell wheel (*b*) which is rotated between two castings by a belt driven from the crankshaft (*c*). There is no contact between the rotor and the casing, but the gaps are kept small to minimize leakage. The belt drive merely overcomes friction and maintains the rotor at a speed proportional to engine speed (usually four or five times faster): It provides no compression work. One casing (the air casing) contains the passage which brings low-pressure air (*f*) to one set of ports and high-pressure air (*e*) from another set of ports in the rotor-side inner casing. The other casing (the gas casing) connects the high-pressure engine exhaust gas (*d*) to one set of ports at the other end of the rotor, and connects a second set of ports to the exhaust system (*g*). Fluid can flow into and out of the rotor channels through these ports. The exhaust gas inlet port is made small enough to cause a significant pressure rise in the exhaust

manifold (e.g., 2 atm) when the engine is operated at its rated power. The pressure wave process does not depend on the pressure and flow fluctuations within the manifold caused by individual cylinder exhaust events: its operation can be explained assuming constant pressure at each set of ports. As the rotor makes one revolution, the ends of each channel are alternatively closed, or are open to a flow passage. By appropriate arrangement of these passages and selection of the geometry and location of the ports, an efficient energy transfer between the engine exhaust gases and the fresh charge can be realized. [44]

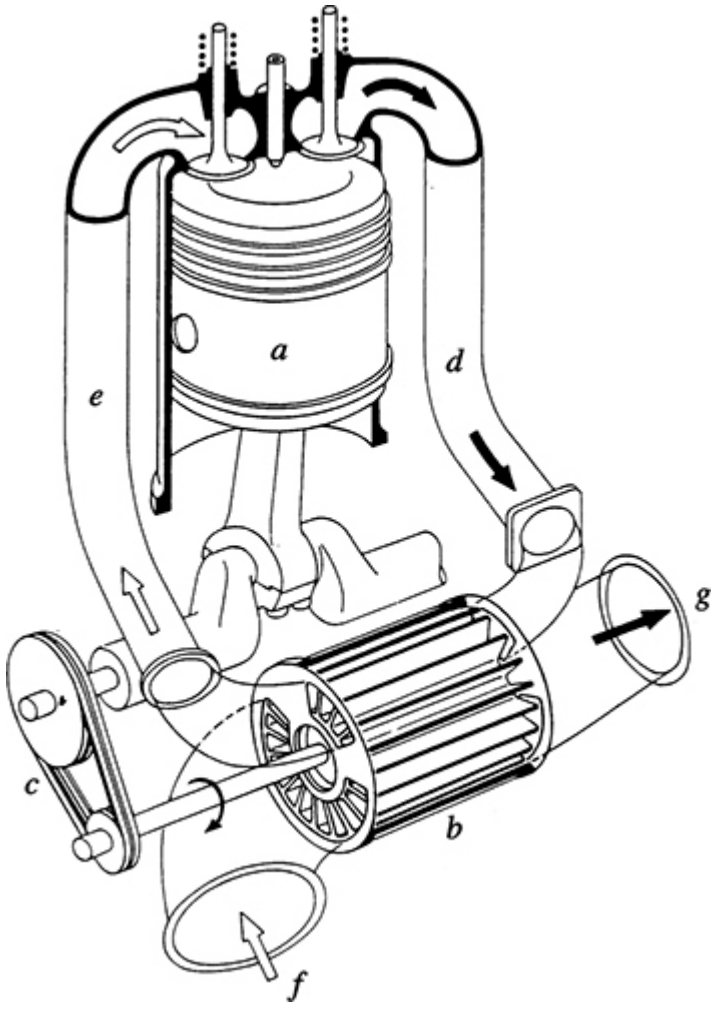


Figure 6.66 Schematic of Comprex supercharger.[44] *a* engine, *b* cell wheel or rotor, *c* belt drive, *d* high-pressure exhaust gas (G-HP), *e* high-pressure air (A-HP), *f* low-pressure air (A-LP), *g* low-pressure exhaust gas (G-LP).

The wave-compression process in the Comprex can be explained in more detail with the aid of Fig. 6.67 where the rotational motion of the channels has been unrolled. Consider the channel starting at the top; it is closed at both ends and contains air at atmospheric pressure. As it opens at the upper edge of the high-pressure gas (G-HP) duct, a compression or shock wave (1) propagates from the right end of the channel toward the left, compressing the air through which it passes. The compressed air behind the wave occupies less space so the high-pressure exhaust gas moves into the channel as indicated by the dotted line. This line is the boundary between the two fluids. As this wave (1) reaches the left end, the channel is opened and compressed air flows into the engine inlet duct (A-HP). The inlet duct is shaped to provide the same mass flow at lower velocity: This deceleration of the air produces a second compression wave (2) which propagates back into the channel. As a result the compressed air leaving the cell on the left has a higher pressure than the driving gas on the right. As this wave (2) arrives at the right-hand side, the high-pressure gas (G-HP) channel closes. An expansion wave (3) then propagates back to the left, separating the now motionless and partly expanded fluid on the right from still-moving fluid on the left. When this wave (3) reaches the left-hand end, A-HP is closed and all the gases in the channel are at rest. Note that the first gas particles (dotted line) have not quite reached the air end of the channel: A cushion of air remains to prevent breakthrough.

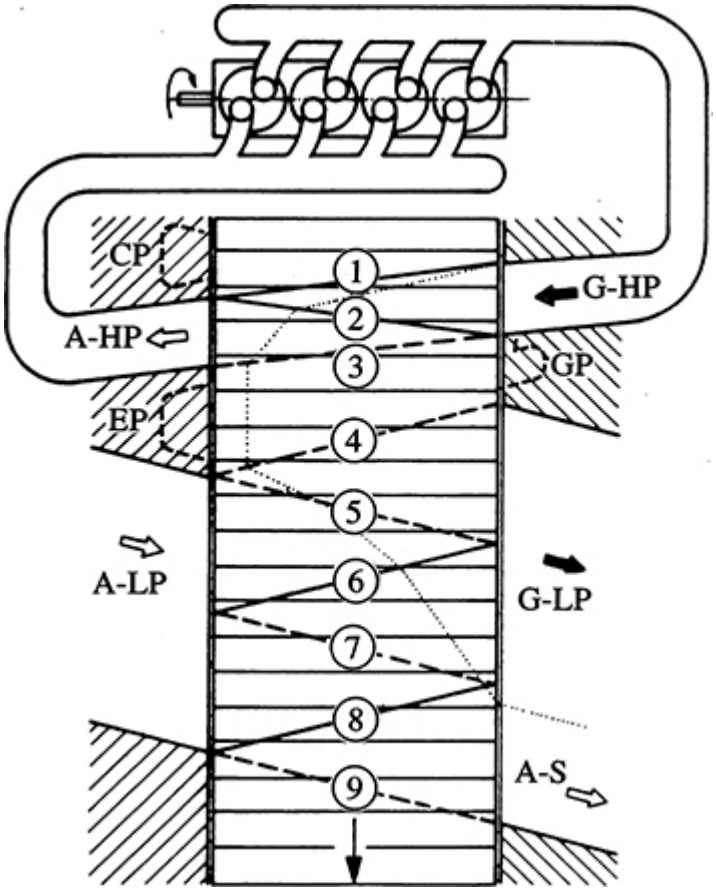


Figure 6.67 Unrolled view of the Comprex pressure-wave process.[44] A air, G exhaust gas, S scavenging, HP high pressure, LP low pressure; CP, EP, GP are pockets.

The cell's contents are still at a higher pressure than the low pressure in the exhaust gas duct. When the right-hand end of the cell reaches this duct, the cell's contents expand into the exhaust. This motion is transferred through the channel by an expansion wave (4) which propagates to the left at sonic speed. When this wave reaches the left-hand end, the cell opens to the low-pressure air duct (A-LP) and fresh air is drawn into the cell. The flow to the right continues, but with decreasing speed due to wave action (5, 6, 7, 8) and pressure losses at each end of the cell. When the dotted line—the interface between air and the exhaust gas—reaches the right end of the cell, all the driving gas has left. The cell is then purged by the scavenging airflow (A-S) and filled with fresh air at atmospheric pressure. At wave (9), the cell is closed at both ends, restoring it to its initial state. [45]

The speed of these pressure waves is the local sound speed and is a function of local gas temperature only. Thus, the above process will only work properly for a given exhaust gas temperature at a particular cell speed. The operating range is extended by the use of "pockets" as shown in Fig. 6.67. The pockets prevent the reflection of sound waves from a closed channel end which would cause a substantial change in flow velocity in the channel. These pockets, marked CP and EP on the air side and GP on the exhaust gas side, allow flow from one channel to adjacent channels via the pocket if the wave action requires it. Thus the device can be tuned for full-load medium-speed operation and still give acceptable performance at other loads and speeds because the pockets allow the particle paths to change without major losses. [44]

The values of compressor efficiency [defined by Eq. (6.41)] are comparable to those of mechanical and aerodynamic compressors over a wide range of engine speeds.[44]

PROBLEMS

6.1. For four-stroke cycle engines, the inlet and exhaust valve opening and closing crank angles are typically: IVO 15° BTC; IVC 50° ABC; EVO 55° BBC; EVC 10° ATC. Explain why these valve timings improve engine breathing relative to valve opening and closing at the beginnings and ends of the intake and exhaust strokes. Discuss the additional design issues that are important.

6.2. For four-stroke engines, the inlet and exhaust valve opening and closing crank angles are typically:

IVO: 15° BTC
IVC: 50° ABC
EVO: 55° BBC
EVC: 10° ATC

(a) Explain why these valve timings improve engine breathing relative to valve opening and closing at the beginnings and ends of the intake and exhaust strokes.

(b) Mention and explain at least one modification you would make to the above valve timings if the engine you are designing is:

- intended for a race car
- turbocharged
- operating at low speeds

(c) With fully variable valve timing to improve the wide-open-throttle (WOT) performance of this spark-ignition engine, explain how you would change these "fixed-valve-timing" opening and closing angles.

6.3. A conventional spark-ignition engine operating with gasoline will not run smoothly (due to incomplete combustion) with an equivalence ratio leaner than about $\phi = 0.8$. It is desirable to extend the smooth operating limit of the engine to leaner equivalence ratios so that at part-throttle operation (with intake pressure less than 1 atmosphere) the pumping work is reduced. Leaner than normal operation can be achieved by adding hydrogen gas (H_2) to the mixture in the intake system. The addition of H_2 makes the fuel-air mixture easier to burn.

(a) The fuel composition with "mixed" fuel operation is $H_2 + C_8H_{18}$—one mole of hydrogen to every mole of gasoline, which is assumed the same as isooctane. What is the stoichiometric air/fuel ratio for the "mixed" fuel?

(b) The lower heating value of H_2 is 120 MJ/kg and for isooctane is 44.4 MJ/kg. What is the heating value per kilogram of fuel mixture?

(c) Engine operation with isooctane and the mixed ($H_2 + C_8H_{18}$) fuel is compared in a particular engine at a part-load condition (brake mean effective pressure of 275 kPa and 1400 rev/min). You are given the following information about the engine's operation:

Fuel	C_8H_{18}	$H_2 + C_8H_{18}$
Equivalence ratio	0.8	0.5
Gross indicated fuel conversion efficiency	0.35	0.4
Mechanical + accessory friction mep	138 kPa	138 kPa
Inlet manifold pressure	46 kPa	?
Pumping mep	55 kPa	?

Estimate approximately the inlet manifold pressure and the pumping mean effective pressure with ($H_2 + C_8H_{18}$) fuel. Explain your method and assumptions clearly. Note that mechanical efficiency η_m is defined as

$$\eta_m = \frac{\text{bmep}}{\text{imep}_g} = \frac{\text{bmep}}{\text{bmep} + \text{mfmep} + \text{pmep}}$$

6.4. With variable-valve-timing spark-ignition engines, by closing the intake valve much later than a standard fixed-valve-timing engine, the pumping work can be reduced.

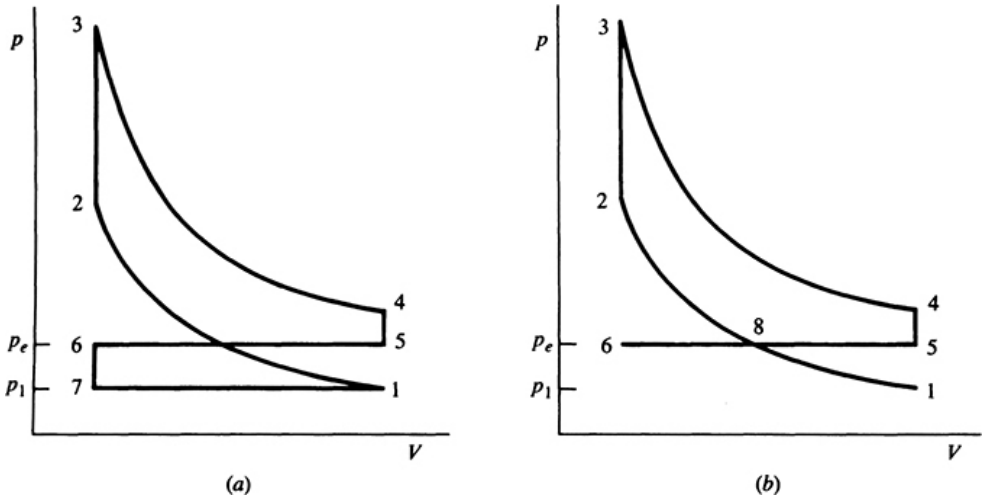
(a) The average brake mean effective pressure (bmep) required to drive a car is 250 kPa. The intake manifold pressure with a standard fixed intake-valve-timing engine (closing 50° ABC) is 50 kPa and the exhaust manifold pressure is 105 kPa. The mechanical plus accessory friction mep is 90 kPa. What indicated (gross) mep must the engine produce? The intake valve closing angle is now retarded so that the intake manifold pressure is 90 kPa. What imep (gross) will now be required? Other parameters remain the same.

(b) Sketch a carefully drawn and proportioned cylinder pressure versus cylinder volume graph showing the *real engine* four-stroke cycles for the fixed and variable intake valve closings, with other details as in (*a*). Clearly identify the two intake and the exhaust manifold pressures, the intake valve closing locations, and ensure that areas on the *p-V* diagram that correspond to any of these various mean effective pressures are in the appropriate proportions. Think carefully about the details of what is happening!

(c) Estimate the ratio of the cylinder volume at which the late intake-valve-timing engine should end the intake process, to the cylinder volume at which the standard intake-valve-timing engine ends the intake process, to give the same bmep (250 kPa) as the standard valve-timing engine. Give an approximate estimate of the late closing angle (you do not need to solve the volume versus crank angle equation to get this crank angle: You can use a simpler approximate approach). Both engines operate at stoichiometric. They have the same indicated fuel conversion efficiency.

6.5. Sketch (*a*) shows an ideal cycle *p-V* diagram for a conventional throttled spark-ignition engine, 1-2-3-4-5-6-7-1. The gas properties cv , cp , γ, R throughout the cycle are constant. The mass of gas in the cylinder is m. The exhaust pressure is p_e.

Sketch (*b*) shows an ideal cycle *p*-*V* diagram 1-2-3-4-5-6-8-1 for a spark-ignition engine with different intake valve timing. The intake manifold is unthrottled; it has essentially the same pressure as the exhaust. To reduce the mass inducted at part load, the inlet valve is closed *partway through the intake stroke* at point 8. The gas in the cylinder at intake valve closing at 8 is then expanded isentropically to 1 with the intake valve closed. The pressure p_1 at the start of compression is the same for both cycles.



(a) Indicate on *p*-*V* diagrams the area that corresponds to the pumping work per cycle for cycles (*a*) and (*b*). Which area is greater?
(b) Derive expressions for the pumping work per cycle W_p in terms of *m*, *cv* , γ, *Tu* (p_e /p_1), and the compression ratio r_c for cycles (*a*) and (*b*). Be consistent about the signs of the work transfers to and from the gas.
(c) For $\gamma = 1.3$, $r_c = 8$, and (p_e /p_1) = 2, find the ratio $W_p(b)/Wp(a)$, assuming the values of T_1 and *m* are the same in both cases.

6.6. Estimate approximately the pressure drop across the inlet valve about halfway through the intake stroke, and across the exhaust valve halfway through the exhaust stroke, when the piston speed is at its maximum for a typical four-stroke cycle spark-ignition engine with $B = L = 85$ mm at 2500 and 5000 rev/min at WOT. Assume appropriate values for any valve and port geometric details required, and for the gas composition and state.

6.7. Compare the engine residual gas fraction data in Fig. 6.26 with ideal cycle estimates of residual gas fraction as follows. Using Eq. (5.47) plot the

fuel-air cycle residual mass fraction x_r against p_i/p_e for $r_c = 8.5$ on the same graph as the engine data in Fig. 6.19 at 1400 rev/min and 27° valve overlap. Assume $T_r = 1400$ K and $(\gamma - 1)/\gamma = 0.24$ in Eq. (5.47) . Suggest an explanation for any significant differences.

6.8. One concept that would increase SI engine efficiency is early intake valve closing (EIVC) where the intake valve closes *before* the piston reaches BC on the intake stroke, thus limiting the amount of charge inducted into the cylinder.

(a) Explain why EIVC improves engine efficiency at part load. (*Hint*: Consider what must happen to the inlet manifold pressure in order to maintain constant mass in the cylinder as the intake valve is closed sooner.)

(b) This part load reduction in charge could be achieved by using late intake valve closing where the intake valve is not closed until the compression stroke has pushed some of the cylinder gases back out into the intake manifold. Based on a comparison of *p-V* diagrams, is this method inferior to EIVC?

6.9. An eight-cylinder turbocharged aftercooled four-stroke cycle diesel engine operates with an inlet pressure of 1.8 atmospheres at its maximum rated power at 2000 rev/min. $B = 128$ mm; $L = 140$ mm; ηv (based on inlet manifold conditions of 1.8 atm and 325 K after the aftercooler) = 0.9. The compressor isentropic efficiency is 0.7.

(a) Calculate the power required to drive the turbocharger compressor.

(b) If the exhaust gas temperature is 650°C and the turbocharger isentropic efficiency is 0.65, estimate the pressure at turbine inlet. The turbine exhausts to the atmosphere.

6.10. The charging efficiency of two-stroke cycle diesel engines can be estimated from measurement of the concentration of O_2 and CO_2 in the burned gases within the cylinder, or in the exhaust blowdown pulse prior to any mixing with fresh air. The engine bore = 125 mm; stroke = 150 mm; compression ratio = 15. The fuel flow rate at 1800 rev/min is 1.6 g/s per cylinder. The conditions used to evaluate the air density for the reference mass are 300 K and 1 atm. The molar concentrations (dry) of CO_2 and O_2 in the in-cylinder burned gases are 7.2 and 10.4% (see Fig. 4.22). The scavenging airflow rate is 80 g/s. Evaluate (*a*) the charging efficiency, (*b*)

the delivery ratio, and (c) the trapping efficiency (assuming the trapped mass equals the reference mass).

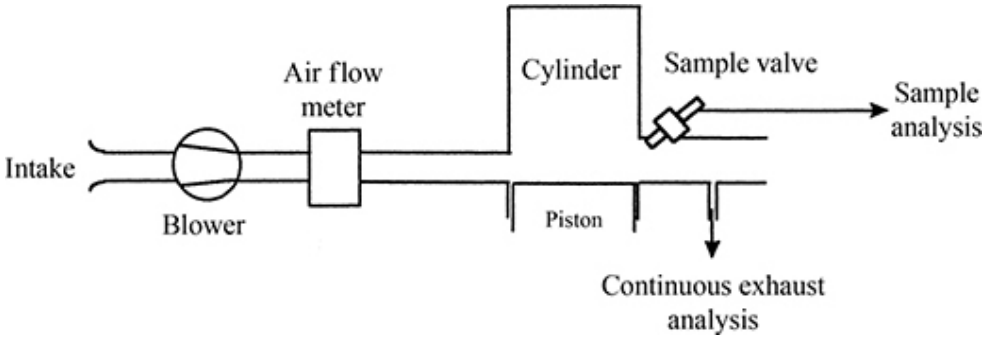
6.11. In order to evaluate the scavenging quality of a two-stroke diesel engine, the composition of in-cylinder gases at the end of combustion has to be determined. For this purpose an electromagnetic gas-sampling valve is being placed next to the exhaust port of a single cylinder test engine. The engine is a two-stroke diesel with direct in-cylinder injection. During the blowdown period (between "exhaust open" and "intake open") the valve samples from the burnt exhaust gases exiting the cylinder into the exhaust system: from an analysis of these samples the equivalence ratio of the in-cylinder gases can be determined. The overall equivalence ratio of the engine (including short-circuiting fresh charge) is determined from a continuous exhaust gas downstream of the exhaust port.

Data:	Bore	$B = 240$ mm
	Stroke	$L = 250$ mm
	Connecting rod length	$L_C = 530$ mm
	Height of intake port	$s_{in} = 48.0$ mm
	Height of exhaust port	$s_{ex} = 80.7$ mm
	Min. opening duration of sampling valve	$t_{min} = 80.7$ mm
	Engine speed	$N = 1000$ rev
	Ambient pressure	$p_{amb} = 1$ bar
	Pressure difference across blower	$\Delta p = 0.32$ bar
	Temperature of scavenge air	$T_{sc} = 313$ K
	Ambient temperature	$T_{amb} = 298$ K
	Volumetric flow rate of scavenge air	$V_{in} = 0.173$ m^3
	In-cylinder equivalence ratio	$\phi_{cyl} = 0.588$
	Overall equivalence ratio	$\phi_{del} = 0.328$

(a) Determine the intake and exhaust timing (in degrees crank angle) of the test engine.

(b) Up to what engine speed can this sampling valve be used for the analysis described?

(c) Determine the delivery ratio, the trapping efficiency, and the charging efficiency of the test engine.



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[a] In a naturally-aspirated engine at wide-open throttle, p_i would be close to atmospheric pressure; in a boosted engine, p_i would be the air pressure at compressor exit.

[b] At higher engine speeds, there is less time per cycle for heat losses to occur. This is only partly offset by the higher heat transfer rates due to higher velocities. Thus the average exhaust gas temperature increases as engine speed increases, and the exhaust gas density decreases.

[c] See Chap. 12 .

[d] This was done with physically based models for each of the factors, appropriately integrated together.[8]

[e] The intake air's temperature (in kelvins, also used in the ideal gas law to calculate density) is not substantially changed, especially with intercooling.

[f] Note that engine friction as speed approaches its highest values has a modest distorting effect on the bmep airflow scaling.

[g] If scavenging is done with fuel-air mixture, as in most two-stroke spark-ignition engines, then mixture mass is used instead of air mass.

[h] The simplest versions of these ideal models assume the processes are isothermal. More complex versions of these models are available (Heywood and Sher [31]).

CHAPTER 7

Mixture Preparation in SI Engines

7.1 SPARK-IGNITION ENGINE MIXTURE REQUIREMENTS

Spark-ignition engines operate by burning a premixed mixture of fuel vapor and air. The task of the engine induction and fuel systems is to prepare such a mixture inside the cylinder from ambient air, and fuel in the tank, that satisfies the requirements of the engine over its entire operating regime. In principle, the optimum fuel-air mixture ratio for a spark-ignition engine is that which gives the required power output with the lowest fuel consumption, consistent with smooth and reliable operation. In practice, the constraints of emissions control may dictate a different mixture composition from this ideal, and may also require a fraction of the exhaust gases to be recycled (EGR—exhaust gas recirculation) into the intake system. The relative proportions of fuel and air that provide the lowest fuel consumption, smooth reliable operation, and satisfy the emissions requirements, at the required power level, depend on engine speed and load. Mixture requirements and preparation are usually discussed in terms of the air/fuel ratio or fuel/air ratio (see Sec. 2.9) and percent EGR [see Eq. (4.2)]. While the fuel metering system is designed to provide the appropriate fuel flow for the *actual* air flow at each speed and load, the relative proportions of fuel and air can be stated more generally in terms of the relative air/fuel ratio λ, or the fuel/air equivalence ratio ϕ, which are the actual air/fuel ratio, or fuel/air ratio, normalized by dividing by the stoichiometric air/fuel or fuel/air ratio [Eqs. (3.8) and (3.9)]. The combustion characteristics of fuel-air mixtures and

the properties of combustion products, which govern engine performance, efficiency, and emissions, correlate best for a wide range of fuels relative to the stoichiometric mixture proportions. Therefore, where appropriate, the relative air/fuel ratio, or the fuel/air equivalence ratio will be used as the defining parameter. A typical value for the stoichiometric air/fuel ratio of gasoline is 14.6. [a] Thus, for gasoline,

$$\lambda \approx \frac{(A/F)}{14.6}; \quad \phi \approx \frac{14.6}{(A/F)} \qquad (7.1a, b)$$

Note that other fuels such as ethyl alcohol, (often blended with gasoline), liquid petroleum gas (LPG), natural gas (NG), and hydrogen are used in spark-ignition engines. Each of these has a different stoichiometric air/fuel ratio. However, their relative air/fuel ratio (or fuel/air equivalence ratio) is still the appropriate mixture composition correlating parameter.

This chapter is about mixture preparation in spark-ignition engines. There is no equivalent chapter on mixture preparation in diesels; that important topic is discussed in Chap. 10 as part of diesel or compression-ignition engine combustion. The reason for the different treatment of this topic for these two types of engines is that, in spark-ignition engines, the fuel-air mixture preparation and the combustion process are largely separable: the first process is normally completed before the second process occurs. In the diesel engine, the preparation of the combustible fuel-air mixture starts when fuel injection is initiated. Combustion then begins, spontaneously, shortly thereafter: the two processes overlap and are, therefore, intimately linked.

The effects of relative air/fuel ratio or equivalence ratio variations on spark-ignition engine combustion, emissions, and performance are discussed more fully in Chaps. 9 , 11, and 15. A brief summary is sufficient here. Mixture requirements are different for full-load (wide-open-throttle, WOT) and for part-load operation. At the former operating condition, full utilization of the *inducted air* to obtain maximum power for a given displaced volume is the critical issue. Where less than the maximum power at a given speed is required, efficient utilization of the *fuel* is the critical issue. At WOT, maximum power for a given volumetric efficiency is obtained with rich-of-stoichiometric mixtures, $\lambda \approx 0.9$, $\phi \approx 1.1$ (see the discussion of the fuel-air cycle results in Sec. 5.5.3). Mixtures that are richer still are at times used to increase volumetric efficiency by increasing the amount of charge cooling

that accompanies fuel vaporization [see Eq. (6.5)], reduces the potential for knock, and holds down exhaust gas temperatures.

At part-load (or part-throttle) operating conditions, there are advantages to diluting the fuel-air mixture, either with excess air or with recycled exhaust gas. This dilution improves the engine's fuel conversion efficiency for three reasons: (1) the expansion stroke work for a given expansion ratio is increased as a result of the change in thermodynamic properties of the burned gases—see Secs. 5.5.3 and 5.7.4; (2) for a given mean effective pressure, the intake pressure increases with increasing dilution, so pumping work decreases—see Fig. 5.10; (3) the heat losses to the walls are reduced because the burned gas temperatures are lower. In the absence of strict engine NO_x emission requirements, excess air is the obvious diluent and at part throttle, engines have historically often operated lean. When tight control of NO_x, HC, and CO emissions is required, operation of the engine with a stoichiometric mixture is advantageous so that a three-way catalyst [b] can be used to clean up the exhaust. The appropriate diluent is then recycled exhaust gases which significantly reduce NO_x emissions from the engine itself. The amount of diluent that the engine will tolerate at any given speed and load depends on the details of the engine's combustion process. Increasing excess air or the amount of recycled exhaust slows down the combustion process and increases its variability from cycle to cycle. A certain minimum combustion repeatability or stability level is required to maintain smooth engine operation. Deterioration in combustion stability therefore limits the amount of dilution an engine can tolerate. As load decreases, less dilution of the *fresh* mixture can be tolerated because the internal dilution of the mixture with residual gas increases (see Sec. 6.4). At idle conditions, the fresh mixture will not usually tolerate any EGR and may need to be stoichiometric or even fuel-rich to obtain adequate combustion stability.

Mixture composition requirements over the engine's load range are illustrated schematically for the two approaches outlined above in Fig. 7.1. If full load torque is to be maximized, and part-load fuel consumption is to be minimized, then ϕ can be varied over the load range as shown. Lower levels of EGR could then be used with a lean NO_x catalyst or trap to realize low NO_x exhaust emissions. If emissions reduction requirements are more demanding and the very efficient three-way catalyst technology is needed, then the engine must be operated with a stoichiometric mixture, and larger

percentages of EGR over the low to mid-low range can be tolerated. When EGR is used, the percentage of recycled exhaust increases from zero at light load to a maximum at mid-load, and then decreases to zero as WOT conditions are approached so air is no longer displaced and maximum bmep can be obtained. Combinations of these strategies are possible.

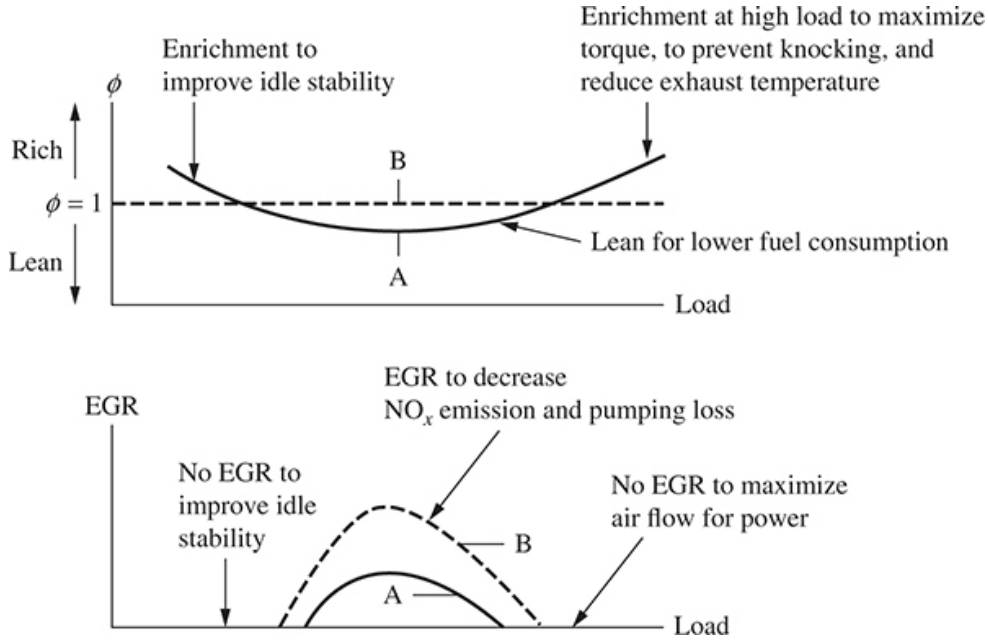


Figure 7.1 Typical mixture requirements for two operating strategies. Top diagram shows the relative air/fuel ratio variation as a function of load. Lower diagram shows recycled exhaust (EGR) schedule. The two strategies are: A, optimized for maximum torque, part-load efficiency, and idle stability; B, optimized for emissions reduction—stoichiometric operation.

In practical spark-ignition engine induction systems, the fuel and air distribution between engine cylinders is not necessarily uniform (and also can vary in each individual cylinder on a cycle-by-cycle basis). The spread in air/fuel ratios between the leanest and richest cylinders (and in EGR levels), over the engine's load and speed range, is now carefully controlled to optimize emissions control in the engine's catalyst system, and avoid excessive combustion variability in the leanest operating cylinder.

Table 7.1 shows the relative amounts (mass percent) of the components air, fuel, residual burned gas, and EGR in the in-cylinder unburned mixture in a gasoline-fueled spark-ignition engine. These proportions, shown at idle, a

typical part-load, and at full load, at low-to-mid engine speed, define the *mixture* we will be discussing in this chapter. The definitions of residual fraction and EGR percentage are given by Eqs. (4.1) and (4.3), respectively. A stoichiometric fuel/air ratio was used for idle and part-load: at WOT a 20% fuel rich mixture was assumed. At the one-quarter load condition 10% EGR was assumed: at mid loads EGR might increase to 15%. The residual (burned gas) fraction is high (some 25%) at idle, and decreases to a low value (about 5%) at WOT for a naturally-aspirated gasoline engine. In a turbocharged spark-ignition engine, the residual gas amounts at part-load and WOT would be about one-third lower because at the same torque levels (with a downsized turbocharged engine) the bmep levels would be higher.

TABLE 7.1 Percentage by mass of in-cylinder unburned mixture components: air, fuel, residual, EGR*

In-cylinder mixture component	Idle Mass %	Part (one-quarter) load Mass %	Full load (WOT) Mass %
Air	70	76.3	88
Gasoline fuel	5	5.2	7.2
Residual gas†	25	9.2	5.0
EGR	0	9.2	0
Total	100	100	100

*Naturally-aspirated gasoline SI engine.
†In a turbocharged engine, the residual gas amounts at the part-load and full-load conditions will be about half these values.

Adjustments to these numbers for different fuels and EGR levels can readily be made using the appropriate equations: Sec. 3.4 and Eqs. (4.1) and (4.3). For example, with ethanol, the stoichiometric fuel/air ratio increases from 0.0655 to 0.111: App. D , Table D.4.

7.2 FUEL METERING OVERVIEW

7.2.1 Mixture Formation Approaches

Historically, carburetors were used to control the fuel flow into spark-ignition engine intake manifolds and distribute the fuel across the air stream.

In a carburetor, the airflow through a converging-diverging nozzle sets up a pressure difference between the carburetor inlet and the throat of the nozzle which depends on the airflow rate. This pressure difference is used to meter the appropriate fuel flow for that airflow. The fuel entered the air stream through the carburetor body, was atomized, and then convected by the air stream past the throttle plate and into the intake manifold. While carburetors developed into complex devices, eventually the more precise fuel-metering requirements for attaining low emissions levels forced their replacement by electronically controlled fuel-injection systems where an electromagnetic fuel injector injects fuel intermittently into the intake manifold or ports. [c]

Several different fuel-injection concepts have been developed and used, starting in about 1980. These are illustrated in Fig. 7.2. The first of these, single-point fuel injection (or central, or throttle-body, fuel injection), is shown in Fig. 7.2 *a*. The fuel is injected upstream of the throttle body, usually with one injector per engine for in-line engines or one per throttle body with V engines. The injector is pulsed (opened) intermittently, synchronized with the operation of each cylinder (usually with ignition system triggering). This approach is the simplest and cheapest of the three shown. Its major disadvantage is that the point of injection is well away from the engine, which, due to the much more extensive fuel wall wetting that occurs, makes precise control of fuel behavior during engine transients more difficult.

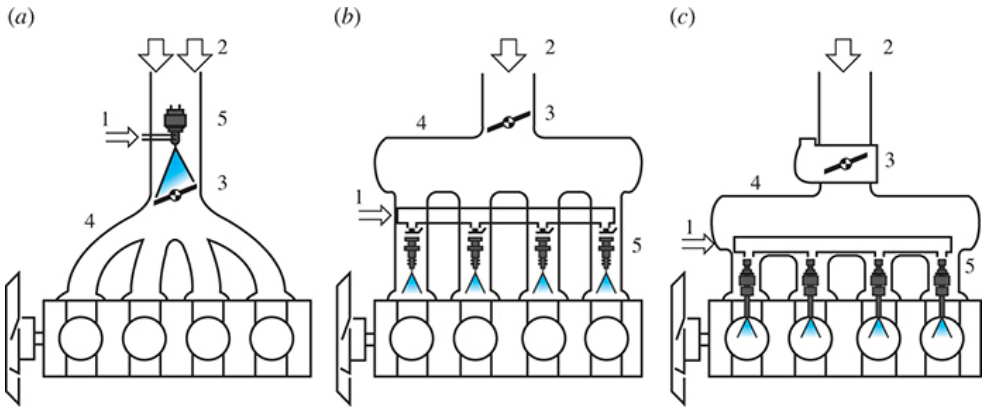


Figure 7.2 Schematics of different fuel-injection system arrangements: (*a*) single point, (*b*) multipoint (port), and (*c*) direct in-cylinder injection (DI). (1) Fuel; (2) Air; (3) Throttle; (4) Intake manifold; (5) Fuel injector. [2] (*Courtesy Robert Bosch GmbH and SAE.*)

Multipoint fuel injection, shown in Fig. 7.2 *b*, has been the dominant approach for the past two decades. Here one injector per cylinder is used, positioned close to the intake manifold intake-port interface, to direct an intermittent fuel spray toward the intake valve. Usually each injector delivers one injection pulse per cylinder per cycle, phased sequentially with each cylinder's gas exchange process. The advantage of the multipoint injection approach is its short distance between point of injection and intake valve, and that the fuel spray can be directed largely toward the hot surface of the valve head. This allows for more precise control of the fuel's metering.

The third approach in Fig. 7.2 *c* is injection of fuel directly into each cylinder of the engine. Gasoline direct-injection (GDI) is being used in automobile SI engines in significant volumes. With GDI, only air (or air with EGR) is inducted into the cylinder. During intake (and/or the compression process) the fuel is injected using an injector located in the cylinder head into the in-cylinder air (and residual gas), to form a well-mixed fuel vapor, air, residual gas (and EGR, if used) mixture. There are two modes of operation. With fuel injection during the intake stroke, an essentially *homogeneous* fuel-air mixture is produced before combustion is initiated. With fuel injection during compression, a stratified in-cylinder fuel-air mixture can be produced, with only part of the air being mixed with the fuel vapor. This *stratified* operation (at part-load) is designed to create a close-to-stoichiometric readily combustible fuel-vapor air "mixture cloud" in contact with the spark plug, the excess air surrounds this "cloud" so that the engine operates fuel lean overall which (like the diesel) results in more efficient engine operation. Stratified direct-injection operation at part-load is usually combined with homogeneous DI operation at higher and full-load. The advantages of direct fuel injection are its inherently faster mixture-preparation control during engine transients (because the fuel is injected right into the cylinder where it will be burned), and the part-load engine efficiency benefits from lean operation when stratified. The primary disadvantage is high fuel-injection system cost.

These three approaches to fuel metering with solenoid actuated fuel injectors will be discussed in more detail in the following sections, 7.3 and 7.4.

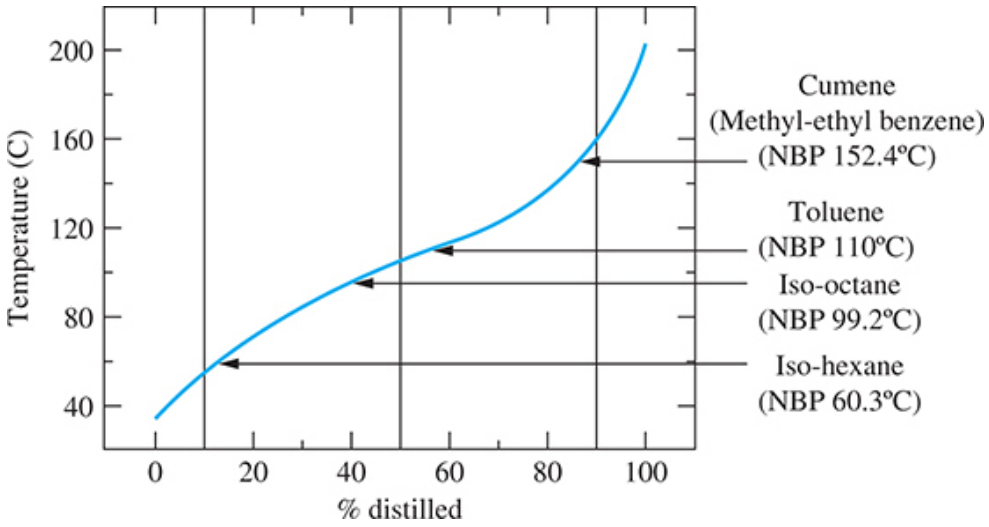


Figure 7.3 Evaporative characteristics of a standard (UTG91) gasoline: equilibrium distillation curve, at atmospheric pressure (ASTM D86).[5] NBP, normal boiling point. T_{10}, T_{50}, T_{90} distillation points are used to calculate the Drivability Index of the fuel; see Eq. (7.2) .

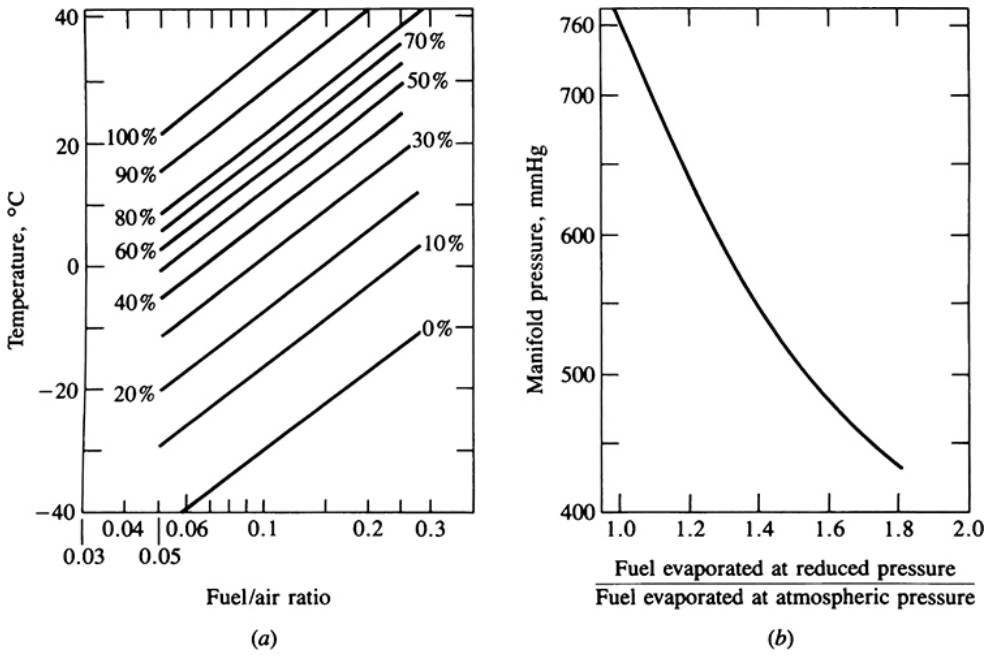


Figure 7.4 (*a*) Percentage of indolene fuel evaporated at equilibrium at 1 atmosphere pressure. (*b*) Effect of pressure on amount of indolene fuel evaporated. [6]

7.2.2 Relevant Characteristics of Fuels

Fuels requirements for internal combustion engines are especially demanding because the engine combustion process is integrated into the power-generating operating cycle. The fuel's characteristics impact the engine's combustion and emissions formation processes, the onset of any abnormal combustion phenomenon such as knock, and especially mixture preparation—the topic of this chapter. Successful mixture preparation requires much more than achieving fast-enough stable combustion under warmed-up engine conditions over the full engine load and speed map. It requires satisfactory engine starting under cold and hot engine conditions. It requires maintaining good combustion and effective emissions control during all engine transients (e.g., during engine warm-up, and vehicle accelerations and decelerations). It requires minimizing the constraint that knocks onset places on engine torque and efficiency. It must also enable meeting regulatory emissions control constraints.

With conventional spark-ignition engine liquid-fuel metering systems, the fuel enters the air stream as a liquid jet. The jet atomizes into droplets that move through the air and start to vaporize. With manifold or port fuel injection, a major fraction of these droplets deposit on the walls of the intake system components. Vaporization of the liquid fuel on these walls then occurs. The flow of liquid fuel along the walls can also be significant. The transport of fuel as vapor, droplets, and liquid streams or films into the cylinder can all be important.

For direct fuel-injection approaches, these fuel spray phenomena occur inside the engine's cylinders, where the surrounding gas is air mixed with the burned residual gases. This in-cylinder charge is thus warmer than the intake air temperature so fuel vaporization occurs at a faster rate, and the wetting of the combustion chamber walls by the spray droplets is thereby reduced. Note also that evaporation of fuel drops through heat transfer with the in-cylinder charge cools the charge and drops its temperature. Evaporation of liquid fuel off the intake port and valve surfaces mostly cools the walls, and not the airflow. This evaporative cooling of the charge with direct-injection has a significant effect on the subsequent temperatures of the unburned mixture as it is compressed: much more so for the alcohols (e.g., E85) than for gasoline due to their higher heat of vaporization per unit chemical energy than gasoline.

The predominant fuel for spark-ignition engines is gasoline (petrol) refined from petroleum (see Sec. 1.11.1). Another liquid fuel in common use is ethanol (ethyl alcohol), either blended with gasoline (e.g., E10, a 10% blend) or as a stand-alone fuel (usually as E85, 85% ethanol and 15% gasoline, by volume). Other fuels in use are LPG, which is a mixture of propane (C_3H_8) and butane (C_4H_{10}), and natural gas, which is predominantly methane (CH_4). LPG is stored in pressurized tanks as a liquid at ambient temperatures and about 500 kPa (80 psia): it vaporizes in the intake system of spark-ignition engines and enters the cylinder as a gas, mixed with air. Natural gas on vehicles is usually stored as compressed gas in high-pressure tanks at up to 7 Mbar (10,000 psi). It can, with difficulty, be stored cryogenically. Storage of NG on the vehicle is expensive and significantly raises the cost of this fuel option. Natural gas is used in large stationary engines to generate power (e.g., for pumping the gas through pipelines). Other spark-ignition engine fuels are the gasoline-equivalent liquid hydrocarbon blends made from oil sands and heavy oils. Also, two synthetic fuels—hydrogen and methanol could be used in SI engines.

The *volatility* of a liquid fuel—its potential for vaporizing at a given temperature and pressure—is the key mixture preparation characteristic. Gasoline (petrol), the standard SI engine fuel, is a mixture of a large number of individual hydrocarbons: it thus has a boiling temperature range rather than a single boiling point. Typically, this range is 30 to 200°C. Individual hydrocarbons have the saturation pressure-temperature relationships of a pure substance. The lower the molecular weight, the higher will be the saturated vapor pressure at a given temperature. The boiling point of hydrocarbons depends primarily on their molecular weight: the vapor pressure also depends on molecular structure. The equilibrium state of a hydrocarbon-air mixture depends therefore on the vapor pressure of the hydrocarbon at the given temperature, the relative amounts of the hydrocarbon and air, and the total pressure of the mixture. The equilibrium fraction of fuel evaporated at a given temperature and pressure can be calculated from Bridgeman charts [3] and the distillation characteristics of the fuel (defined by the ASTM distillation curve [4]). The (equilibrium) distillation curve of a standard test gasoline (UTG91) is shown in Fig. 7.3. This U.S. regular (91 RON) gasoline, *at atmospheric pressure*, has a boiling point range of about 30 to 200°C. The boiling points of several gasoline

components are shown. Note that these temperatures do not denote the engine air (or wall surface) temperatures required to vaporize the fuel components: the fuel vapor pressure in the engine mixture is well below 1 bar. For a stoichiometric gasoline vapor-air mixture at 1 bar, the fuel vapor partial pressure is about 0.02 bar. Thus the fuel components vaporize at much lower temperatures than the atmospheric pressure distillation curve in Fig. 7.3 indicates. Figure 7.4 *a* shows the percentage of a standard gasoline (indolene) evaporated at equilibrium at 1 bar *mixture* pressure as a function of fuel/air ratio: the stoichiometric gasoline fuel/air ratio is 0.0685. Figure 7.4 *b* shows the effect of reducing the intake manifold pressure below 1 bar (760 mmHg) on the amount of indolene fuel evaporated. While insufficient time is usually available in the manifold to establish equilibrium, the trends shown are indicative of what happens in practice: lower pressures increase the relative amount of fuel vaporized significantly.

Under warmed-up engine conditions, the vaporization of gasoline fuel is not a significant issue. Under cold-starting conditions, with stringent hydrocarbon emissions requirements, achieving a rapid engine start-up with stable combustion is more challenging: use of fuel-rich vaporized fuel-air mixtures which burn faster significantly increases engine HC emissions.

Additional tests and parameters are used to define a fuel's volatility: (gasoline, or blends with ethanol).

The *Reid vapor pressure* (RVP) test is used to determine the vapor pressure of gasolines and other refinery products having a vapor pressure below atmospheric. The test apparatus has two chambers. One is filled with a liquid fuel sample at 0 to 4°C (32 to 40°F); the other, four times the size of the liquid-containing volume and fitted with a pressure gauge, filled with atmospheric air, is rapidly connected to the fuel chamber. The assembly is then immersed in an electrically heated water bath at 37.8°C (100°F). After 5 minutes, the assembly is shaken and the pressure gauge is read. The immersion is repeated for another 2 minutes; the assembly is removed, shaken again, and the pressure read again. The procedure is repeated until two successive readings within 0.35 kPa (0.05 psi) are recorded. Typical Reid vapor pressures for gasolines are about 0.5 bar (7 psi).

The RVP of a fuel (with higher RVP corresponding to a more volatile fuel) is a measure of how quickly and smoothly a cold engine will start and warm up. Also, air pollution regulatory agencies use RVP to limit gasoline vapor

pressure to control the fuel system evaporative emissions of smog forming hydrocarbons.

A *Drivability Index* (DI) is used (in the United States) to characterize the impact of a gasoline's volatility on SI engine in-vehicle drivability. Such drivability aspects are ease of engine starting, warm-up driving and acceleration transients; and with respect to vehicle driving, smoothness, hesitations, and surges. The DI is defined from the fuel's distillation curve, Fig. 7.3, using the T_{10}, T_{50}, and T_{90} temperatures; the temperature for a specific fuel at which 10, 50, and 90% of the fuel is vaporized, respectively. The formula for DI is:

$$\mathrm{DI} = 1.5\,T_{10} + 3\,T_{50} + T_{90} \tag{7.2}$$

with the temperatures in °F. T_{10} describes a fuel's ability to vaporize quickly and enable cold starting. T_{90} is a measure of the high boiling point gasoline components which impact fuel economy—the higher molecular weight hydrocarbons have a higher heating value. T_{50} defines the fuel's average behavior. The DI of a typical gasoline is between 1100 and 1175. It varies seasonally to a maximum of some 1250 in areas with hot summers. Blending modest amounts (≤10%) of ethanol with gasoline requires raising DI values to about 1300.

The composition of the fuel-vapor air mixture in spark-ignition engines operating with a homogeneous charge is constrained by several requirements: fast enough combustion (over the full operating map), emissions control constraints, complete air (oxygen) utilization at full load, and high engine efficiency. For homogeneous charge operation, a stoichiometric mixture satisfies these requirements effectively: a choice reinforced by the widely used three-way catalyst technology, which needs a stoichiometric composition exhaust gas stream to achieve its high effectiveness. (Note that even in stratified direct-fuel-injection SI engines, combustion of the fuel-air mixture locally, at close-to-stoichiometric composition, is preferred.)

The stoichiometric mixture air/fuel ratio depends on the molecular composition of the fuel, as discussed in Sec. 3.4 . Here we focus on gasolines, ethanol, and blends of ethanol with gasoline, the most commonly used SI engine fuels. For stoichiometric operation, the amount of air required

per unit injected fuel is given by the $(A/F)_s$: one can also work with its inverse, $(F/A)_s$. Figure 7.5 shows the stoichiometric (A/F) and (F/A) as a function of the hydrocarbon's molar H:C ratio. The average value for gasoline is close to 14.6, or 0.0685 for $(F/A)_s$. The approximate range in H:C ratio is indicated. These numbers are for hydrocarbons containing carbon and hydrogen only, for both individual compounds and blends or mixtures.

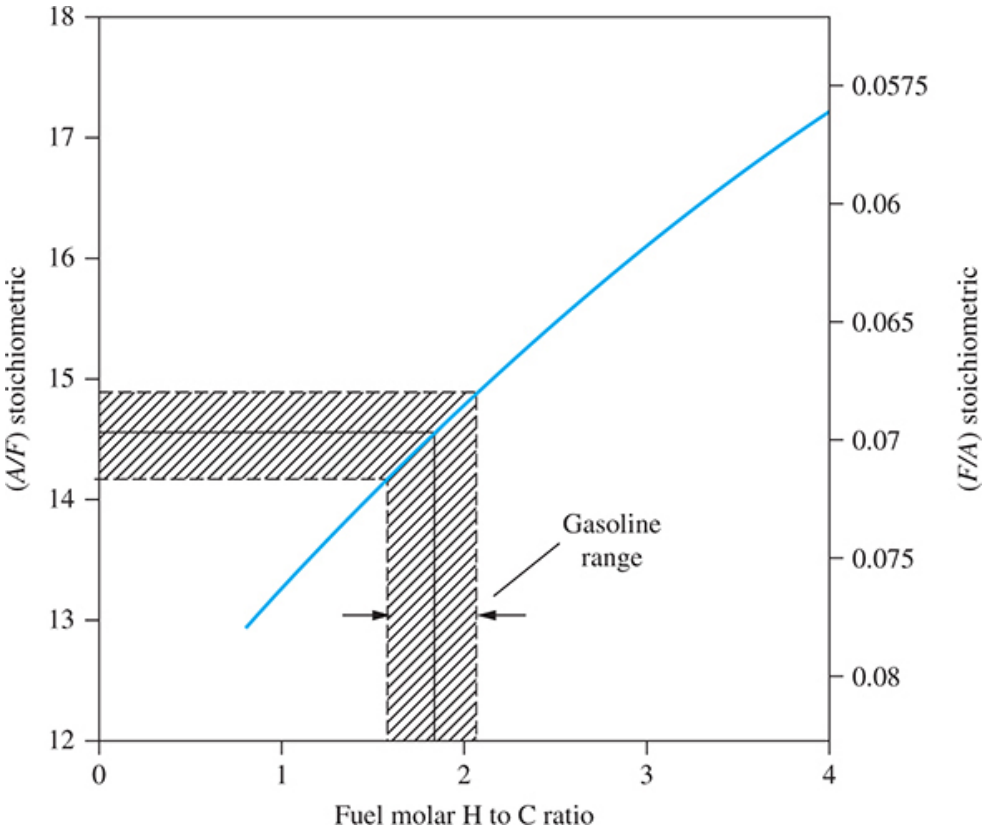


Figure 7.5 Stoichiometric air/fuel ratio for air-hydrocarbon fuel mixtures as a function of the fuel's molar H/C ratio.

Figure 7.6 shows these stoichiometric ratios for gasoline-ethanol mixtures (also for methanol). Note that the addition of oxygen to the carbon and hydrocarbon combustibles changes the mass of fuel relative to the mass of air in a stoichiometric mixture significantly. The molar proportion of oxygen in methanol is a factor of 2 higher than in ethanol. The curves for methanol blends with gasoline closely parallel those for the ethanol blends, and are displaced downward.

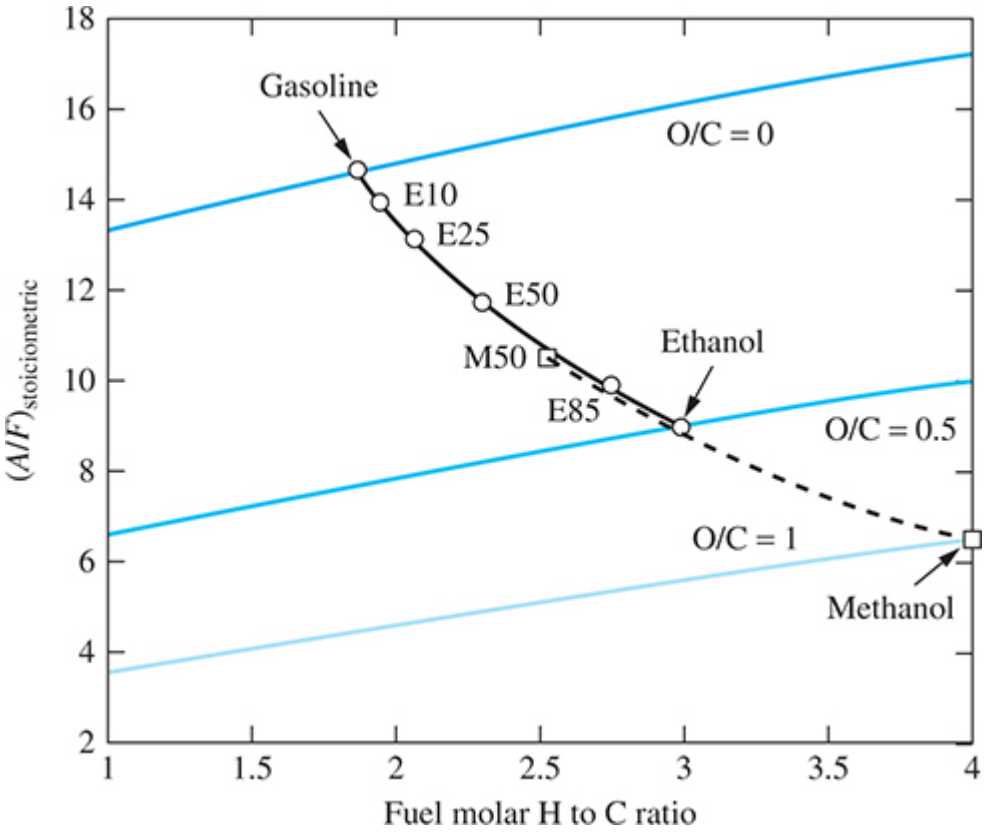


Figure 7.6 Stoichiometric air/fuel ratio of mixtures of air with hydrocarbon-alcohol fuel blends, as a function of fuel's molar H/C ratio and O/C ratio. (E10 means a blend of 10% ethanol with 90% gasoline, by volume.)

Note that for these significant changes in the stoichiometric mass of fuel per unit mass of air, the torque (and bmep) of a given engine at a given intake manifold pressure changes little. This is because an internal combustion engine is a device with given size cylinders, and the chemical energy content of a stoichiometric fuel-air mixture, per unit mass or per unit volume of mixture, is almost independent of the fuel composition: see the lower heating values of such stoichiometric fuel-air mixtures (in MJ/kg) in Table D.4, App. D . When expressed per unit volume of stoichiometric mixture, the hydrogen-air mixture value lines up more closely than the MJ/kg value. The amount of oxygen available in the air for carbon and hydrogen oxidation is the key factor in this common value of mixture chemical energy density.

Natural gas and LPG are also used as spark-ignition engine fuels. They, of course, are gases at ambient pressure and temperature. Their composition

varies depending on the source of these fuels and their processing. This variation in molecular (and thus elemental) composition affects the fuel's stoichiometric air-fuel ratio and heating value, modestly. With NG the antiknock octane rating can be affected more significantly. Also, with NG, the presence of small (and varying) percentages of nitrogen and carbon dioxide dilute the fuel's hydrocarbons of which methane CH_4 dominates (see typical NG composition in Table D.4). Note that in close-to-stoichiometric fuel-vapor air mixtures, these smaller fuel molecules CH_4, C_3H_8, also ethanol C_2H_6O, and methanol CH_4O, displace a significant fraction of the air. The partial pressure of air in an atmosphere pressure natural-gas fuel-air mixture is some 10% lower than with gasoline (see Fig. 6.3).

Hydrogen has been explored as an SI engine fuel. Combustion of hydrogen produces no CO_2, a greenhouse gas. However, hydrogen is really an *energy carrier* rather than a fuel since it must be manufactured. The standard production process for hydrogen is steam reforming of NG (where the carbon is fully oxidized to CO_2 and removed, and the water decomposed in the process provides additional hydrogen). Flame propagation in premixed hydrogen-air mixtures is extremely rapid—much faster than in equivalent stoichiometry hydrocarbon-air mixtures. Thus hydrogen-fueled SI engines usually operate with well lean of stoichiometric mixtures, and are boosted to obtain acceptable bmep levels and burn rates.

7.3 CENTRAL (THROTTLE-BODY) FUEL INJECTION

Central or throttle-body fuel-injection systems, where one or two electronically controlled injectors meter the fuel into the airflow directly above the throttle body, can be used: see Fig. 7.2. They provide straightforward electronic control of fuel metering at lower cost than multipoint port injection systems. However, the problems associated with slower transport of fuel than the air from upstream of the throttle plate to the cylinder must now be overcome. Figure 7.7 shows a drawing of one such system: the Bosch Mono-Jetronic low-pressure single-point injection system for four-cylinder engines. An injector, placed above the throttle plate, meters

the fuel in response to calibrations of airflow, measured or based on intake manifold pressure, air temperature, engine speed, and other signals. The fuel spray directs the fuel toward the crescent shaped opening between the throttle plate and its housing. The airflow provides effective atomization of the liquid fuel, minimizing liquid fuel deposition on the manifold walls. The high air velocity created by the pressure drop across the throttle opening shears and atomizes the liquid sheet on the throttle plate. Vigorous mixing of fuel and air then occurs, especially at part throttle, and provides good mixture uniformity and distribution between cylinders. Injector fuel delivery is synchronized with the ignition pulses. The injection duration is controlled with an exhaust gas oxygen sensor (see Sec. 7.8) to provide a stoichiometric mixture to each cylinder. The injection duration is extended to provide additional fuel during cold starts and engine warm-up, and provide injection enrichment during accelerations. The throttle valve potentiometer recognizes changes in throttle valve position. During accelerations, additional fuel beyond the steady-state requirement is injected to offset delays in fuel reaching each open intake valve to maintain the desired in-cylinder stoichiometry. As WOT is approached, mixture enrichment can be provided to control knock and maximize torque. The injector operates with a fuel supply pressure of about 2 bar (absolute). [2]

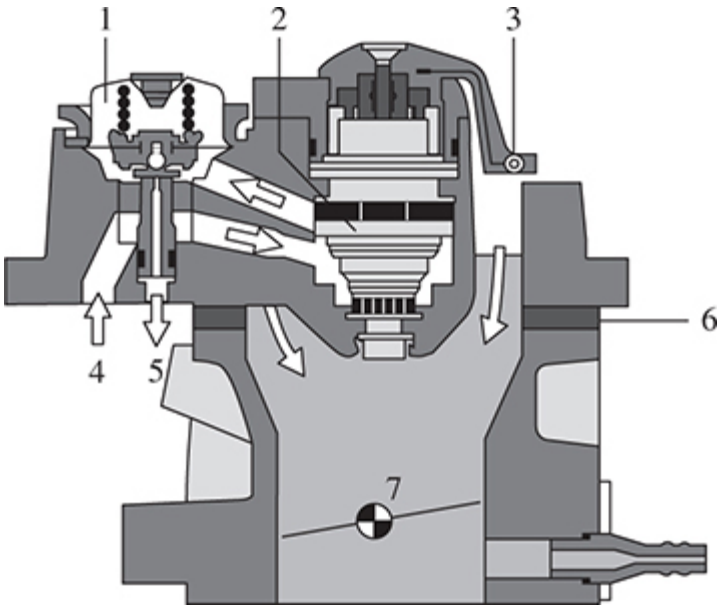


Figure 7.7 Schematic of central fuel injection system, upstream of the throttle: (1) Pressure regulator; (2) Injector; (3) Air temperature sensor; (4)

Fuel inlet; (5) Fuel return; (6) Insulator plate; (7) Throttle. Bosch Mono-Jetronic unit (*Courtesy Robert Bosch GmbH and SAE*). [2]

7.4 PORT (MULTIPOINT) FUEL INJECTION

From about 1980, over a decade or so, a steady transition occurred from the use of carburetors to meter the gasoline flow in gasoline spark-ignition engines, to use of port fuel injection where the fuel is injected into the intake port of each engine cylinder. Such systems require one fuel injector per cylinder. While there are both mechanical and electromagnetically controlled injection systems, both are electronically controlled. The advantages of port fuel injection are increased power and torque through improved volumetric efficiency and more uniform fuel distribution, more rapid engine response to changes in throttle position, and more precise control of the in-cylinder air-fuel ratio. Electronically controlled fuel injection allows the amount of fuel injected per cycle for each cylinder to be varied in response to inputs derived from sensors, which define actual engine operating conditions. Two basic approaches have been developed; the major difference between the two is the method used to determine the airflow rate: direct measurement, or estimation based on air density and engine speed.

7.4.1 System Layout, Components, and Function

Figure 7.8 shows a schematic of a port-fuel-injection system where the airflow is measured. The electrically driven fuel pump delivers the fuel through a filter to the fuel line. A pressure regulator maintains the pressure in the line at a fixed value (e.g., 270 kN/m ^{2}, 39 psi, usually relative to manifold pressure to maintain a constant fuel pressure drop across the injectors). Branch lines lead to each injector. Depending on the fuel supply approach (see Sec. 7.9) the excess fuel may be returned to the tank. The inducted air flows through the air filter, past the airflow meter, then the throttle to the intake manifold plenum volume. Separate runners (and branches) lead to each inlet port and engine cylinder. An electromagnetically actuated fuel-injection valve (see Fig. 7.9) is located either in the downstream end of each intake manifold runner or the entrance to the intake port. The major

components of the injector are the valve housing, the injector needle and its pintle, the armature to which the needle is connected, the helical spring, and the solenoid coil. When the solenoid is not excited, the solenoid armature is forced, with its seal, against the valve seat by the helical spring and closes the fuel nozzle. When the solenoid coil is excited, the armature is attracted and lifts the needle about 0.15 mm so that fuel can flow through the calibrated annular passage around the valve stem. The front end of the injector needle is shaped as a pintle with a ground top to atomize the injected fuel. The relatively narrow spray cone of the injector minimizes the wetting of the intake port wall with liquid fuel. The mass of fuel injected per injection is controlled by varying the duration of the current pulse that excites the solenoid coil. In a 2-liter four-cylinder engine at typical part-load conditions, it is about 10 mg of fuel; at WOT it is about 40 mg. Typical injection times for automobile applications range from about 1.5 to 10 ms. [2] Many different designs of port fuel injectors are available ranging from single hole to two or more multi-hole nozzles. Spray targeting, an important injection system characteristic, is discussed below. Droplet sizes for port fuel injector sprays (with injection pressures of 3 to 7 bar) are in the 80 to 200 μm diameter range.

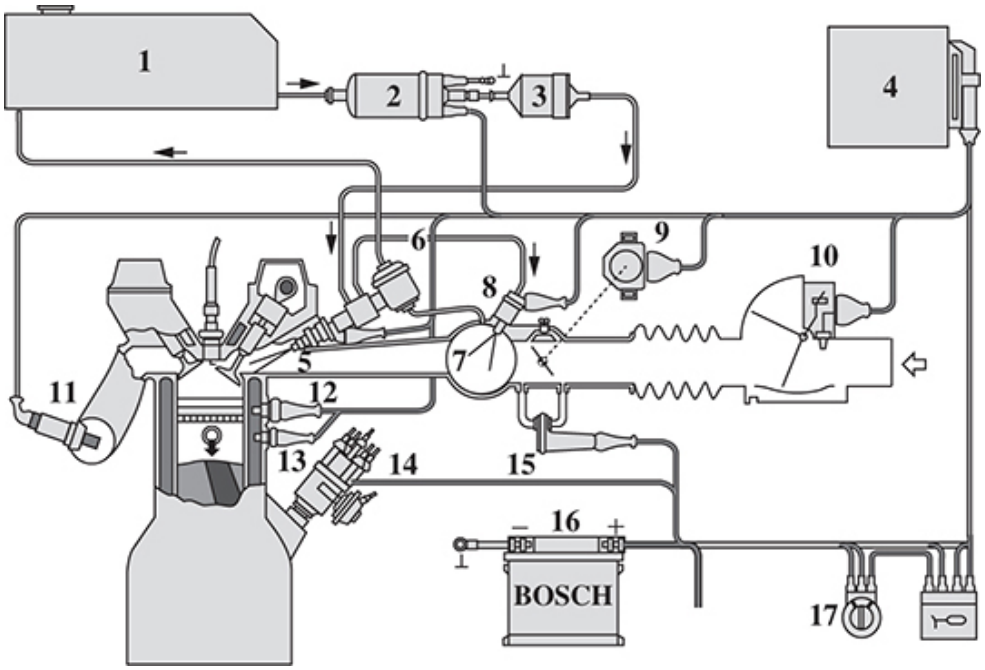


Figure 7.8 Multipoint port fuel-injection system: schematic of Bosch L-Jetronic System: (1) Fuel tank; (2) Electric fuel pump; (3) Fuel filter; (4)

Electronic control unit; (5) Injector; (6) Fuel pressure regulator; (7) Intake manifold; (8) Electric cold-start valve; (9) Throttle valve switch; (10) Airflow sensor; (11) Exhaust oxygen sensor; (12) Thermal-time switch; (13) Engine-temperature sensor; (14) Ignition distributor; (15) Auxiliary-air valve; (16) Battery; (17) Ignition switch. (*Courtesy: Robert Bosch GmbH and SAE.* [2])

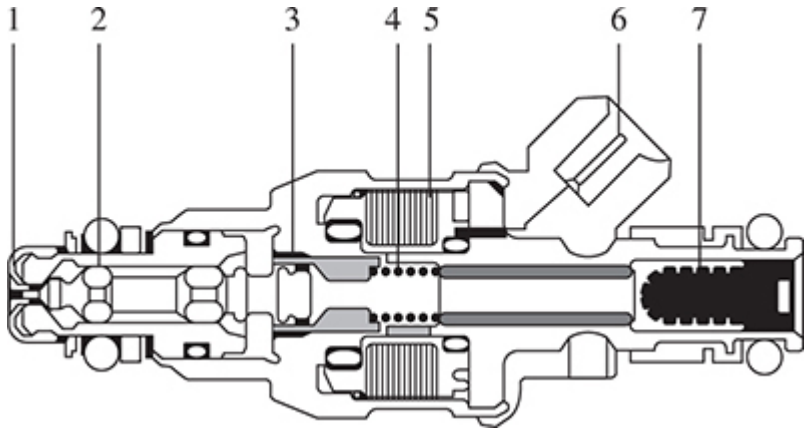


Figure 7.9 Cross-section of port fuel injector: (1) Pintle; (2) Valve needle; (3) Armature; (4) Spring; (5) Solenoid winding; (6) Electrical terminals; (7) Fuel strainer. (*Courtesy: Robert Bosch GmbH and SAE.* [2])

The multipoint (port) fuel injection system shown in Fig. 7.8 measures the airflow directly. The airflow meter is placed upstream of the throttle, just downstream of the air filter. Several approaches to airflow measurement have been used. One approach is that shown in Fig. 7.8, where the air flow deflects a sensor plate against a constant return force spring. [2] Another is the hot-film air mass flow meter shown in Fig. 7.10. The advantages of direct air-flow measurement are: [2] (1) automatic compensation for tolerances, combustion chamber deposit buildup, wear and changes in valve adjustments; (2) the dependence of volumetric efficiency on speed and exhaust backpressure is automatically accounted for; (3) less acceleration enrichment is required because the air-flow signal precedes the filling of the cylinders; (4) improved idling stability; and (5) lack of sensitivity of the system to EGR since only the fresh air flow is measured.

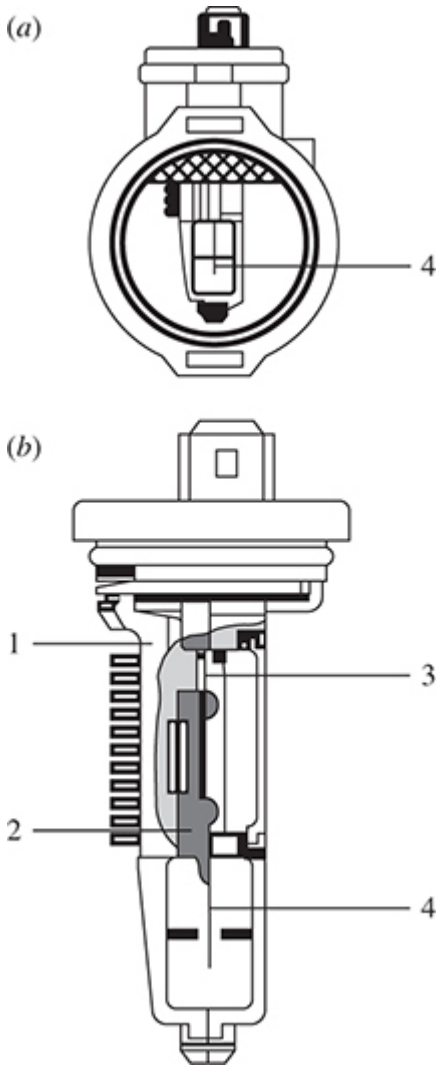


Figure 7.10 Hot-film air mass-flow meter: (*a*) Housing; (*b*) Hot-film sensor within the housing: (1) Heat sink; (2) Spacer; (3) Driver stage; (4) Metallic film sensor element.[2] (*Courtesy Robert Bosch GmbH and SAE.*)

The mass of air inducted per cycle to each cylinder, m_a, varies as

$$m_a \alpha \frac{\dot{m}_a}{N} \tag{7.3}$$

Thus the primary signals for the electronic control unit (ECU) are airflow rate and engine speed. The injector pulse width, which at fixed Δp across the injector determines the amount of fuel injected, is inversely proportional to engine speed and directly proportional to air flow. The engine block

temperature sensor, starter switch, and throttle valve switch provide input signals for the necessary adjustments for cold-start, warm-up, idling, and WOT enrichment (when used).

For cold-start enrichment, one (or more) cold-start injector valve may be used to inject additional fuel into the intake manifold plenum as shown in Fig. 7.8. Since short opening and closing times are not important, this valve can be designed to provide extremely fine atomization of the fuel to minimize the enrichment required.

An alternative to direct airflow measurement is air mass estimation based on other sensor signals. The appropriate coil excitation pulse duration or width is set by the ECU. In the speed-density system, the primary inputs to the engine's ECU are the outputs from the manifold pressure sensor, the engine speed sensor (usually integral with the distributor), and the temperature sensors installed in the intake manifold to monitor air temperature and engine block to monitor the water-jacket temperature—the latter being used to indicate fuel-enrichment requirements during cold-start and warm-up. For warm-engine operation, the mass of air per cylinder per cycle m_a is given by

$$m_a = \eta_v(N)\rho_a(T_i, p_i)V_d = \frac{\eta_v V_d p_i}{R_a T_i} \tag{7.4}$$

where η_v is the volumetric efficiency, N is engine speed, ρ_a is the inlet air density, and V_d is the displaced volume per cylinder. The ECU then determines the pulse width, which excites the injector solenoids, which provides the desired amount of injected fuel. The pulse width depends primarily on the manifold pressure; it also depends on the variation in volumetric efficiency η_v with speed N and variations in air density due to variations in air temperature. The control unit also initiates mixture enrichment during cold-engine operation and during accelerations that are detected by the throttle sensor.

Port fuel-injection systems offer several options regarding the timing and location of each injection relative to the intake event. The spray is directed toward the back of the intake valves. With electronically controlled injection systems, the fuel is injected intermittently toward the intake valves. The fuel-injection pulse width to provide the appropriate mass of fuel for each

cylinder air charge varies from about 1.5 to 10 ms over the engine load and speed range. In crank angle degrees this varies from 10 to 20° at light load and low speed to up to 300° at maximum speed and load. Thus the pulse width varies from being much less than to greater than the duration of the intake stroke. In the early fuel injection systems, all injectors were operated simultaneously. Now, to achieve mixture uniformity sequential injection timing is used, where the phasing of *each* injection pulse relative to its cylinder's intake valve lift profile is the same. Figure 7.11 shows the fuel injection timing for each cylinder of a four-cylinder engine, at mid-load and speed. Fuel injection into the intake port toward the back of the (closed) intake valve ends prior to the start of the intake stroke. This timing, with injection toward a closed intake valve, results in most of the injected fuel impinging on the valve head and stem, and on the port walls close to the valve seat. More of the fuel then vaporizes before entry into the cylinder than would be the case with open (intake) valve injection. Also, injection toward the end of the valve closed period allows engine transients to be followed better: the time lag in the information the ECU uses to calculate the amount of fuel to be injected is then at its minimum. Note, some fuel metering strategies use open valve injection during engine starting as a way to get more of the injected fuel, rapidly, into the cylinder. Engine performance and emissions do change as the timing of the start of injection relative to inlet valve opening is varied. Injection toward an open intake valve usually results in higher HC emissions—more liquid fuel droplets pass directly into the cylinder. Injection with valve lift approaching its maximum, or decreasing, is least desirable because the air flow velocities through the valve open area are becoming lower, and atomization vaporization and mixing are slower (see Sec. 6.3).

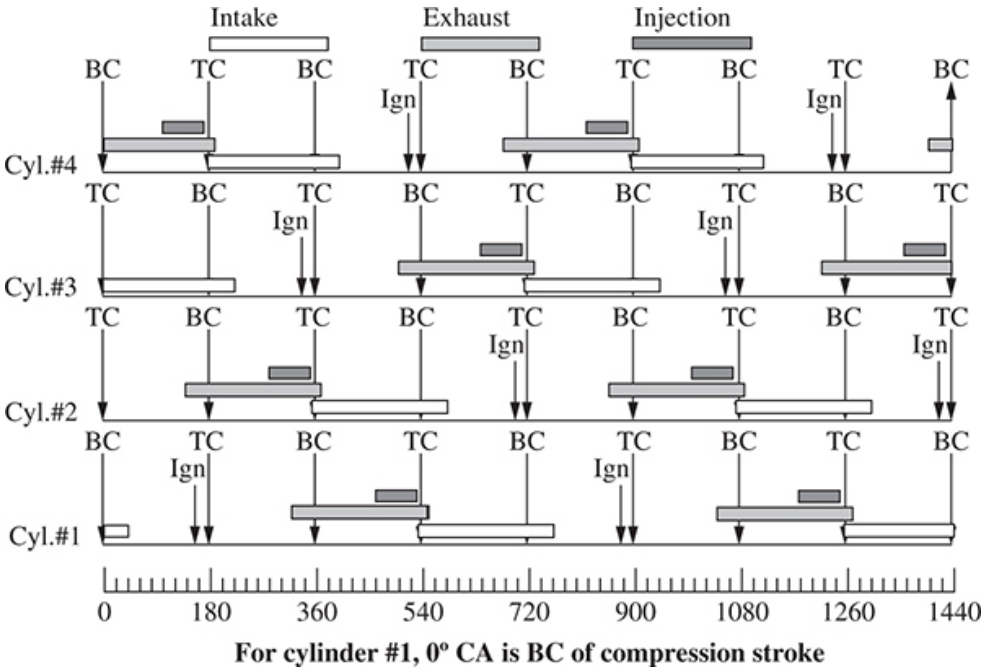


Figure 7.11 Event diagram showing the intake and exhaust processes, and fuel injection pulse, and ignition for the individual cylinders of a four-stroke cycle four-cylinder engine.

7.4.2 Fuel Spray Behavior

The overall objective of the geometric layout of a port-fuel-injection system is to place most of the liquid fuel on the intake valve surface (the hottest surface available) near the valve seat toward the end of the valve closed period. Figure 7.12 shows the preferred geometry with the injector close enough to the intake valve head so that, with realistic spray spreading angles, the amount of port wall wetting is minimized. Since the intake valve head in a warmed-up engine reaches a temperature in the 150 to 200°C range, it is the surface that will most effectively evaporate liquid fuel. In most cylinder head designs, it proves infeasible to locate the injector that close to the valve, and use that straight a port design. Thus significant wetting of the intake port wall occurs. The port walls are at a temperature close to that of the engine coolant.

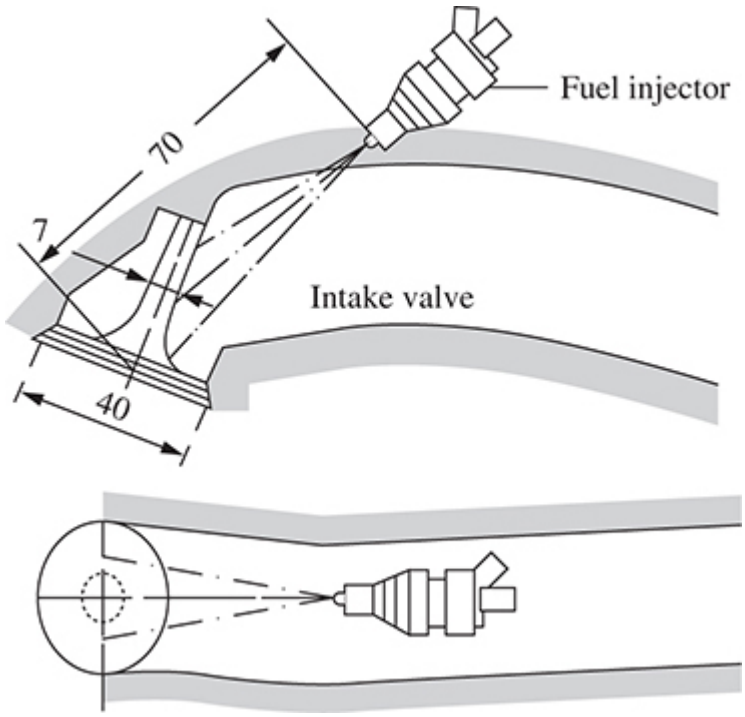


Figure 7.12 Preferred geometric targeting arrangement of fuel injector and intake valve head. Dimensions in mm.

Figure 7.13 illustrates additional spray targeting complexities in injector location and intake port and valve design. Shown are various potential intake manifold, port and valve layouts, of multi-valve cylinder heads for four-valve per cylinder (two intake valves) engines. With the simplest layout (Fig. 7.13*a*), with two intake valves and one fuel injector per cylinder, the spray cannot fully target each intake valve head, and substantial wetting of the wall dividing the two intake ports is inevitable. The other schematics add additional issues. In some cylinder head designs, port throttles are incorporated as shown to close off one of the intake ports at part load to generate a swirling flow within the cylinder (see Sec. 8.3) which speeds up the engine's combustion process which improves the engine's operating smoothness, and its NO_x emissions control potential. The increasing complexity and resulting wall-wetting issues of (*b*) and (*c*) are clear: the cost of these increasingly complex approaches increases (*a*)→(*b*)→(*c*)→(*d*).

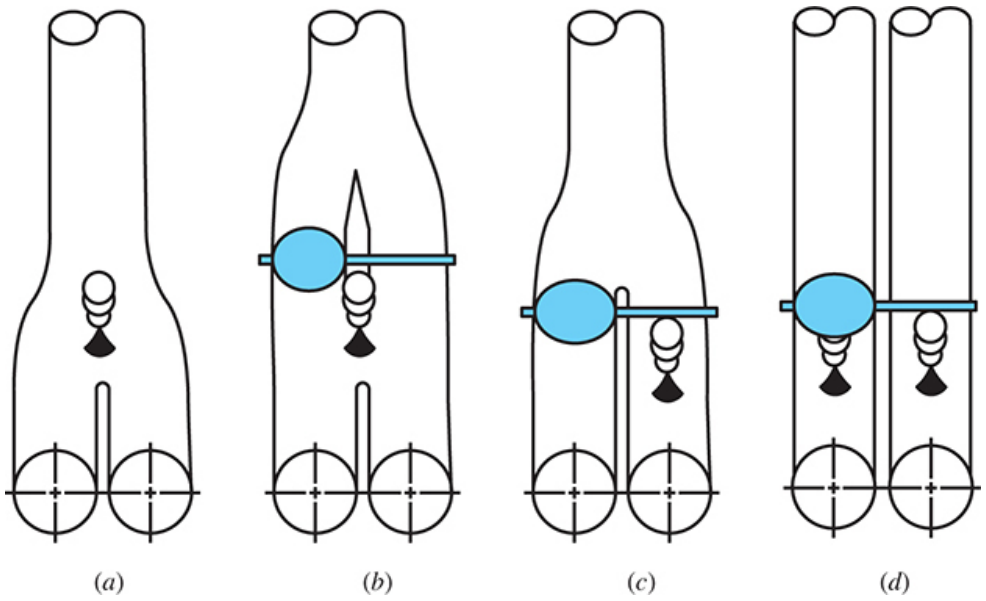


Figure 7.13 Four-valve-per-cylinder engine intake runner, port and port-throttling arrangements, illustrating fuel-injector spray targeting challenges.

Many different designs of fuel injectors for multipoint injection systems intended to improve mixture preparation have been developed and used. Spray geometry and droplet size distribution are key injector parameters. Figure 7.14 shows two examples of photographs of port fuel-injector sprays: (a) a single hole spray with a 30° spreading angle, once the liquid column exiting the injector nozzle breaks up into droplets; and (b) a dual spray injector which would fit with the two intake valve with short separating wall geometry shown in Fig. 7.13*a*. A single hole, two, four, and up to 12-hole nozzle designs, which produce the appropriate number of sprays of different spreading angles—narrower (~10°) and broader (~ 25°)—have been tried. Injection pressures are typically in the range of 3 to 7 bar. The droplet size distribution in injection designs is in the 80 to 200 μm range. For the droplets to follow the air flow around bends, as needs to occur as the intake flow goes past the valve seat and around the valve head into the cylinder, they must have a diameter close to or less than about 20 μm: thus most of the droplets in standard sprays impinge on the port and valve surfaces. To achieve these small average drop sizes in the fuel spray requires different spray generating mechanisms: one example is air-assist atomization where pressurized air flows through the injector and, in the injector nozzle, shatters the fuel into these much smaller drop sizes.

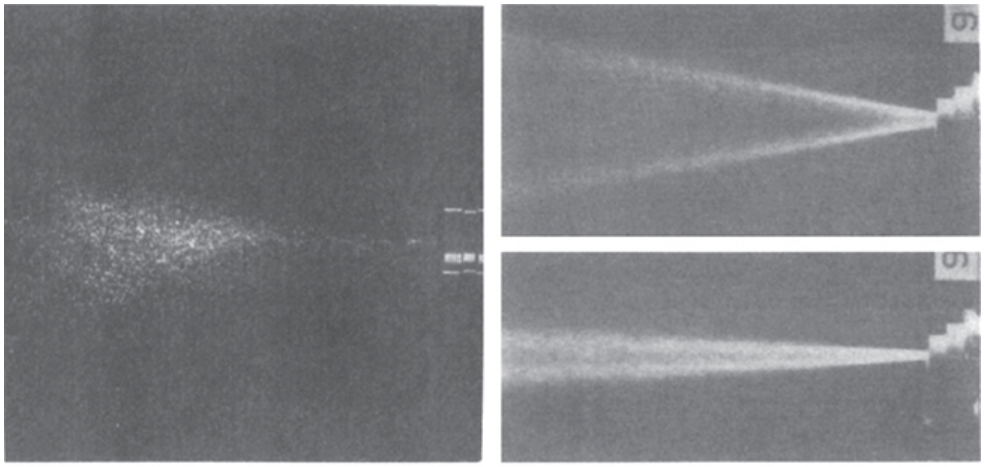


Figure 7.14 (*a*) Short time-exposure photograph of liquid spray from Bosch-type port fuel-injector like that shown in Fig. 7.9. (*Courtesy: Robert Bosch*.) (*b*) Twin sprays from a two-stream port fuel-injector designed for four-valve-per-cylinder engines. [8]

As mentioned several times above, the valve and port surface temperatures have a major impact on the liquid fuel vaporization process from these surfaces. The temperatures of these surfaces start at ambient levels in a cold engine, and steadily rise during warm up. Figure 7.15 shows how the thermal environment in the intake port evolves following start-up at light load and low speed. Once the engine starts firing, heat is transferred from the hot in-cylinder burned gases to the combustion chamber (and exhaust port) surfaces. This thermal energy is then transferred to the coolant. The intake valve-head temperature rises rapidly during the first minute or so of operation: the coolant (and thus intake port wall) temperature rises more slowly. After about 10 minutes of operation, these temperatures stabilize with the coolant at about 90°C, the port slightly higher (due to backflow of hot residual gases into the intake; see below), and the intake valve between about 150 and 200°C. The eventual valve temperature will depend on engine load and speed.

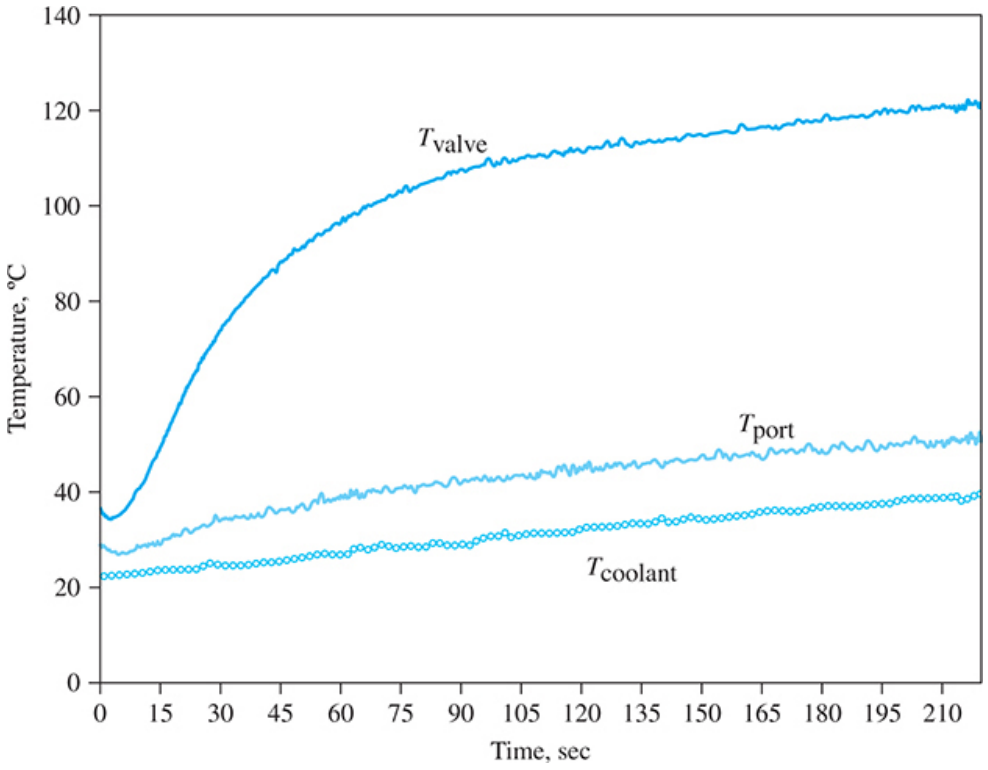


Figure 7.15 Intake port and valve surface, and coolant temperatures, as a function of time following an ambient temperature engine start.[9]

7.4.3 Reverse Flow Impacts

An important phenomenon in mixture preparation with port fuel-injection systems is the reverse blowdown of hot residual burned gases from the cylinder into the intake port at lighter loads and low to moderate speeds. In light-duty vehicle applications, much engine operation is with intake manifold pressures (in naturally-aspirated gasoline engines) of about 0.5 bar or lower. Under these conditions, the ratio of the intake manifold pressure to the exhaust pressure is less than the critical ratio of about 0.5 (see App. C) so the burned gas velocity in this reverse blowdown process, past the valve seat and head, is close to sonic. Strong jets emanate from this steadily opening annulus between the valve-head and valve-seat, flowing into the intake port. These jets disrupt, blast backward, and atomize the liquid fuel that has accumulated in an annular ring around the intake port side of the intake valve seat. This atomizing backflow vaporizes some of this fuel: it also redistributes liquid fuel over a larger port surface area; see Ref. 10.

The relationship between this backflow, the exhaust, cylinder and intake pressures, and the valve events, under these partly loaded conditions, is as follows. Around TC, from intake valve opening (IVO) to exhaust valve closing (EVC), burned gases flow from the exhaust port into the cylinder because $p_e > p_c$: from IVO to the point where $p_c = p_i$ (the cylinder and intake pressures equalize), burned gas flows from the cylinder back into the intake port. At this point, the flow in the intake reverses direction (backward to forward), pulled by the descending piston during the intake stroke. These burned gases in the intake (which have mixed vigorously with the fresh air there) flow back into the cylinder along with the fresh air.

Also, at lower engine speeds, after BC of the intake stroke, there is a reverse (backward) displacement flow as the piston moves up the compression stroke prior to IVC. This reverse displacement flow pushes air, fuel vapor, and burned residual gases (mixed together) back into the intake. At higher engine speeds, the intake-air-flow's momentum toward the still-open intake valve overcomes this upward piston motion, and this reverse flow does not occur (see Sec. 6.2.4).

7.5 AIR FLOW PHENOMENA

7.5.1 Flow Past the Throttle Plate

Except at or close to WOT, the throttle assembly provides the minimum flow area in the entire intake system. Under typical road-load conditions with a standard SI engine, more than 90% of the total pressure loss occurs across the throttle plate. The minimum-to-maximum flow area ratio is large—typically of order 100. Throttle plate geometry and parameters are illustrated in Fig. 7.16. A throttle plate of conventional design as shown creates a three-dimensional flow field. At typical part-throttle operating conditions, the throttle plate angle is in the 20 to 45° range and jets issue from the "crescent moon"-shaped open areas on either side of the throttle plate. The jets are primarily two dimensional. Figure 7.17 shows photographs taken of a two-dimensional hydraulic analog of a steady flow past a throttle plate at different throttle open angles. The path lines traced by the particles in the flow indicate the relative magnitude of the flow velocities. [18] The flow

accelerates as it divides on either side of the throttle plate. There is a stagnation point on the upstream side of the plate. The wake of the throttle plate contains two vortices, which rotate in opposite directions. The air jets exiting the throttle openings are at or near the sonic velocity. There is little or no mixing between the two jets.

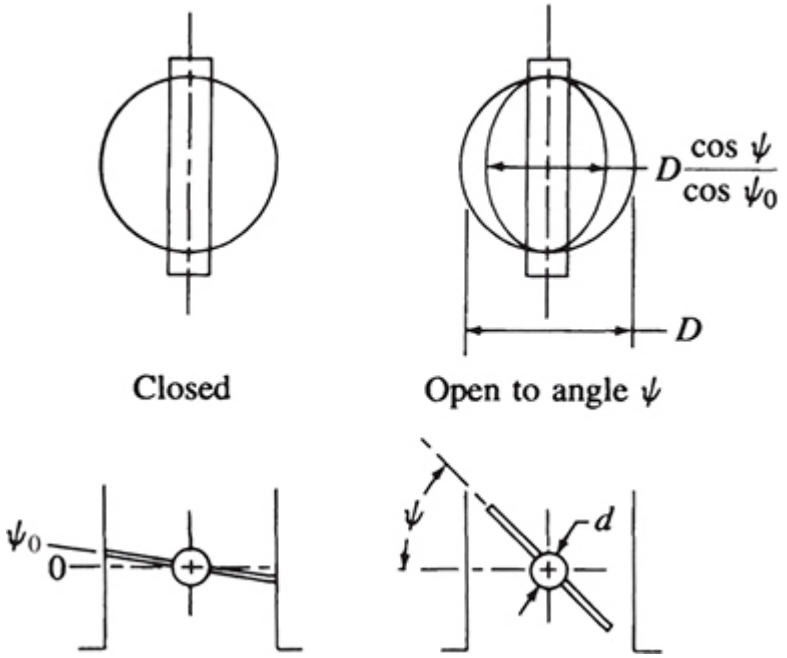


Figure 7.16 Throttle plate geometry.[11]

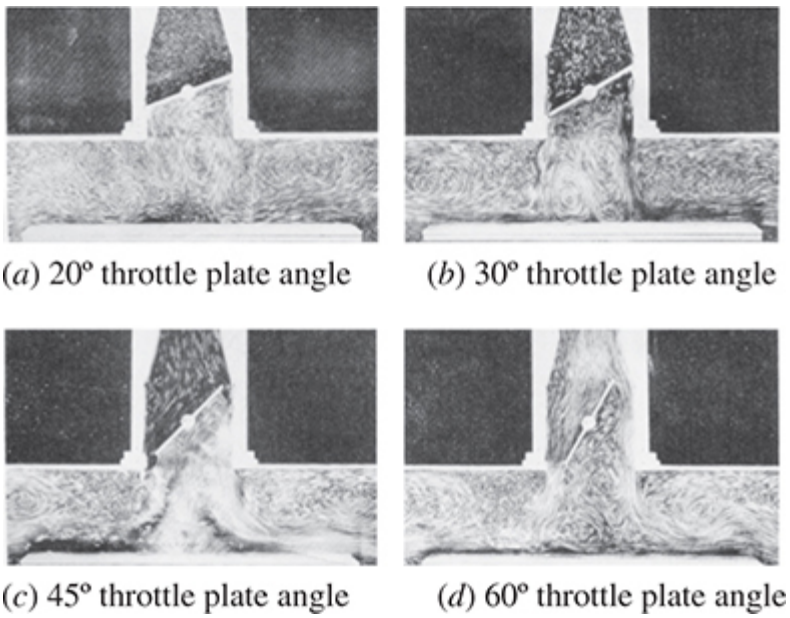


Figure 7.17 Photographs of flow in two-dimensional hydraulic analog of flow past a throttle plate onto the manifold plenum floor at different throttle plate angles.[6]

In analyzing the flow past the throttle plate, the following factors should be considered: [11, 12]

- The throttle plate shaft is usually of sufficient size to affect the throttle open area.
- To prevent binding in the throttle bore, the throttle plate is usually completely closed at some non-zero angle (10° or so).
- The discharge coefficient of the throttle plate is less than that of a smooth converging-diverging nozzle, and varies with throttle angle, pressure ratio, and throttle plate Reynolds number.
- Due to the manufacturing tolerances involved, there is often some leakage area even when the throttle plate is closed against the throttle bore. This leakage area at small throttle openings is one reason why gasoline engine management systems have a solenoid-controlled throttle by-pass passage to more accurately manage the idle airflow (see Fig. 7.8).
- The measured pressure drop across the throttle depends (±10%) on the circumferential location of the downstream pressure tap.
- The pressure loss across the throttle plate under the actual flow conditions (which are unsteady even when the engine speed and load are constant) may be less than under steady flow conditions.

The throttle plate open area A_{th}, as a function of angle ψ for the geometry in Fig. 7.16, is given by [11]

$$\frac{4A_{th}}{\pi D^2} = \left(1 - \frac{\cos\psi}{\cos\psi_0}\right) + \frac{2}{\pi}\left[\frac{a}{\cos\psi}(\cos^2\psi - a^2\cos^2\psi_0)^{1/2} + \frac{\cos\psi}{\cos\psi_0}\sin^{-1}\left(\frac{a\cos\psi_0}{\cos\psi}\right) - a(1-a^2)^{1/2} - \sin^{-1}a\right] \tag{7.5}$$

where $a = d/D$, d is the throttle shaft diameter, D is the throttle bore diameter, and ψ_0 is the throttle plate angle when tightly closed against the throttle bore.

When $\psi = \cos^{-1}(a\cos\psi_0)$, the throttle open area reaches its maximum value ($\approx \pi D^2/4 - dD$). The throttle plate discharge coefficient (which varies with A_{th}), and minimum leakage area, must be determined experimentally.

The mass flow rate through the throttle valve can be calculated from standard orifice equations for compressible fluid flow [see App. C, Eqs. (C.8) and (C.9)]. For pressure ratios across the throttle less than the critical value ($p_T/p_0 = 0.528$), the mass flow rate is given by

$$\dot{m}_{th} = \frac{C_D A_{th} p_0}{\sqrt{RT_0}} \left(\frac{p_T}{p_0}\right)^{1/\gamma} \left\{\frac{2\gamma}{\gamma-1}\left[1-\frac{p_T}{p_0}\right]^{(\gamma-1)/\gamma}\right\}^{1/2} \tag{7.6}$$

where A_{th} is the throttle plate open area [Eq. (7.5)], p_0 and T_0 are the upstream pressure and temperature, p_T is the pressure downstream of the throttle plate (assumed equal to the pressure at the minimum area: i.e., no pressure recovery occurs), and C_D is the discharge coefficient (determined experimentally). For pressure ratios greater than the critical ratio, when the flow at the throttle plate is choked,

$$m_{th} = \frac{C_p A_{th} \rho_0}{\sqrt{RT_0}} \gamma^{1/2} \left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/2(\gamma-1)} \tag{7.7}$$

The relation between airflow rate, throttle angle, intake manifold pressure, and engine speed for a 4.7-dm^3 (288-in^3) displacement eight-cylinder production engine is shown in Fig. 7.18. While the lines are predictions from a quasi-steady computer simulation, the agreement with data is excellent. The figure shows that for an intake manifold pressure below the critical value ($0.528 \times p_{atm} = 53.5$ kN/m^2 = 40.1 cmHg), the air flow rate at a given throttle position is independent of manifold pressure and engine speed because the flow at the throttle plate is choked.

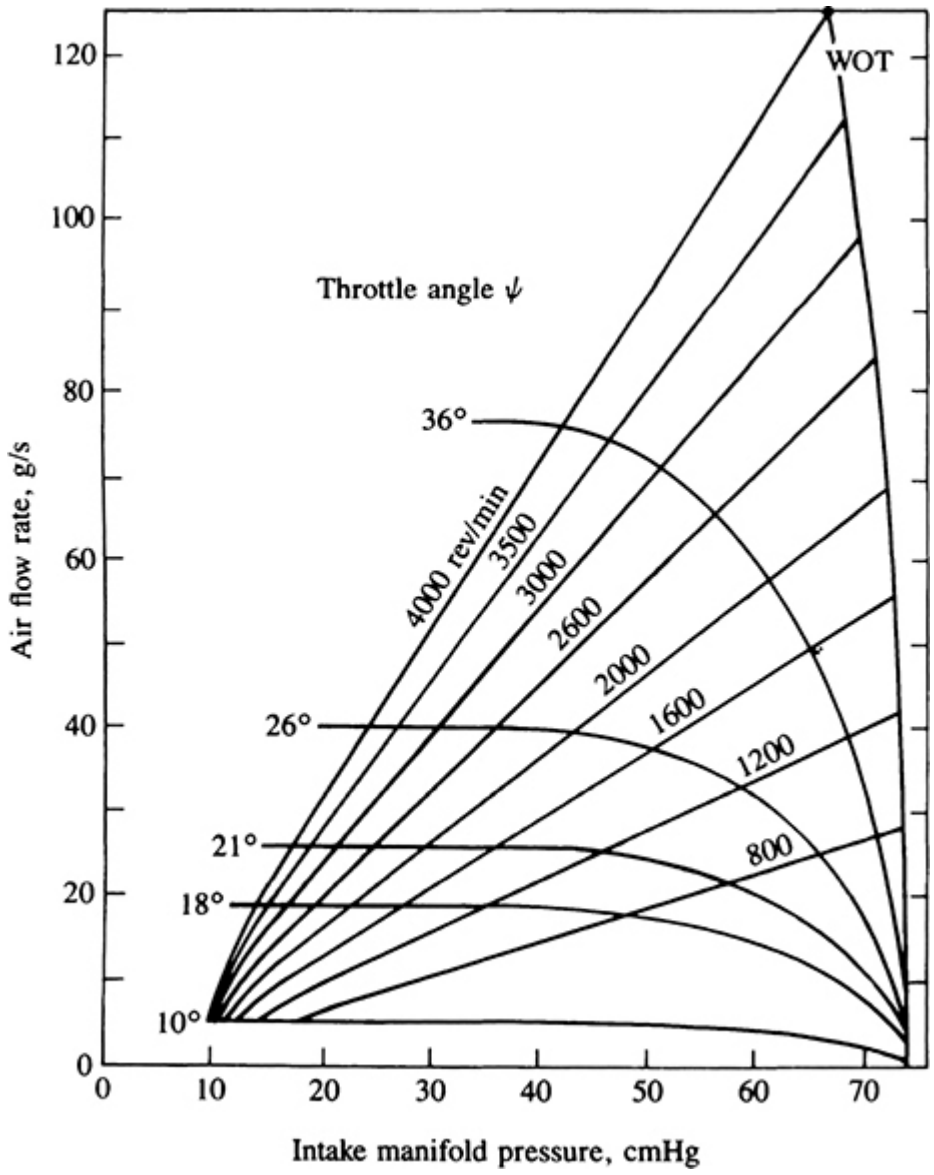


Figure 7.18 Variation in air flow rate past a throttle, with intake manifold pressure, throttle angle, and engine speed. 4.7-liter displacement eight-cylinder engine. [11]

Traditionally, in vehicle engines, the driver controlled the throttle open area to obtain the desired engine torque via the position of the accelerator pedal, which was linked mechanically to the throttle. With electronic throttle control, now widely used, the airflow required to produce the desired torque is calculated based on the position (and thus the open area) of the throttle plate.

7.5.2 Flow in Intake Manifolds

The details of the airflow in intake manifolds are complex. The combination of pulsating flow into each cylinder, different geometry flow paths through the plenum (below the throttle), through each of the manifold runners to each inlet port, and the mixing of EGR with the air just downstream of the throttle plate, under steady-state engine operating conditions are difficult enough areas to untangle. During engine transients, when the throttle position is changed, the fact that the processes which govern the air and the fuel flow to the cylinder are then substantially different introduces additional problems. This section reviews our current understanding of these airflow phenomena.

Intake manifolds consist typically of a plenum, to the inlet of which bolts the throttle body, feeding individual runners which lead to each cylinder. Important design criteria are: low air flow resistance; good distribution of air between cylinders; runner and branch lengths that take advantage of ram and tuning effects (see Secs. 6.2.4 and 6.2.6). Some of the design choices are illustrated in Fig. 7.19, which shows an intake manifold for a modern four-cylinder engine. In this design the four branches that link the plenum beneath the throttle with the intake ports are similar in length and geometry, to provide closely comparable flow paths and tuning benefits. The manifold shown is made of a nylon composite and manufactured by a lost core molding process. This allowed larger runner width and flow area, and increased runner taper or narrowing, plenum to port, increasing airflow and thus engine power.

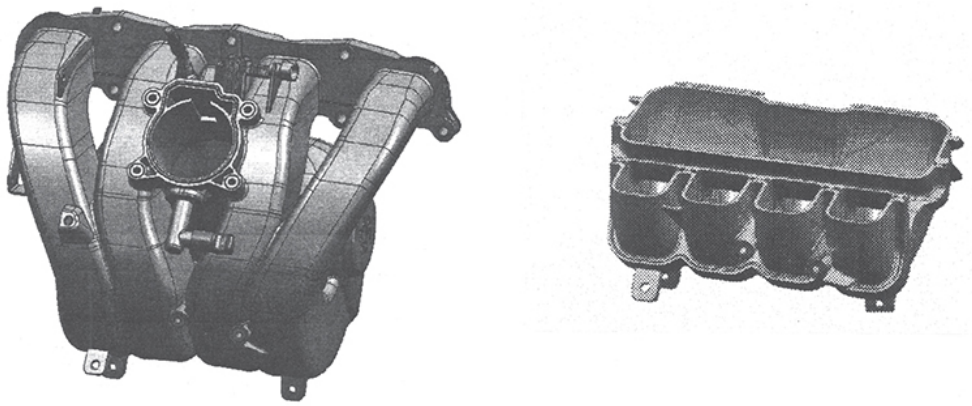


Figure 7.19 Intake manifold for a four-cylinder gasoline spark-ignition engine: (*a*) throttle and manifolds; (*b*) plenum (bottom half) and manifold

entrances.[13]

With port fuel-injection systems, the task of the intake manifold is to control the airflow (and EGR flow at low- to mid-loads). Fuel does not have to be transported from the throttle body through the entire manifold. Thus, larger and longer runners and branches, with larger angle bends, can be used to provide equal runner lengths and take greater advantage of ram and tuning effects. With port fuel injection it is not necessary to heat the manifold.

Many different intake manifold arrangements are used in practice. Different cylinder arrangements (e.g., four, V-six, in-line-six, etc.) provide quite different air distribution problems and require different solutions.

The airflow out of the manifold occurs in a series of pulses, one going to each cylinder. Since each pulse is pulled into the cylinder by the descending piston during the intake stroke, the pulse shape follows that of the piston speed versus crank angle curve (Fig. 2.2). Each pulse is approximately sinusoidal in shape. For four- and eight-cylinder engines, these flow pulses sequence such that the outflow is essentially zero between pulses. For six-cylinder arrangements the pulses will overlap. When the engine is throttled (as discussed in Sec. 7.4), backflow from the cylinder into the intake manifold occurs during the early part of the intake process until the cylinder pressure falls below the manifold pressure. Backflow can also occur early in the compression stroke before the inlet valve closes, due to rising cylinder pressure. The flow at the throttle will fluctuate as a consequence of the pulsed flow out of the manifold into the cylinders. At high intake manifold vacuum, the flow will be continuously inward at the throttle and flow pulsations will be small. When the outflow to the cylinder which is undergoing its intake stroke is greater than the flow through the throttle, the cylinder will draw air from the rest of the intake manifold. During the portion of the intake stroke when the flow into the cylinder is lower than the flow through the throttle, air (and EGR) will flow back into the rest of the manifold. At WOT when the flow restriction at the throttle is a minimum, flow pulsations at the throttle location will be much more pronounced.

Detailed measurements of the flow velocities in intake manifolds confirm the above description. At low- to mid-speeds, the airflow within the active manifold runner is approximately one dimensional and quasi-steady, and during the intake stroke follows the piston speed, as indicated in Fig. 7.20.

The mean (averaged across the flow area) flow velocity $\hat{U}$ closely follows the instantaneous piston velocity S_p. These two velocities scale together: the maximum intake manifold flow velocity is about 10 times the mean piston speed, or about 6 times the maximum piston speed (these numbers depending on the location in the manifold runner since the cross-sectional area vary). Note that at the low engine speed shown in Fig. 7.20, as the piston moves up the cylinder during the compression stroke, it pushes some of the in-cylinder mixture back into the intake prior to the intake valve closing. As engine speed increases, the momentum of the air in the manifold runner increasingly opposes this backflow: at mid-engine speed this backflow has largely disappeared. At higher engine speeds this momentum (ram) effect continues the positive airflow after BC: see Sec. 6.2.4 . Also at higher engine speeds, the higher frictional flow resistance (which depends on $\hat{U}^2$) and pressure-wave-induced flow rate variations distort this quasi-steady behavior to some degree.

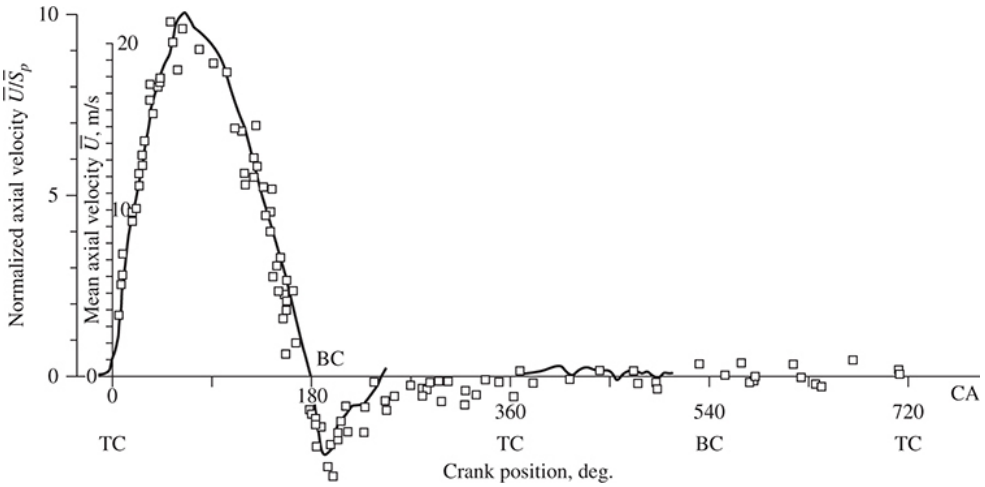


Figure 7.20 Mean and instantaneous intake manifold air velocities during the engine's operating cycle in a production diesel engine: speed = 717 rev/min; mean piston speed = 2.2 m/s; stroke = 92 mm.[14]

The airflows to each cylinder of a multi-cylinder engine, even under steady operating conditions, are not identical. This is due to differences in runner layout and length, as well as other geometric details of the flow path to each cylinder. Also, as each cylinder's intake flow commences, air is drawn from the branch or runner leading to the cylinder, the plenum, and the

other runners feeding the plenum, as well as past the throttle. This "drawing down" of other parts of the intake manifold depends on the arrangement of the plenum, and runners, and the firing order of the cylinders. Thus the air flow to each individual cylinder is affected by the details of its own branch, how that branch connects to the rest of the intake manifold, and the cylinder firing order. [15] Modern manifold designs keep these differences between the air flows to individual cylinders modest: variations in the average air mass flow rate to each individual cylinder are less than a couple percent.

Typical quantities that characterize manifold air flow are given in Table 7.2 for a four- and an eight-cylinder naturally-aspirated gasoline engine. The volume of air pulled into each cylinder per cycle is about the same as the volume of one direct flow path between the throttle plate and inlet valve. Thus, one intake stroke loads the manifold, and the next one pulls that air into the cylinder.

TABLE 7.2 Parameters that characterize manifold air flow

Engine geometry	I-4*	V-8†
Typical flow-path distance between throttle bore and intake valve, cm	33	30
Average intake-passage flow area, cm^2	10	16
Volume of one direct flow path from throttle bore to intake valve, cm^3	350	500
Range of speeds, etc.	**Maximum**	**Minimum**
Crankshaft, rev/min	6500–5500	600
Peak air velocity in manifold branch, m/s	180*, 140†	15
Peak Reynolds number in manifold branch	5×10^5	5×10^4
Duration of individual cylinder intake process, ms	35	3–4

*1.8-liter four-cylinder in-line SI engine.[16]
†5.6-liter V-8 cylinder SI engine.[17]

An additional phenomenon becomes important when engine load is changed by opening or closing the throttle: the mass of air in the induction system volume takes a finite time to adjust to the new engine operating conditions. For example, as the throttle is opened the airflow into the manifold increases as the throttle open area increases. However, due to the finite volume of the manifold, the pressure level in the manifold increases more slowly than would be the case if steady-state conditions prevailed at each throttle position. Thus, the pressure difference across the throttle is larger than it would be under steady flow conditions and the throttle airflow

overshoots its steady-state value. The airflow into each cylinder depends on the pressure in the manifold, so this lags the throttle airflow. This transient airflow phenomenon affects fuel metering. For throttle-body injection, fuel flow should be related to throttle airflow. For port fuel injection, fuel flow should be related to the airflow into the cylinder. Actual results for the airflow rate and manifold pressure in response to an opening of the throttle (increase in throttle angle) are shown in Fig. 7.21. The overshoot in throttle airflow and lag in manifold pressure as the throttle angle is increased are evident. Opposite effects will occur for a decrease in throttle angle—reduction in airflow through the throttle.

7.5.3 Air Flow Models

Several approaches to modeling the flow in an intake manifold have been proposed. One simple manifold model that describes many of the above phenomena is the *plenum* or *filling and emptying* approach. It is based on the assumption that at any given time, the manifold pressure is uniform. The continuity equation for air flow into and out of the intake manifold is

$$\frac{dm_{a,m}}{dt} = \dot{m}_{a,\text{th}} - \sum \dot{m}_{a,\text{cyl}} \tag{7.8}$$

where $m_{a,m}$ is the mass of air in the manifold, and $\dot{m}_{a,\text{th}}$ and $\dot{m}_{a,\text{cyl}}$ are the air mass flow rates past the throttle and into each cylinder, respectively. The flow rate past the throttle is given by Eq. (7.6) or (7.7). For manifold pressures sufficiently low to choke the flow past the throttle plate, the flow rate is independent of manifold pressure. The mass flow rate to the engine cylinders can be modeled at several levels of accuracy. The airflow through the valve to each cylinder can be computed from the valve area, discharge coefficient, and pressure difference across the valve; or a sine wave or $S_p(\theta)$ function can be assumed. In the general case, Eq. (7.8) must be combined with the first law for an open system (see Sec. 14.2.2). For calculating the manifold response to a change in load or throttle setting, simplifying assumptions can be made. A quasi-steady approximation for the air flow into the cylinder

$$\sum \dot{m}_{a,\mathrm{cyl}} = \frac{\eta_v \rho_{a,m} V_d N}{2}$$

is usually adequate, and the air temperature can be assumed constant. Then, using the ideal gas law for the manifold, $p_m V_m = m_{a,m} R_a T_m$, Eq. (7.8) can be written as

$$\frac{dp_m}{dt} + \frac{\eta_v V_d N}{2V_m} p_m = \dot{m}_{a,\mathrm{th}} \frac{RT_m}{V_m} \tag{7.9}$$

Both η_v and $\dot{m}_{a,\mathrm{th}}$ have some dependence on p_m [e.g., see Eq. (6.2)]. In the absence of this weak dependence, Eq. (7.9) would be a first-order equation for p_m with a time constant $\tau = 2V_m/(\eta_v V_d N) \approx V_m/\dot{V}_{\mathrm{cyl}}$ which is two to four times the intake stroke duration. The smooth curves in Fig. 7.21 are predictions made with Eq. (7.9) and show good agreement with the data. The plenum model is useful for investigating manifold pressure variations that result from load changes. It provides no information concerning pressure variations associated with momentum and wave effects.

The dynamics of the flow in multi-cylinder intake (and exhaust) systems can be modeled using one-dimensional unsteady compressible flow equations. Historically, the governing equations were solved using the method of characteristics (see Benson et al. [18]). Finite difference techniques are now used. [19] The assumptions usually made in this unsteady flow analysis are:

- The intake (and exhaust) system can be modeled as a combination of pipes, junctions, and plenums.
- Flow in the pipes is one dimensional and no axial heat conduction occurs.
- States in the engine cylinders and plenums are uniform.
- Boundary conditions are considered quasi steady.
- Coefficients of discharge, heat transfer, pipe friction, and bend losses for steady flow are valid for unsteady flow.
- The gases can be modeled as ideal gases.

This approach to intake and exhaust flow analysis is discussed more fully in Sec. 14.3.4 .

7.6 FUEL FLOW PHENOMENA: PORT FUEL INJECTION

7.6.1 Liquid Fuel Behavior

With port-fuel-injection liquid-fuel metering systems, the fuel enters the air stream as a liquid jet, which atomizes into droplets. These drops then move through the air and mostly deposit on the walls of the intake ports and valve heads. Some of the smaller droplets vaporize; vaporization of the now hotter liquid fuel on the walls also occurs. The flow of liquid fuel along the walls is significant: the transport of fuel as vapor, droplets, and liquid streams or films can all be important. The fuel transport processes in the intake port region are obviously extremely complex.

The details of the fuel transport process are different for multipoint port fuel-injection systems than for throttle-body injection systems. For the latter systems, fuel must be transported past the throttle plate and through the complete intake manifold. For the former systems, the liquid fuel is injected into the inlet port, toward the back of the intake valve. For all these practical fuel metering systems, the quality of the mixture entering the engine is imperfect. The fuel, air, recycled exhausts, mixture is not homogeneous; the fuel may not be fully vaporized as it enters the engine. The charge going into each cylinder is not usually uniform in fuel/air ratio throughout its volume, and the distribution of fuel between the different engine cylinders is not exactly equal. During engine transients (e.g., during vehicle accelerations and decelerations) when engine fuel and air requirements and manifold conditions change, it is obvious that the above fuel transport processes will not all vary with time in the same way. Thus, in addition to the transient non-quasi-steady airflow phenomena described in the previous section, there are transient fuel-flow phenomena. These have to be compensated for in the fuel metering strategy.

Gasoline (which may be blended with 5, 10, or more percent ethanol), the standard spark-ignition engine fuel, is a mixture of a large number of individual hydrocarbons: it has a boiling temperature range rather than a single boiling point as discussed in Sec. 7.2.2 . That discussion indicates how the equilibrium fraction of fuel evaporated at a given temperature and pressure can be calculated. Figure 7.3 shows the effect of mixture temperature on percent of indolene fuel (a specific gasoline) evaporated. While insufficient time is usually available in the manifold to establish equilibrium, the trends shown are indicative of what happens in practice: lower pressures increase the relative amount of fuel vaporized and fuel heating from warm or hot surfaces is required to vaporize a substantial fraction of the fuel in the engine intake. With port fuel-injection systems the fuel is injected onto the back of the inlet valve (and surrounding port walls), usually while the valve is closed or only partly open. Vaporization of liquid fuel off the valve and walls occurs, enhanced by the backflow of hot residual gases from the cylinder (especially at part-load). Even under fully warmed-up engine conditions, some fuel is carried into the cylinder as liquid.

These fuel pathways into and within the cylinder are illustrated diagrammatically, and in a flow diagram, in Fig. 7.22. During the injection period, liquid fuel drops in the spray impinge on the back of the intake valve head, and on the port walls approaching the valve seat, adding to the liquid films already there. This occurs with both closed-valve injection (the most common approach at part-load and lower speeds), as well as open-valve injection. When the intake valve opens, under throttled conditions (manifold pressure below atmospheric), the blowdown backflow (see Sec. 7.4) can significantly redistribute these liquid films. When, following any backflow, the forward flow of fresh charge commences and the piston accelerates down the cylinder, the high velocity airflow past the valve seat and head strips liquid from these films, and shatters that liquid into small drops which are carried by the airflow into the cylinder. Fuel vaporization from airborne drops and these films adds more fuel vapor to the in-cylinder fuel, air, residual gas, mixture. Inside the cylinder, larger drops can impinge on the combustion chamber walls forming liquid films on the intake valve head, the cylinder head, the liner, even the piston, depending on geometric details and the temperatures of the air-residual gas mixture and these wall surfaces. When the engine is cold—during starting and the first few minute or so of operation, there can be substantial liquid on these walls (both in the intake

port and the cylinder). In a warmed-up engine, only a few percent of the injected fuel enters the cylinder as a liquid.

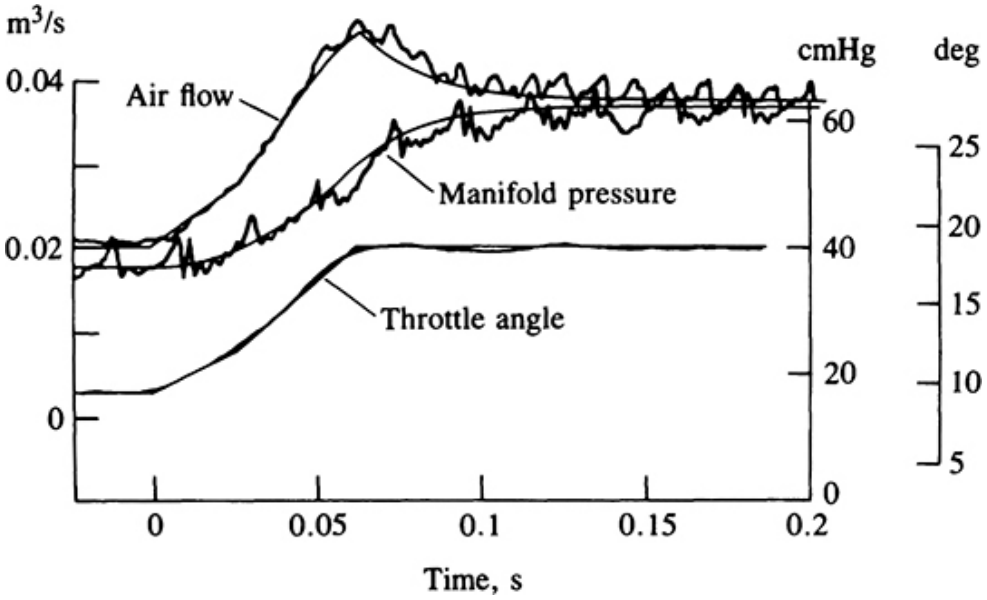


Figure 7.21 Throttle angle, intake manifold pressure, and airflow rate past the throttle versus time for 10° part-load throttle opening. 5-liter V-8 engine. [17]

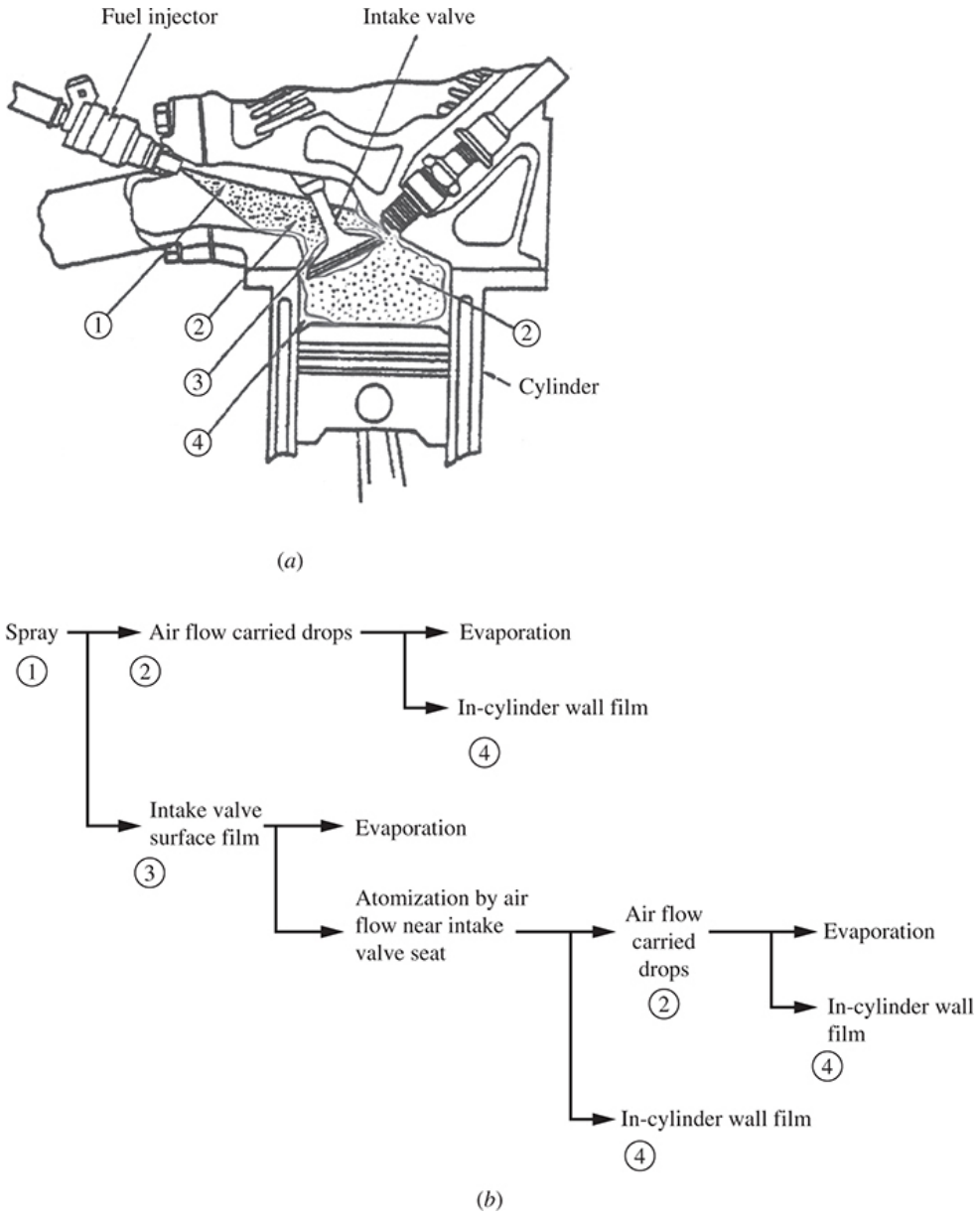


Figure 7.22 (*a*) Schematic of fuel behavior during port-fuel-injection gasoline-engine intake process. (*b*) Fuel flow diagram from port fuel injector into and within the cylinder. [20]

During cold engine starting, substantial additional fuel beyond that required if fully vaporized to produce a stoichiometric in-cylinder mixture, is injected in the first (and sometimes second) injection for each cylinder. This is because, as Fig. 7.4 *a* and *b* indicate, at these ambient temperatures, only a moderate fraction of the fuel injected will vaporize. However, with excess

(liquid) fuel in the intake port from these first injections, succeeding injections must be reduced below the nominal stoichiometric requirement to (over time) vaporize and burn the initial excess.

Calibrating the engine to manage the fuel metering at these different thermal conditions and through these thermal transients requires time-consuming engine testing. Superimposed on these thermal transients are engine throttle-change transients where the driver accelerates or decelerates the vehicle. These latter transients have characteristic time constants of 0.1 to 1 seconds. Engine start-up transients have similar time constants. Engine warm-up can take several minutes, depending on the initial engine temperature, and the engine load and speed (engine heat transfer from the hot burned gases to the cylinder walls varies significantly with these two parameters). Section 7.6.2 discusses controlling the fuel metering during engine acceleration and deceleration transients in more detail.

Figure 7.23 illustrates several of the above described fuel behaviors with photos extracted from intake port and in-cylinder high-speed videos taken under lightly loaded lower-speed operating conditions in engines with appropriate optical access. The white areas in the photos indicate where light is being reflected from liquid fuel on the walls (around the valve seat, and the valve boss and stem), from drops in the fuel injection spray (Fig. 7.23*b*), and drops produced by the high velocity exhaust backflow (Fig. 7.23*d*). In sequence, once the engine starts firing (one or a few injections), it then takes several injections into each intake port to build-up significant liquid fuel close to the valve seat and on the valve head and stem. Then as liquid accumulates in these regions, a quasi-steady behavior becomes apparent.

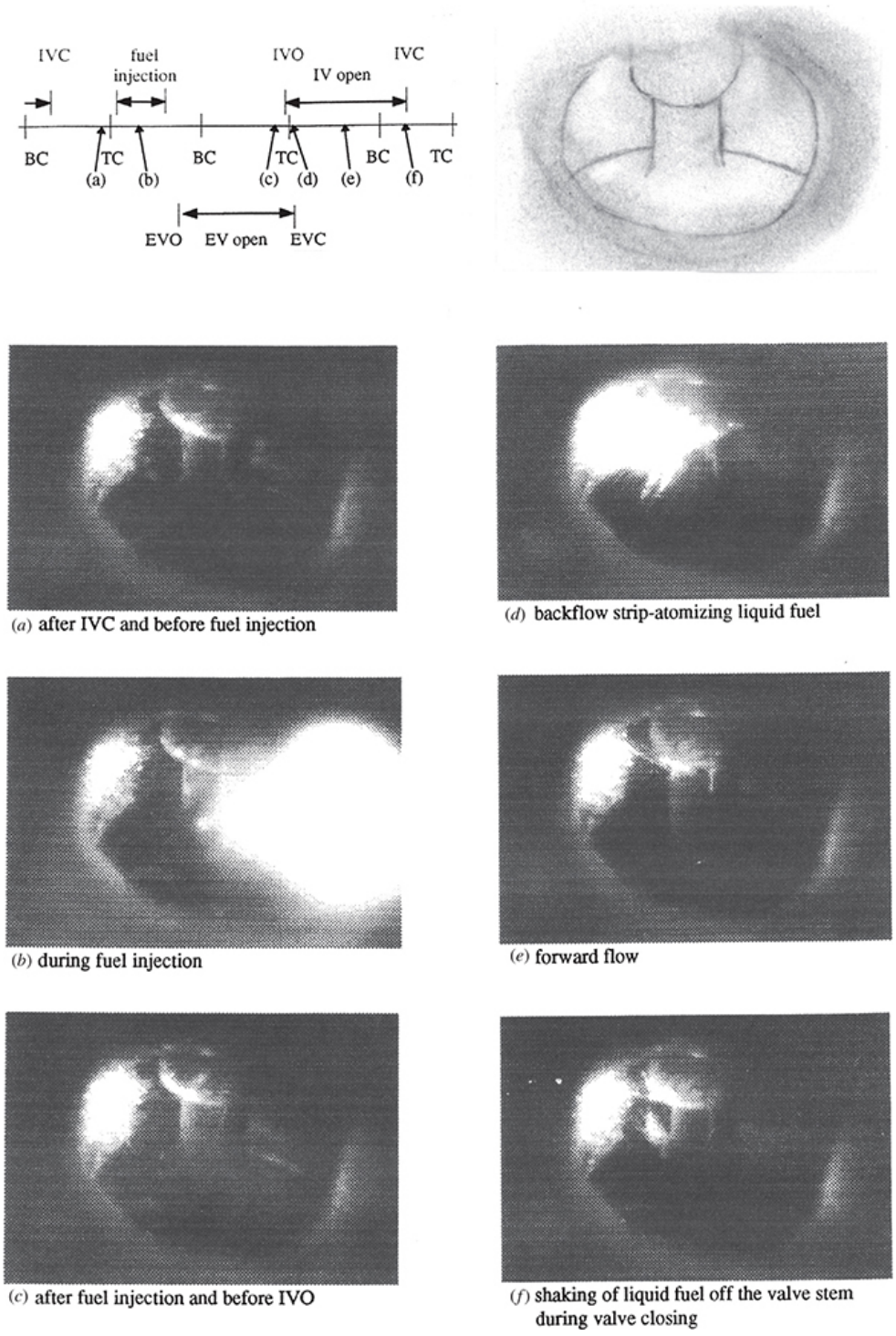


Figure 7.23 Images from video visualizing fuel spray and liquid film behavior in one intake port of a four-valve cylinder head gasoline engine: stoichiometric operation at 1200 rev/min, 0.5 bar intake pressure, cylinder head at 25°C. White regions indicate where light is being reflected from liquid fuel (as films or moisture on wetted surfaces), or from droplets in the

primary fuel-injected spray, or produced by the exhaust backflow atomizing liquid fuel that has accumulated around the valve seat.[10]

During the hot cylinder gas blowback into the intake port, there is obviously no fuel transport into the cylinder. But once liquid fuel has accumulated near the valve seat, the high velocity reverse flow starting at IVO causes substantial atomization and transport of fuel droplets backward into the intake port. When the forward flow of gases from the intake port past the valve seat commences (about 30° ATC at which time the piston is moving at about two-thirds of its mean piston speed: see Fig. 2.2), that forward flow drags liquid fuel from the films on the port walls toward the valve seat region and atomizes that liquid fuel on the port walls and on the valve head into fine drops that enter the cylinder in a short burst or "puff" some 20 to 30° later. The duration and extent of that burst decreases as the engine starts to warm up and the amount of liquid fuel on the port and valve walls decreases. As the forward airflow continues beyond this point, there is no obvious sign of continuing liquid fuel transport into the cylinder (through vapor fuel flow may continue). Also, under engine conditions where there is still substantial liquid fuel in the port near the valve seat, when the intake valve closes, the liquid fuel between the valve head and seat is squeezed out of the closing gap splattering large drops of fuel into the combustion chamber (where they impinge on the cylinder liner) and also back into the intake port. While the engine is still cold and in-cylinder liquid films persist on the valve head, cylinder head (and maybe other surfaces), the passage of the flame over these films (which are thick enough to "ripple") causes no visible change in their appearance. As the surface temperatures steadily rise, the films become thinner, smaller in area extent, transition in appearance to a "moist surface," and then disappear. [21]

Fuel Droplet Behavior

With fuel injection systems, the liquid fuel atomizes as it enters the air stream. The velocity of the liquid jet once it exits the nozzle is high enough to shatter the flowing liquid, and its interaction with the airflow further atomizes the fuel. Typical droplet-size distribution varies over the load and speed range. Droplet diameters in the 25 to 100 μm range are usually assumed to be representative: larger drops are also produced. The liquid fuel

drops are accelerated by the surrounding air stream and start to vaporize. Vaporization rates have been calculated using established formulas for heat and mass transfer between a droplet and a surrounding flowing gas stream (see Ref. 22 for a review of methods of calculating droplet vaporization rates). Calculations of fuel vaporization show that the temperature of the liquid fuel droplets decrease rapidly (by up to about 30°C), and the fraction of the fuel vaporized in the intake port is small (in the 2 to 15% range).

Liquid fuel drops, due to their density being many times that of the air, will not exactly follow the air flow. Droplet impaction on the walls may occur as the flow changes direction, when the inertia of the droplets causes them to move across the streamlines. The equation of motion for an individual droplet in a flowing gas stream is

$$\left(\frac{1}{6}\pi D_d^3 \rho_f\right)\mathbf{a} = m_d\mathbf{g} - \frac{1}{2}(\mathbf{v}_d - \mathbf{v}_g)|\mathbf{v}_d - \mathbf{v}_g|\rho_g C_D \frac{\pi D_d^2}{4} \tag{7.10}$$

where D_d is the droplet diameter, ρ_f and ρ_g are liquid and gas densities, $\mathbf{v}_d$ and $\mathbf{v}_g$ are the droplet and gas velocities, $\mathbf{a}$ is the droplet acceleration, $\mathbf{g}$ acceleration due to gravity, and C_D is the droplet drag coefficient. For $6 < \mathrm{Re} < 500$ the drag coefficient of an evaporating droplet is a strong function of the Reynolds number, Re: for example,

$$C_D = 27\,\mathrm{Re}^{-0.84} \tag{7.11}$$

where $\mathrm{Re} = (\rho_g D_d\,|\mathbf{v}_d - \mathbf{v}_g|/\mu_g)$.

Studies of droplet impaction and evaporation using the above equations and typical manifold conditions and geometries indicate the following. For 90° bends, drops of less than 10 μm diameter are essentially carried by the gas stream (<10% wall impaction); almost all droplets larger than 25 μm impact on the walls. Droplet sizes produced by secondary atomization away from the fuel nozzle exit depend on the local gas velocity: higher local relative velocities between the gas and liquid produce smaller drop sizes. Approximate estimates which combine the two phenomena outlined above show that at low engine air flow rates, almost all of the fuel will impact on the port walls as the flow turns as it approaches the valve head. At high air

flows, because the drops are smaller, a substantial fraction of the drops may stay entrained in the air flow. Secondary atomization as the air flows over fuel films at the valve seat is important to the fuel transport process: the high air velocities at the edge of the valve seat and head produce droplets of order or less than 10 μm diameter. However, coalescence and deposition on the walls and subsequent re-entrainment are likely to increase the mean droplet size.

Estimates of droplet evaporation rates in the manifold indicate the following. With a residence time in the manifold of about 10 ms (100 deg. CA at 1800 rev/min), only drops of size less than about 10 μm will evaporate. Thus, only drops small enough to be carried by the air stream are likely to vaporize in the intake: the larger droplets will impact on the walls, anyway.

Fuel-Film Behavior

The fuel which impacts on the wall will also vaporize and, depending on where fuel deposition occurs and the local wall geometry, may be dragged by the flowing air along the wall as a liquid film or rivulet. If the vaporization rate off the wall is sufficiently high, then a liquid film will not build up. Any liquid film or pool on the intake port or valve walls is important because it introduces additional fuel transport processes—deposition, liquid transport, and evaporation—which together have a much longer time constant than the air transport process. Thus changes in the air and the fuel flow rate into each engine cylinder, during a change in engine load, will not occur in phase with each other unless compensation is made for the slower fuel transport. Models for following these transients are discussed next.

7.6.2 Transients: Fuel-Film Models

Several models of the behavior of liquid fuel wall-films have been developed. One approach analyzes the liquid fuel mass on the walls of the intake port and valve as one or more "puddles." Figure 7.24 illustrates the fuel transport processes involved. (The puddles are real: the model is conceptual.) A portion x of the injected fuel in the spray enters the puddle; fuel leaves the puddle through vaporization and, as liquid pulled from the puddle by the flowing air. The equation for the rate of change of the puddle fuel mass $m_{f,p}$ is

$$\dot{m}_{f,p} = \dot{m}_{f,\text{in}} - \dot{m}_{f,\text{out}} = x\,\dot{m}_{f,m} - \frac{m_{f,p}}{\tau} \tag{7.12}$$

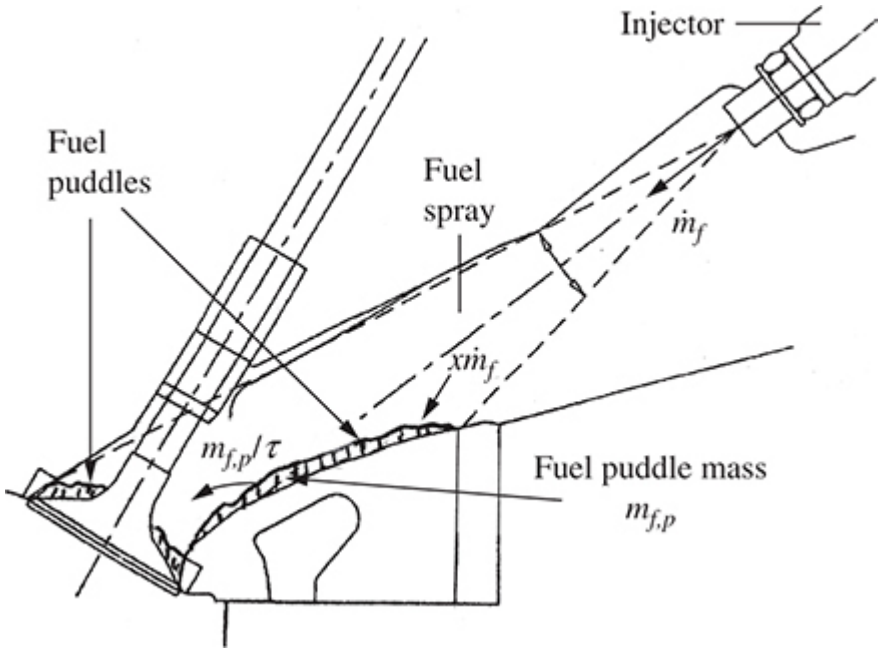


Figure 7.24 Schematic of x – τ transient fuel spray and puddle model indicating how a fraction (x) of the injected fuel in the spray ($\dot{m}_f$) impacts the port and valve head surfaces forming a "puddle" before entering the cylinder: τ is the average fuel residence time in the puddle.

where $\dot{m}_{f,p}$ is the mass of fuel in the puddle, $\dot{m}_{f,m}$ is the metered fuel flow rate, and x is the fraction of the metered flow that enters the puddle. It is assumed that the re-entrainment plus evaporation rate is proportional to the mass of fuel in the puddle divided by the characteristic time τ of the re-entrainment/evaporation process. The puddle behavior predicted by this model in response to a step increase in engine load is shown in Fig. 7.25 *a*. The fuel delivery rate to the cylinder, $\dot{m}_{f,c}$, is given by

$$\dot{m}_{f,c} = (1-x)\dot{m}_{f,m} + \frac{m_f}{\tau} \tag{7.13}$$

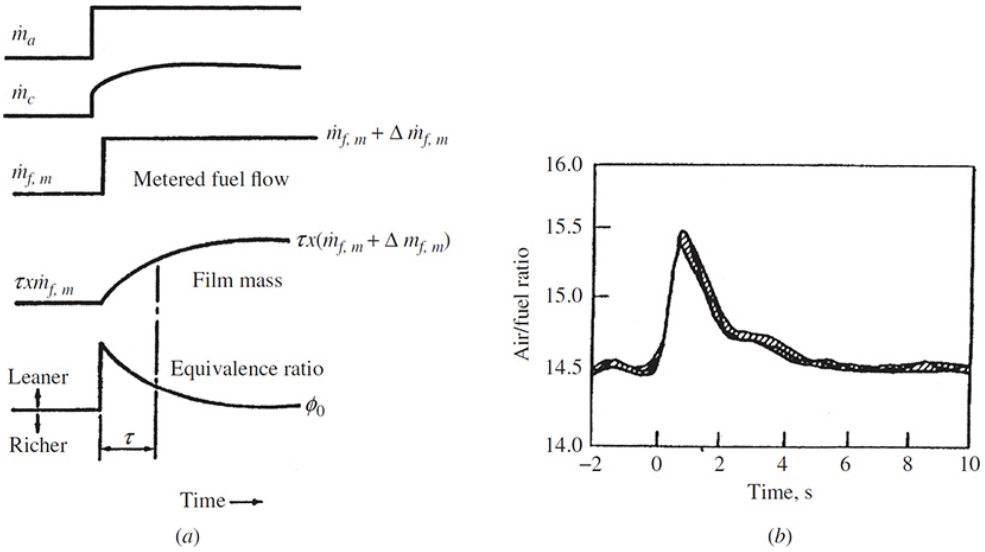


Figure 7.25 (*a*) Predicted behavior of the fuel film for an uncompensated step change in engine operating conditions. (*b*) Observed variation in air/fuel ratio for an uncompensated throttle opening at 1600 rev/min which increased manifold pressure from 48 to 61 cmHg. [23]

Because only a fraction of the fuel, (1 - x), flows directly with the air into the cylinder, when the throttle is opened rapidly a lean air/fuel ratio excursion is predicted. Figure 7.25 shows that this behavior (without any metering compensation) is observed in practice. Estimates of the volume of fuel in the puddle (for a 5-liter V-8 engine) are of order 1000 mm ^{3}, and increase with increasing load and speed. The time constant is of order 1 second for a fully warmed-up engine; it varies with engine operating conditions and is especially sensitive to the puddle, and therefore wall temperature. Such models have been used primarily to develop fuel metering strategies which compensate for the fuel transport lag. [23] Since the valve head and port wall temperatures differ significantly and warm-up at different rates, these puddles or x - τ models need additional sophistication and substantial calibration.

With these multipoint port fuel-injection systems, the fuel transport processes can be summarized as follows: the air flow to each cylinder depends on throttle open angle, engine speed, geometric details of the intake manifold and port design, and valve timing and lift. Because the manufacture and operation of individual fuel injectors are not identical, there is some variation in fuel mass injected cylinder-to-cylinder and cycle-to-cycle. Since

individual cylinder airflows depend on the design of the manifold, whereas the amount of fuel injected does not, uniform air distribution is especially important with these port injection systems. The fuel vaporization and transport processes depend on the duration of injection and the timing of injection pulse(s) relative to the intake valve-lift profile. Much of the injected fuel will impinge on the port walls, valve stem, and the valve head, especially when injection toward a closed intake valve occurs. Backflow into the intake port of hot residual gases at part-load operation will have a significant effect on fuel vaporization. Compensation for fuel lag during transient engine operation is still required; sudden throttle openings are accompanied by a "lean spike" in the mixture delivered to the engine unless fuel compensation—injection of additional fuel—is utilized. Thus wall wetting, evaporation of liquid fuel off the walls, and liquid flow along these walls are all likely to be important components of the mixture preparation process.

With port fuel-injection systems, some liquid fuel does enter the cylinder, and droplets are present in the cylinder during intake and compression. Measurements made of the distribution, size, and number density of these in-cylinder fuel droplets indicate that during intake with open-valve fuel injection, the droplet number density in the clearance volume increased to a maximum at the end of injection (injection lasted from 45 to 153° ATC) and then decreased due to evaporation during compression to a very small value at the time of ignition. Average in-cylinder droplet size during intake was 10 to 20 μm in diameter; it increased during compression as the smaller drops in the distribution evaporated. At the conditions tested, some 10 to 20% of the fuel was in droplet form at the end of injection. At ignition, the surviving droplets contained a negligible fraction of the fuel. During injection, the distribution of droplets across the clearance volume was non-uniform. It became much more uniform with time, after injection ended. [24]

7.7 DIRECT FUEL INJECTION

7.7.1 Overview of Direct-Injection Approaches

Direct-injection of gasoline (or another liquid fuel) into the cylinders of a spark-ignition engine offers several advantages. The two primary operational advantages of DI are first, that the fuel goes directly into the combustion chamber (the cylinder) where it will be burned (see Fig. 7.2); second, the fuel distribution within the cylinder can be controlled and *stratified* by mixing the injected fuel with only part of the in-cylinder air, and confining the fuel-air mixture to a specific region of the combustion chamber. The first of these advantages allows more rapid and precise control of mixture preparation as engine operating requirements change since the fuel transport delay from the intake port into the cylinder with port fuel injection (Sec. 7.6.2) is removed. Better engine-in-vehicle drivability results. The second opportunity allows the engine to operate fuel lean overall, like a diesel—at least at part-load—while maintaining an easily combustible close-to-stoichiometric fuel-vapor air mixture in the proximity of the spark plug.

Secondary benefits follow from these. Fuel vaporization within the air charge in the cylinder cools the charge, allowing higher compression ratios to be used since that cooling delays the onset of knock. It also results in the engine inducting more air thereby increasing WOT torque. Tighter control of mixture preparation, and especially stratified overall-lean operation, improve the engine's specific fuel consumption (as does the higher compression ratio). Thus, direct fuel injection offers increased output, higher fuel conversion efficiency, though at higher fuel-system cost.

Early efforts to develop direct-injection spark-ignition engines go back many decades. The primary objective then was stratified operation, and even a multi-fuel operating capability: see Sec. 1.7.4 . With the fuel injection technology then available, and our much more limited understanding of fuel spray behavior, these efforts were not successful. Improvements in fuel injection technology and our knowledge of and capability for analyzing in-cylinder flows, spray behavior, ignition, and combustion, have resulted in production direct-injection gasoline engines being developed and marketed in the late 1990s. [25, 26, 27] These engines operated stratified (lean overall) at low- to mid-loads and transitioned to operation with a homogeneous stoichiometric fuel-vapor air mixture at mid- to higher-loads. Stratified operation requires fuel injection to occur during the latter part of the engine's compression stroke (*late injection*) to restrict the extent of fuel-air mixing that occurs. The homogeneous stoichiometric operating mode requires *early injection*, during the intake stroke, to allow as much time as possible for

more complete fuel-air mixing. Direct-injection applications have focused more on homogeneous (stoichiometric) operation with early fuel injection since the very high effectiveness of the three-way catalyst technology [d] makes this an exceptionally low emissions DI engine concept with superior fuel economy. DI engines also offer significant additional benefits when turbocharged. While stratified DI engine concepts do offer enhanced fuel consumption at lighter loads, developing highly effective lean-NO_x catalysts is challenging: see Sec. 11.6.3 . These two DI classifications—homogeneous and stratified—are now widely used. Extensive discussion of GDI spark-ignition engines can be found in Zhao et al., [28] and Ando and Acroumanis. [29]

7.7.2 DI Mixture Preparation Processes

The fuel-air mixture preparation processes in direct-injection spark-ignition engines connect with the engine's ignition and combustion processes in a much more detailed way than with port fuel injection into the intake. The reasons are obvious: the mixture preparation takes place within the same space (the cylinder) where combustion will occur, and the time available for mixing is shorter. The fuel drops within the injected spray must be dispersed appropriately (throughout the full combustion chamber with homogenous operation—in a specific region of the combustion chamber with stratified operation). The drops must vaporize rapidly and should not impinge significantly on the chamber walls. The in-cylinder flow, which is generated by the details of the flow past the intake valve head and seat, and then the cylinder head and piston crown geometry, must enhance this droplet distribution and vaporization, and the fuel-vapor, air, residual, and EGR (if used) mixing processes to achieve the desired homogeneous or stratified mixture objective.

The factors involved in these two DI mixture preparation processes—homogeneous and stratified—are the following: the desired mixture composition (its stoichiometry), the fuel injector and injection timing details, the geometry of the combustion system—injector location and orientation, nozzle design, intake generated in-cylinder flow (tumble or swirl), cylinder head and piston crown geometry (e.g., bowl-in-piston shape for stratified operation), spark plug gap location. Several of these vary with engine load

and speed; satisfactory mixture preparation must be achieved over the full engine operating range.

Table 7.3 summarizes the commonly made choices for these several mixture preparation factors for the two different DI applications: homogeneous and stratified engine operation. Note that when the stratified mixture approach is used at part-load, this requires homogeneous operation at higher-loads (and speeds). Most of the development and design effort has focused on direct-injection engines with gasoline fuel (which may contain some 10% ethanol): DI ethanol (or E85) fueled engines are also an option. With homogeneous mixture combustion where creation of a relatively uniform fuel-air mixture is the objective, a convenient injector location is below the intake ports. With a wide-angle spray and a tumbling [e] in-cylinder motion, good fuel dispersion, vaporization, and mixing can be achieved. Fuel injection during the mid-portion of the intake stroke provides adequate mixture preparation time and holds liner and piston wetting with liquid fuel to a minimum. With homogeneous-charge engine operation, other engine processes are similar to those with port fuel injection (except for the in-cylinder evaporative cooling of the charge with DI; see below).

TABLE 7.3 Major mixture preparation features of homogeneous and stratified direct-injection SI engines

Feature	Homogeneous mixture	Stratified mixture
Load/speed range	Full load and speed range	Part-load, low- to mid-speed
Nat. asp./turbo	Both	Both: but only at part-load
Stoichiometry	Stoichiometric *A/F*	Lean overall: locally close-to-stoichiometric
Mixture preparation approach	Early injection: mid-intake stroke Wide spray dispersion: uniform mixture	Late injection: mid to late compression stroke. Confined mixture cloud around spark plug
Fuel injector: Location	Usually below the intake ports	Below intake ports (wide spacing); or close to spark plug (narrow spacing)*
Design	Usually multi-hole nozzle with wide spray angle	Various designs to fit with stratified combustion approach
Fuel supply pressure (MPa)	5–15	5–20
Airflow	Tumble, to improve fuel-air mixing and realize faster combustion	Specific airflow (tumble or swirl) to match with spray and bowl-in-piston geometry
Piston geometry	Close to conventional piston crown design	Specific bowl-in-piston to direct and confine developing fuel spray
Emission control approach	Spark retard (cold start), EGR, three-way catalyst	Spark retard (cold start), lean with EGR, lean-NO_x storage catalyst

*Wide or narrow spacing refers to injector nozzle to spark plug spacing: see text.

With stratified charge operation, the injector nozzle and spark-plug locations, the fuel injection details, the combustion chamber geometry, and the in-cylinder airflow are all key components of the various combustion approaches used. These are often grouped into three air and fuel flow mixture preparation categories: wall-guided, air-guided, and spray-guided. In practice, two basic approaches have evolved: (1) where the injector nozzle and spark plug are *widely spaced*; (2) where these two components are *closely spaced*. See Fig. 7.26. These sketches illustrate in (*a*) how, with widely spaced injector nozzle and spark plug, both wall-guiding with a specially shaped piston crown combined with air-guiding (e.g., tumble) are used to create the desired fuel spray behavior—an approximately stoichiometric fuel-vapor/air mixture cloud that arrives at the spark plug by time of ignition. In Fig. 7.26 *b*, the spark plug electrodes are placed within the developing fuel spray: it is the spray behavior (guiding) that brings a combustible mixture to the spark plug location at the time of ignition.

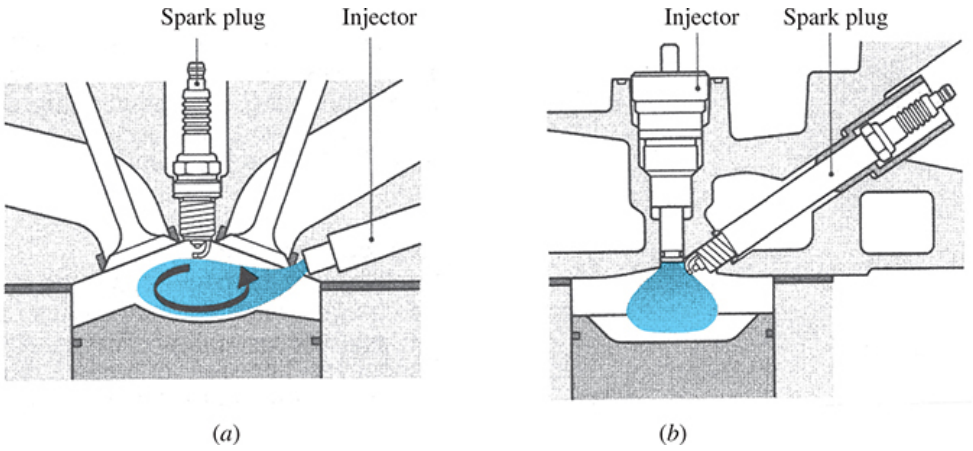


Figure 7.26 Two major categories of mixture preparation for ignition and combustion used in direct-injection stratified-charge SI engines: (*a*) widely spaced injector and spark plug (wall- and air-guided); (*b*) closely spaced injector and spark plug (spray-guided).[7]

Figure 7.27 shows four different examples of production wall- and air-guided, widely spaced, stratified DI engine concepts. One can see how the piston crown shape, air flow, and fuel spray characteristics all combine to enable a stable (fast and repeatable) spark-ignited combustion process.

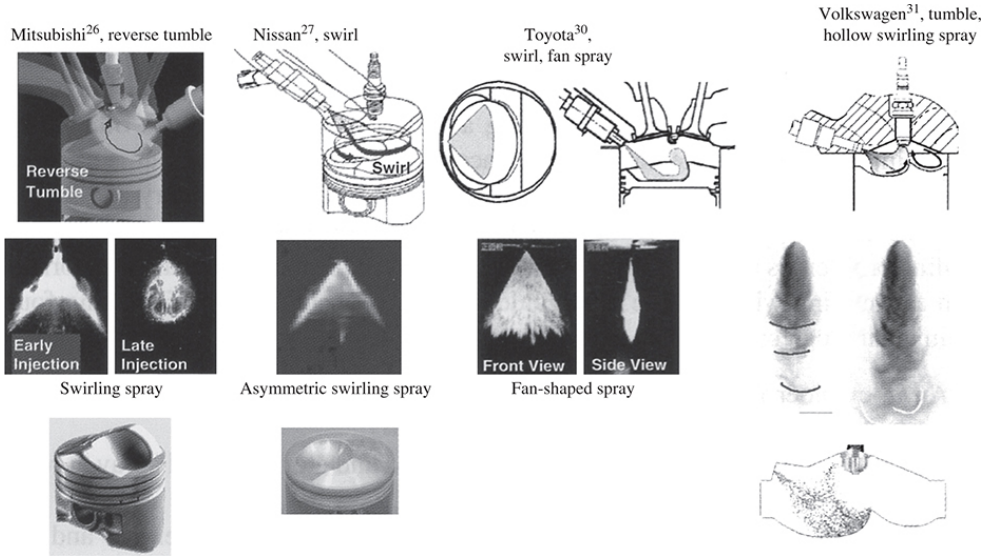


Figure 7.27 Examples of stratified air- and wall-guided gasoline direct-injection engines showing their combustion chamber geometries, airflow patterns, and fuel spray coinfigurations.[26, 27, 29, 30, 31]

Note that with lean operation at part-load, the engine intake will be less throttled than with stoichiometric operation, at a given torque. While slightly less fuel is required to produce the torque when lean because the engine is more efficient, significantly more airflow is required. Thus the intake manifold pressure when operating lean at a given torque is higher, and the pumping work (negative at part-load) is reduced. This is one of several factors that make lean engines at part-load more efficient.

An important feature of DI mixture preparation is the charge cooling that occurs due to the in-cylinder evaporation of the fuel droplets as the fuel spray moves through the chamber. By drawing the thermal energy required to vaporize the liquid fuel from the in-cylinder gases, these gases are cooled significantly. If the liquid fuel is at a lower temperature than the in-cylinder charge, then there is a fuel heating process involved, as well, but that produces a relatively small change in charge temperature. The mixture temperature change produced by the vaporization of liquid fuel drops embedded in an air plus residual gas mass (with both fuel and charge initially at the same temperature) is given by:

$$\Delta T = \frac{m_f}{(m_a + m_f)} \frac{h_{fg}}{c_p} (1 - x_r) \tag{7.14}$$

where liquid and vapor fuel heating (or cooling)—a less than 10% effect—has been neglected. Values for ΔT for gasoline, ethanol, an ethanol water mixture, and methanol are given in Table 7.4. These charge cooling effects are substantial: approximately 20 deg C for gasoline and 100 deg C or so for the alcohols. Adding 20% water to ethanol, using an expanded version of Eq. (7.14) , with h_{fg} for water 2442 kJ/kg, increases the ethanol evaporative cooling impact by about 50%.

TABLE 7.4 Evaporative cooling effects of fuel drops in stoichiometric mixtures of liquid SI engine fuels

Fuel	$(A/F)_s$	h_{fg} kJ/kg	$h_{fg}/Q_{HV} \times 10^3$	ΔT cooling deg. C*
Gasoline	14.6	350	8.0	19
Ethanol	9.0	910	30	82
Ethanol/water	20% (H_2O)	2442 (H_2O)	52	123
Methanol	6.47	1100	49	132

*From Eq. (7.14): c_p (mixture) $\approx$ 1.06 kJ/kg·K, $x \approx 0.05$.

Charge cooling during the intake process affects the amount of air inducted by increasing the density of the in-cylinder charge. Figure 7.28 shows the effect of several variables on the charge cooling temperature difference (at TC, at the end of the compression stroke), and the change in specific volume (inverse of the density) of the in-cylinder mixture at the end of intake. The solid arrow shows the spread from all the fuel vaporizing off the combustion chamber (or intake port and valve walls) at the top, to 100% of the fuel evaporated by heat transfer with the air.[f] Experimentally based efforts to quantify actual charge cooling with various fuels indicate that the fraction of the maximum or ideal ΔT shown in Table 7.4 achieved is above 80% for gasoline DI engine operation (in a turbocharged engine operating at stoichiometric) and 60 to 70% for E85 DI operation. [32] This DI fuel-evaporation charge cooling effect also impacts the onset of knock, and therefore the engine's compression ratio. Lower charge temperatures at the start of compression lead to lower unburned mixture temperatures at spark and at the crank angle of maximum cylinder pressure—the mixture conditions at which combustion onset and knock onset occur, respectively. Thus the onset of knock is delayed, and a higher engine compression ratio and engine turbocharging with higher boost can be used to increase engine output and, with engine downsizing, increase efficiency. Note also that use of alcohol and alcohol/gasoline blends give significant additional knock suppression beyond gasoline alone, an approach that can be realized with two separate tanks on the vehicle, or with the separation of the high octane fuel species (e.g., ethanol) through on-board separation of the fuel into higher and lower octane components.

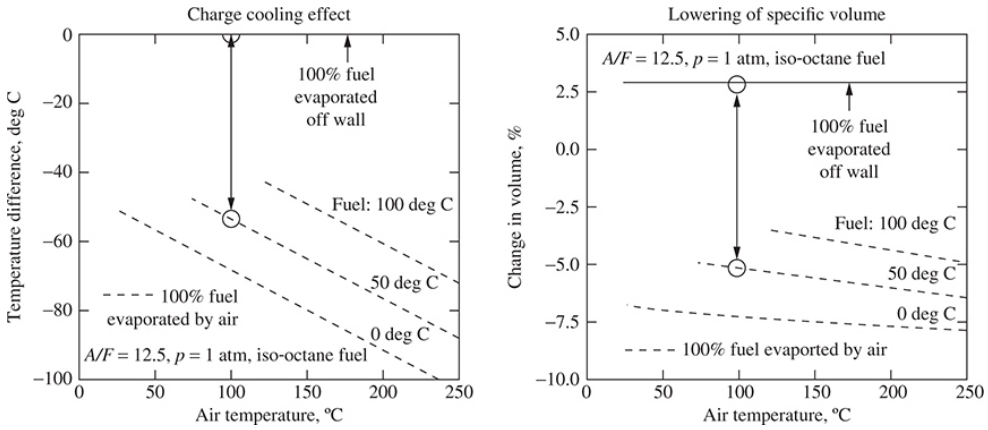


Figure 7.28 Mixture temperature reduction (at TC of compression), and decrease in the in-cylinder gas-mixture specific volume at the end of the intake process, as a function of initial air and fuel temperature, for 100 and zero percent fuel evaporation off the walls. $A/F = 12.5$, $p = 1$ atm, isooctane fuel.[33]

Note that when fuel evaporation within the cylinder occurs during the closed valve portion of the compression stroke, its impact on charge temperature increases. Fuel that then evaporates, e.g. (Δm_f), causes a temperature change proportional to ($\Delta m_{f,v} h_{fg}/ c_v$) rather than ($\Delta m_{f,v} h_{fg}/ c_p$) which quantifies the impact at constant pressure during the intake process. The ratio of c_p / c_v for the air, residual, fuel vapor charge is about 1:35.

7.7.3 DI Engine System and Components

Figure 7.29 shows a schematic of the overall engine management system for a direct-injection turbocharged gasoline engine.[34] Air enters the system via the air flow meter and turbocharger compressor, upper right. Exhaust gases leave the engine, moving right from the center, and exit via the turbine/wastegate combination, oxygen sensors, and catalysts at lower right. The mixture preparation components determine the air mass flow (compressor-intercooler/manifold pressure, electronic throttle, manifold/port swirl control valve, cam phaser for intake valve timing control) and the fuel flow (in-tank fuel pump, fuel rail, high-pressure pump, fuel-injection pressure-regulator, direct fuel injector). The engine's electronic control unit

determines the fuel mass required based on the desired torque and the corresponding required air mass (based on the intended stoichiometry of the mixture—stratified or homogeneous). The exhaust gas oxygen sensors monitor the exhaust gas mixture's relative air/fuel ratio (normally stoichiometric or lean if operating stratified). The boost pressure required for the airflow is controlled by the wastegate (through the amount of exhaust gas flow it bypasses around the turbine). The mixture preparation management system in these automotive applications, therefore, sets the fuel and air charge required for the desired engine output, generates the necessary fuel injection pressure, defines the start of injection, meters the required fuel quantity for that accelerator pedal and throttle setting, and introduces that fuel into the cylinder in the appropriate way. This sets up the fuel, air, residual gas (and EGR, not shown in Fig. 7.29) mixture within the cylinder both overall and especially at the spark location to achieve a stable low-emission combustion process. [7]

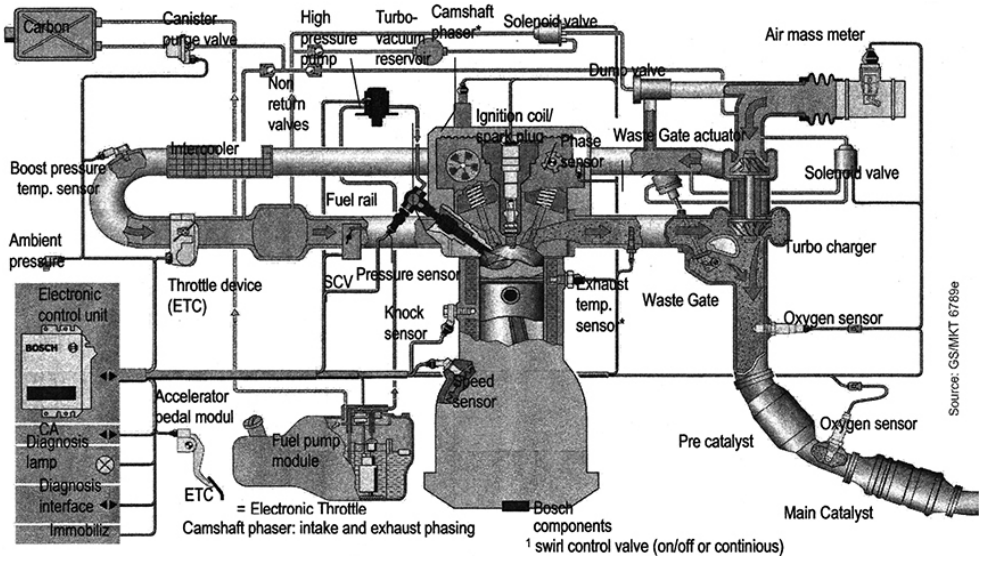


Figure 7.29 Layout schematic of DI engine management system.[34]

Figure 7.30 shows the normal mixture preparation objectives of this DI management system on an engine load and speed map for stratified DI operation and lean overall, at part-load and low- to mid-speed, and homogeneous mode operation at higher loads. The vertical scale is load or bmep or normalized torque. The relative air/fuel ratio λ is shown in each region. While the exact boundaries between these regions may vary, engine

concept to engine concept, this use of stratified and homogeneous operation, the values of λ and EGR in each mode, and the reasons for and impacts of these choices, are typical.

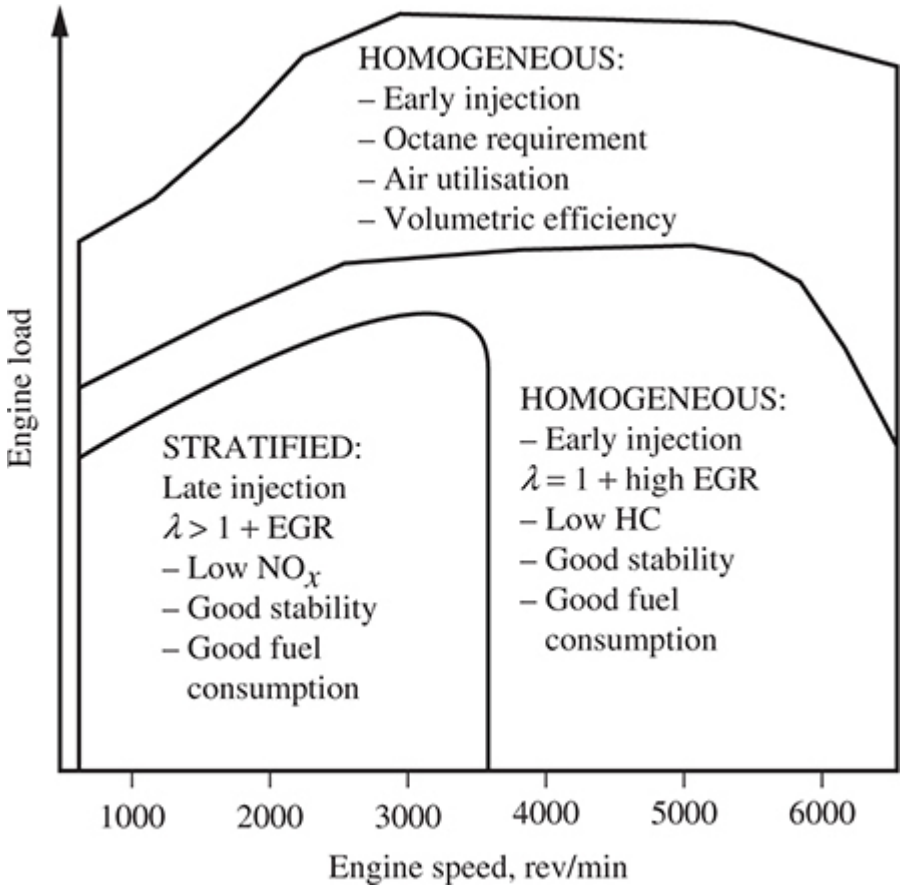


Figure 7.30 Typical stratified GDI engine operating map showing the different operating regimes.[35]

Nozzle Designs and Spray Characteristics

Fuel injectors have been specifically developed for the higher pressures required for GDI spark-ignited engines. The injected liquid fuel must be atomized into small droplets and appropriately dispersed throughout the appropriate region of the cylinder or combustion chamber. For homogeneous operation, a uniform distribution within the cylinder is desired: for stratified operation the spray must provide a readily ignitable mixture in the vicinity of the spark plug, and within a restricted volume. A close-to-stoichiometric mixture region is necessary for stable ignition and combustion. An example of a swirl-type injector, a commonly used high-pressure DI injector, is shown

in Fig. 7.31 *a* along with the signals that activate the fuel injection, Fig. 7.31 *b*. The fuel is forced by the swirl guide to rotate about the needle axis so that it exits the nozzle with a swirling motion. This use of a swirling motion decreases the drop size and helps disperse the spray. Such a swirl-type injector was used in two of the stratified DI engines shown in Fig. 7.27. The quantity of fuel injected depends on the difference between the fuel rail pressure and the in-cylinder gas pressure, and on the needle open time. A short needle opening and closing time is necessary for the desired metering precision. To achieve this, the injector is triggered by a digital signal, which the injector driver stage uses to activate the injector armature and needle, with a high initial current pulse to open the injector rapidly. The modest delays involved, and the needle opening and closing times, are included in the control logic.

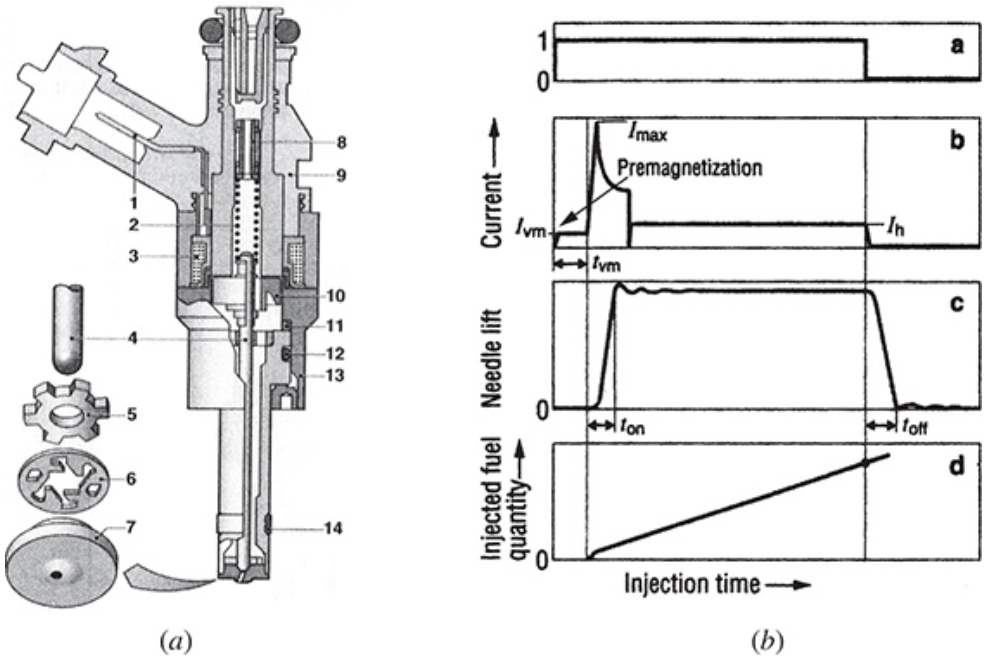


Figure 7.31 (*a*) High-pressure DI fuel injector layout diagram: (1) Electrical connection; (2) Spring; (3) Coil; (4) Nozzle needle; (5) Guide washer; (6) Swirl plate; (7) Seat washer; (8) Adjusting nozzle; (9) Housing; (10) Solenoid armature; (11) Shim; (12) O-ring; (13) Valve housing; (14) Teflon ring. (*b*) Signal profiles for activating the injector: (a) Triggering signal calculated by ECU; (b) Injector current; (c) Needle lift; (d) Amount of fuel injected. [7] (*Courtesy Robert Bosch GmbH and SAE.*)

Other types of injector designs are used. Multi-hole nozzles and outward-opening pintle nozzles are two significantly different injector designs and are illustrated in Fig. 7.32. Also shown in that figure are photographs of the fuel sprays each of these injectors—swirl-type, outward opening pintle, and multi-hole nozzles—produce. Swirl-type fuel injectors have been widely used in stratified systems where wall and flow spray guiding are utilized in combination. Variations in injection pressure cause the dispersion angle of the spray to change which can prove useful in transitioning from the narrower spray required for stratified operation to the wider spray required for homogeneous operation. Multi-hole nozzles, and outward-opening pintle nozzles, produce clearly defined spray boundaries (Fig. 7.32) and thus are good candidates for spray-guided stratified DI systems. Figure 7.33 illustrates the injector, fuel spray, and spark plug locations for a multi-spray nozzle, and an outward-opening swirl nozzle for such spray-guided DI mixture preparation and combustion systems. The spark-plug electrodes are located just within the fuel spray boundary where the local relative fuel-vapor air mixture is close to the stoichiometric composition so that ignition and flame development occur rapidly.

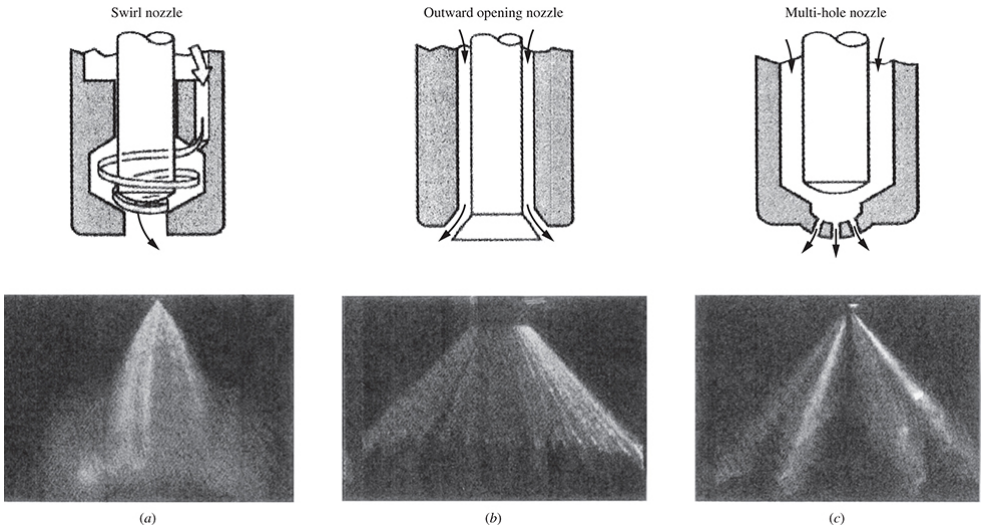


Figure 7.32 Three types of high-pressure injector nozzles and the spray patterns they generate: (*a*) swirl nozzle; (*b*) outward opening nozzle; (*c*) multi-hole nozzle.[29]

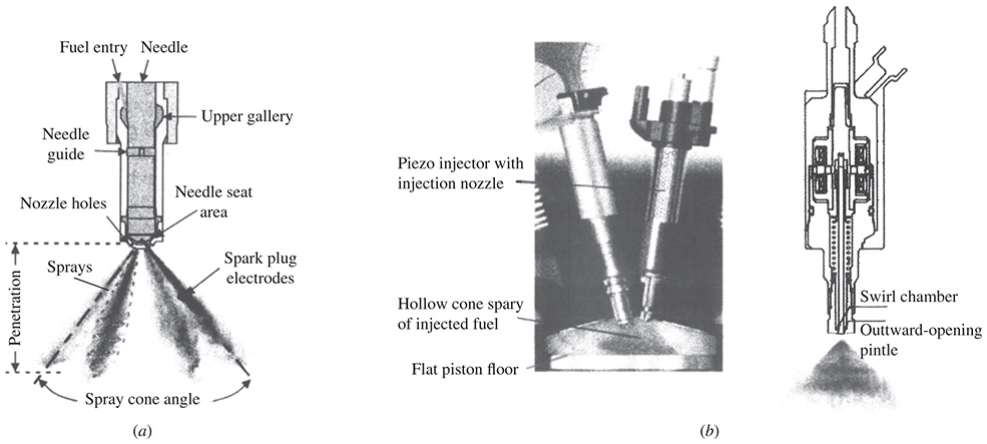


Figure 7.33 (*a*) Multi-hole injector for spray-guided gasoline direct-injection engine, with its several fuel sprays. (*b*) Outward-opening pintle injector with swirl chamber nozzle with its wide hollow-cone spray. [29]

7.8 EXHAUST GAS OXYGEN SENSORS

It is possible to reduce engine emissions of the three gaseous pollutants—hydrocarbons, carbon monoxide, and oxides of nitrogen—with a single catalyst in the exhaust system if the engine is operated at close to the stoichiometric air/fuel ratio. Such systems (a three-way catalyst and closed-loop sensor) are described in more detail in Sec. 11.6.3 . The engine operating air/fuel ratio is maintained close to stoichiometric through the use of a sensor in the exhaust system, which provides a voltage signal dependent on the oxygen concentration in the exhaust gas stream. This signal is the input to a feedback system, which controls the fuel and air feeds to the engine's cylinders.

The sensor, called an oxygen or lambda sensor—λ being the symbol used for the relative air/fuel ratio, Eq. (3.9) —uses as its basis an electrolytic cell with a solid electrolyte through which at elevated temperatures the current is carried by oxygen ions. The electrolyte is yttria (Y_2O_3) stabilized zirconia (ZrO_2) ceramic which separates two gas-containing regions (the exhaust manifold and the atmosphere), which have different oxygen partial pressures. The cell can be represented as a series of regions, separated by interfaces as follows:

Exhaust	Metal	Ceramic	Metal	Air
p'_{O_2}	M_e	ZrO_2, Y_2O_3	M_e	p''_{O_2}

p''_{O_2} is the oxygen partial pressure of the air (≈ 20 kN/m^2) and p'_{O_2} is the equilibrium oxygen partial pressure in the exhaust gases. At elevated temperatures, an electrochemical reaction takes place at the metal electrodes:

$$O_2 + 4M_e \rightleftharpoons 2O^{2-}$$

and the oxygen ions transport the current across the cell. The open-circuit output voltage of the cell V_0 can be related to the oxygen partial pressures p'_{O_2} and p''_{O_2} through the Nernst equation:

$$V_0 = \frac{RT}{4F} \ln\left(\frac{p''_{O_2}}{p'_{O_2}}\right) \tag{7.15}$$

where F is the Faraday constant. Equilibrium is established on the exhaust gas side electrode by the catalytic activity of the platinum catalyst on the metal electrode. The oxygen partial pressure in equilibrated exhaust gases decreases by many orders of magnitude as the equivalence ratio changes from 0.99 to 1.01, as shown in Fig. 7.34 *a*. Thus the sensor output voltage increases rapidly in this transition from a lean to a rich mixture at the stoichiometric point, as shown in Fig. 7.34 *b*. Since this transition is not temperature dependent, it is well suited as a sensor signal for a feedback system.

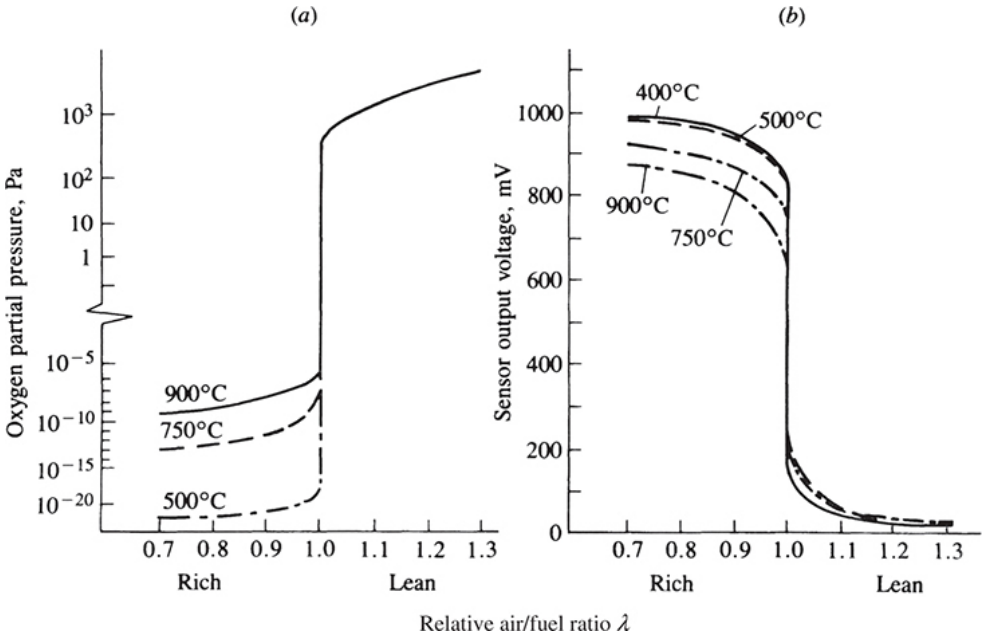


Figure 7.34 Oxygen-sensor electrolytic cell operating characteristics. Variation as a function of relative air/fuel ratio and temperature of: (*a*) oxygen partial pressure in equilibrated combustion products and (*b*) sensor output voltage. [36]

Figure 7.35 shows a cross-section drawing of simple lambda sensor, screwed into the wall of the exhaust manifold. The senor is electrically heated to provide rapid warm-up, following engine start-up. A close to the exhaust port location gives the shortest flow time from the fuel injector location (upstream of the engine) to the sensor—a delay time which is important in the operation of the feedback system. The sensor is made of ZrO_2 ceramic stabilized with Y_2O_3 to give adequate electrical conductivity. The inner and outer electrodes are 10-µm thick porous platinum layers to provide the required catalytic equilibration. The outer electrode which is exposed to the exhaust gases is protected against corrosion and erosion by a 10-µm spinal coat and a slotted shield. Air passes to the inner electrode through holes in the protective sleeve. The protective sleeve and housing are made from heat- and corrosion-resistant steel alloys. Such sensors were first developed for air/fuel ratio control close to the stoichiometric value.

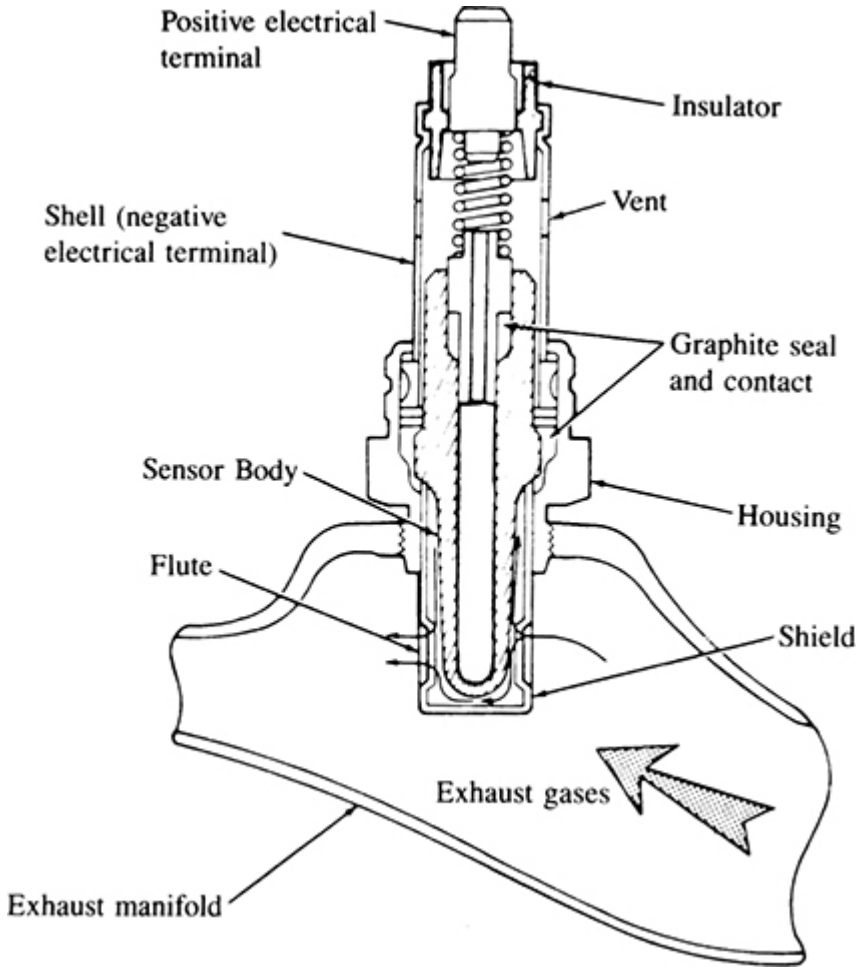


Figure 7.35 Cross-section drawing of exhaust oxygen (lambda) sensor.[37]

Wide-range lambda sensors are used when fuel lean or fuel rich measurement of exhaust gas relative air-fuel ratio is required. Wide-band sensors combine the conventional exhaust gas oxygen sensor described above, with a second such sensor operating as an oxygen pump. Figure 7.36 *a* shows a schematic of such a sensor and its electrical circuit. The oxygen pump cell (at the top) is paired with an exhaust gas oxygen concentration (Nernst) cell beneath it, on either side of a diffusion gap to which exhaust gas is supplied. If the exhaust gas in the diffusion gap is not stoichiometric, a current I_p is passed through the oxygen pump cell, driven by an applied voltage difference in the appropriate direction, to either transfer oxygen atoms out of or into the gas in the diffusion gap, depending on whether the gas is lean or rich, to bring it to the stoichiometric composition (as detected by the Nernst cell, relative to its reference-air cell just below it, with its

constant 21% oxygen concentration). An electrical heater is mounted underneath these sensing elements to heat and then control the temperature of the sensor. The response (pump current) to restore the exhaust gas relative air/fuel ratio λ to stoichiometric, versus λ, is shown in Fig. 7.36 *b*. Such wide-band exhaust gas oxygen sensors are important in any lean stratified DI engine concepts. They can also be used in diesels.

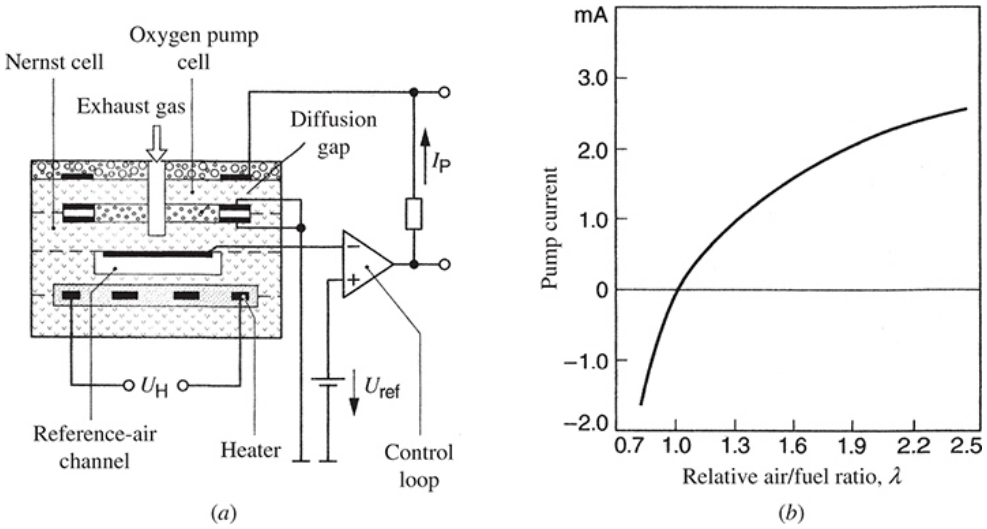


Figure 7.36 (*a*) Schematic of wide-range exhaust oxygen (lambda) sensor. (*b*) Sensor signal (pumping current) as a function of lambda. [7]

For closed-loop feedback control at close-to-stoichiometric engine operation, use is made of the sensor's low-voltage output for lean mixtures and high-voltage output for rich mixtures in the following way. A control voltage reference level is chosen at about the mid-point of the steep transition in Fig. 7.37 *b*. In the ECU the sensor signal is compared to the reference voltage in the comparator as shown in Fig. 7.37 *a*. The comparator output is then integrated in the integral controller whose output varies the fuel quantity linearly in the opposite direction to the sign of the comparator signal. There is a time lag τ_L in the loop composed of the transport time of fuel-air mixture from the point of fuel admission in the intake system to the sensor location in the exhaust, and the sensor and control system time delay. Because of this time lag, the controller continues to influence the fuel flow rate in the same direction, although the stoichiometric reference point has been passed, as shown in Fig. 7.37 *b*. Thus, oscillations in the relative

air/fuel or fuel/air equivalence ratio delivered to the engine exist even under steady-state conditions of closed-loop control. This behavior of the control system is called the *limit cycle*. The frequency f of this limit cycle is given by

$$f_{LC} = \frac{1}{4\tau_L} \tag{7.16}$$

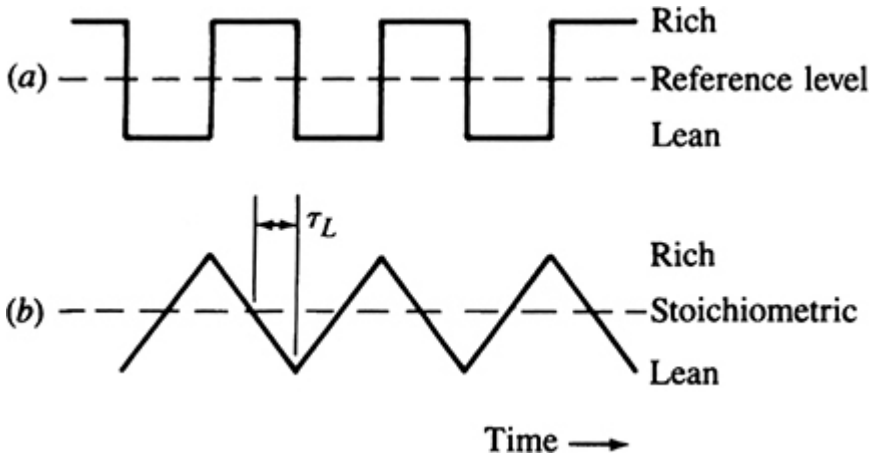


Figure 7.37 Operation of electronic control unit for closed-loop feedback: (*a*) sensor signal compared with reference level; (*b*) controller output voltage—the integrated comparator output.[38]

and the change in equivalence ratio, peak-to-peak, is

$$\Delta\phi = 2K_{\tau_L} \tag{7.17}$$

where K is the integrator gain (in equivalence ratio units per unit time).

Depending on the details of the three-way catalyst used for cleanup of the three pollutants CO, HC, and NO_x in the exhaust, the optimum average equivalence ratio may not be precisely the stoichiometric value. Furthermore, the reference voltage for maximum sensor durability may not correspond exactly to the stoichiometric point or the desired catalyst mean operating point. While a small shift ($\sim \pm 1\%$) in operating point from the stoichiometric can be obtained by varying the reference voltage level, larger shifts are obtained by modifying the control loop to provide a steady-state bias. One method of providing a bias— asymmetrical gain rate biasing—

uses two separate integrator circuits with different gain rates K^+ and K^- to integrate the comparator output, depending on whether the comparator output is positive (rich) or negative (lean). An alternative biasing technique incorporates an additional delay time τ_D so that the controller output continues decreasing (or increasing) even though the sensor signal has switched from the high to the low level (or vice versa). By introducing this additional delay only on the negative slope of the sensor signal, a net lean bias is produced. Introducing the additional delay on the positive slope of the sensor signal produces a net rich bias. [38]

Note that the sensor only operates at elevated temperatures. The sensor heater is used immediately after engine start-up to ensure that the feedback system operates as soon as possible to obtain the appropriate fuel-air mixture for satisfactory engine operation. Two lambda sensors are used in emission control systems with modern three-way catalysts, which contain *oxygen storage capability* through incorporating ceria onto the catalyst washcoat (see Sec. 11.6.3). A sensor upstream of the catalyst provides feedback control for the engine's mixture preparation system. The oscillations deliberately induced in the relative air/fuel ratio around the stoichiometric value (of about ± 0.3% of lambda at close to 1 Hz frequency), in the fuel-air mixture entering the engine, are used to ensure there is always sufficient oxygen stored in the catalyst to offset or absorb any random short-term rich or lean mixture excursions. With its oxygen storage capacity still effective, the three-way catalyst will absorb these fluctuations in λ and the catalyst exit burned gases will have an almost constant stoichiometric air/fuel ratio. The second oxygen sensor is thus positioned downstream of the catalyst to ensure its oxygen storage is functioning effectively since any oscillations are much reduced below the input λ oscillations. This second oxygen sensor is also used to improve the precision of the feedback signal obtained from the first sensor.

7.9 FUEL SUPPLY SYSTEMS

Port Fuel Injection

Fuel supply and delivery with intake-manifold injection have evolved through several stages. These are illustrated in Fig. 7.38. Two critical

requirements for fuel supply systems are ensuring that adequate fuel is available to the injectors for any conceivable change in demand, and enabling high injection precision, both cylinder-to-cylinder and cycle-by-cycle. In what has been the standard system, Fig. 7.38 *a*, under most operating conditions excess fuel was supplied to the fuel rail which operates at a constant pressure differential above the intake manifold pressure (controlled by the pressure regulator). Thus, adequate fuel was always available. The excess fuel supplied to the fuel rail was returned to the tank via a return line. One consequence of this fuel flow from the tank to the engine and back to the tank was fuel heating: the temperature of the fuel in the tank steadily rose creating problems with excessive hydrocarbon vapor pressures in the fuel supply system. Returnless fuel systems were developed (see Fig. 7.38 *b*), where only the fuel required by the injectors was supplied to the fuel rail at the regulated pressure, and there is no return line to the tank from the rail. The additional fuel conveyed by the electric fuel pump returns directly to the tank without flowing to the engine compartment and back. This returnless system reduces the in-tank fuel temperatures by about 10 deg. C thus reducing fuel vaporization by approximately one-third. [7]

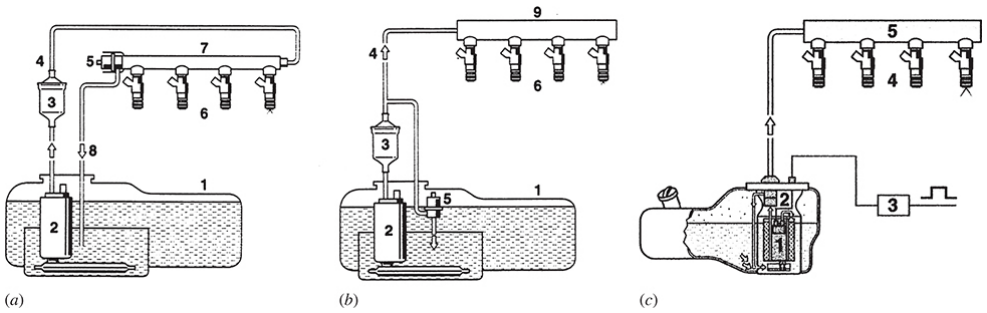


Figure 7.38 Fuel supply and delivery systems with manifold (port) fuel injection. (*a*) System with fuel return to tank. (*b*) Returnless system: (1) Fuel tank; (2) Electric fuel pump; (3) Fuel filter; (4) High-pressure line; (5) Pressure regulator; (6) Injectors; (7) Fuel rail, with return flow; (8) Return line; (9) Fuel rail, no return flow. (*c*) Demand-controlled system with no return line: (1) Electric fuel pump (with fuel filter); (2) Pressure relief valve and pressure sensor; (3) Timing module; (4) Injectors; (5) Fuel rail, no return flow. [2] (*Courtesy Robert Bosch GmbH and SAE.*)

The third intake-manifold injection system shown in Fig. 7.38 *c*, the demand-controlled system, further reduces fuel heating and improves the precision of fuel metering. Here, the fuel pump only supplies the quantity of fuel actually required by the engine, and controls the fuel rail pressure by means of a closed control loop in the engine control unit. The fuel quantity the pump delivers is controlled by varying the pump's operating voltage. Adjustments which vary the fuel rail pressure can be used to aid with cold and hot engine starting, and to increase the dynamic range of the injector system by raising the pressure difference across the injector and thereby increasing the fuel flow rate (e.g., when the engine is operating turbocharged). Thus, the pressure difference across the injector is taken into account, along with injector open duration, in determining the quantity of fuel injected. This improves the precision of the fuel metering.

Direct Fuel Injection

With direct-injection spark-ignition engines, the liquid fuel (normally gasoline) must be injected at higher pressures since wide spray dispersion, deep spray penetration, and smaller drop sizes are required. Also, when operating stratified with late injection the in-cylinder pressure during injection can be several atmospheres. Note that the time available for injection is less than with port injection systems, so higher fuel flow rates are required.

The fuel system comprises a low-pressure system, which uses the same components and layout as the port-fuel-injection supply system, and a high-pressure system. The key elements of the high-pressure system are a high-pressure pump (5 to 15 MPa), a high-pressure accumulator (fuel rail), a high-fuel-pressure sensor, and a pressure control valve or limiter. Figure 7.39 shows one such system—a demand-controlled fuel supply and delivery system. Here the high-pressure pump only supplies the fuel rail with the amount of fuel the injectors require; there is no excess supply to be returned. A mechanical pressure-relief valve is incorporated in the fuel pump. An alternative is a continuous-supply system where the pump fuel delivery pressure cannot be adjusted. The pressure of the fuel not required for injection, or to maintain rail pressure, is controlled by the pressure-control valve and the excess returned to the low-pressure system. This approach requires additional engine power to compress the excess fuel to the primary pressure, and this additional compression and venting combined with fuel

circulating cause a rise in fuel temperature. The demand-control fuel system avoids these drawbacks. [7]

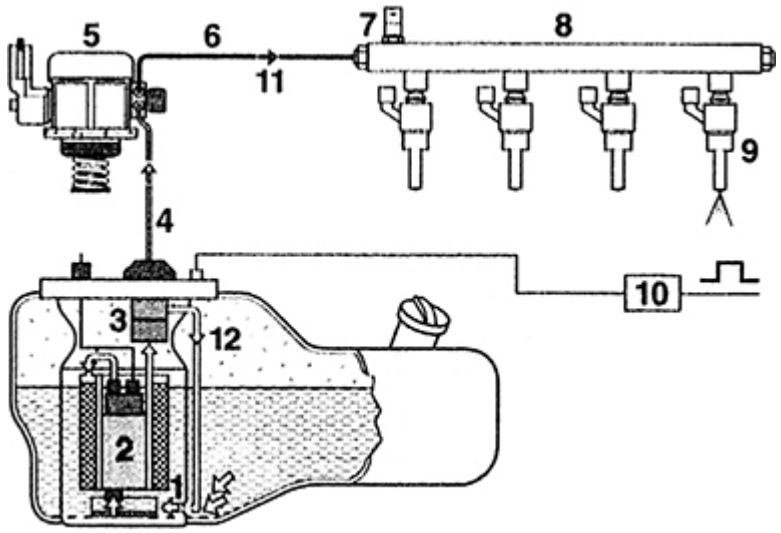


Figure 7.39 Fuel supply and delivery with demand-controlled direct fuel injection system: (1) Suction jet pump; (2) Electric fuel pump with filter; (3) Pressure sensor, and relief valve; (4) Low-pressure line; (5) Demand-controlled high-pressure pump, fuel-supply control valve, pressure-limiting valve; (6) High-pressure line; (7) High-pressure sensor; (8) Fuel rail; (9) High-pressure injectors; (10) Electric pump control; (11) Fuel flow direction.[7] (*Courtesy Robert Bosch GmbH and SAE.*)

7.10 LIQUID PETROLEUM GAS AND NATURAL GAS

These two fuels are quite widely used in spark-ignition engines. They can be used in vehicles dedicated to each of these fuels: they can be used in dual fuel vehicles where the LPG or NG fuel system is added in parallel to the gasoline fueling system, and the driver can switch from one to the other.

Liquid Petroleum Gas

LPG is a mixture of propane and butane. It can be stored in a cylindrical tank in the liquid state under pressure (up to 27 bar). The liquid LPG fill volume is restricted to 80% of the tank volume to allow the liquid gas to expand at

higher temperatures. When used in engines, LPG is injected into the intake manifold in similar fashion to multipoint gasoline injection. A common fuel rail and the individual injectors inject the LPG either continuously at variable pressure, or intermittently. The liquid LPG may be vaporized in a pressure regulator prior to injection, or may be injected as a liquid and then vaporizes in the intake manifold. The fuel system uses an exhaust gas oxygen sensor to maintain close-loop stoichiometric mixture control, to enable use of a three-way catalyst. Because its fuel molecules (propane and butane) are smaller than those typical of gasoline (average C_7H_{13}), the LPG fuel vapor displaces more air in the fuel-air mixture (see Fig. 6.3) and, accordingly, the engine's maximum power and torque are reduced by about 5%. Due to its higher H:C ratio, use of LPG produces about 10% less CO_2 emissions than equivalent use of gasoline. Hydrocarbon emissions with LPG are lower than equivalent gasoline-engine HC emissions.

Natural Gas

Natural gas is used as a spark-ignition engine fuel, both in vehicles as an alternative to gasoline, and in buses and larger engines for power generation as an alternative to diesel. Its main component is methane, CH_4, whose concentration can vary from about 80 to close to 100%. It is usually stored on the vehicle as compressed gas in high pressure (up to 700 bar) in cylindrical reinforced tanks. While it can be stored cryogenically, that has not yet proved practical. As compressed gas, its stored energy density in an appropriate fuel tank is about four times lower than that of liquid hydrocarbons. Natural gas engine-fuel-systems operate similarly to those used in multipoint-injected gasoline engines. The NG is usually injected into the intake manifold. A low-pressure common rail supplies the injector valves that inject intermittently. Due to its displacement of air in the fuel-air mixture, NG-powered engines have about 10 to 15% lower output than when operated (naturally-aspirated) with gasoline. However, it has a high antiknock octane rating, which as a stand-alone fuel would allow significantly higher engine compression ratios (about 13:1). Variations in NG composition (methane is the high antiknock component) in practice reduce this benefit. Compared to gasoline engines, natural gas engines have lower CO_2 emissions by 20 to 30% due to its lower C:H ratio than gasoline (by about a factor of 2). Its air pollutant emissions can be made very low. Methane, a major component of

the hydrocarbon emissions, is not photochemically active in the atmospheric smog chemistry though it is one of the significant greenhouse gases.

PROBLEMS

7.1 The relative air/fuel ratio and the EGR rate in a conventional spark-ignition engine varies from no load (idle) to full load, at a fixed engine speed, as shown in Fig. 7.1. (By load is meant the percentage of the maximum brake torque at that speed.) Using formats similar to those shown, draw *carefully proportioned qualitative* graphs of the following parameters versus load (0 to 100%):

Combustion efficiency, η_c

Gross indicated fuel conversion efficiency, $\eta_{f,ig}$

Gross indicated mean effective pressure, imep $_g$

Brake mean effective pressure, bmep

Mechanical efficiency, η_m

Indicate clearly where the maximum occurs if there is one, and where the value is zero or unity or some other obvious value, if appropriate. Provide a *brief justification* for the shape of the curves you draw.

7.2 The four-cylinder spark-ignition engine shown in the figure uses an oxygen sensor in the exhaust system to determine whether the exhaust gas composition is lean or rich of the stoichiometric point, and a port fuel injection system with feedback to maintain engine operation close to stoichiometric. However, since there is a time delay between a change in the fuel/air ratio at the injector location and the detection of that change by the sensor (corresponding to the flow time between the injector and the sensor), the control system shown results in oscillations in fuel/air ratio about the stoichiometric point.

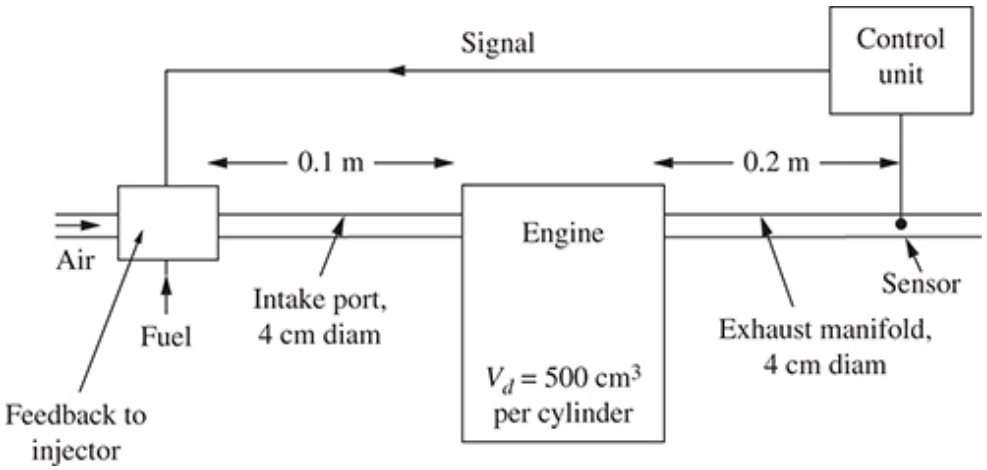
(a) Estimate the average flow time between the injector and the sensor at an engine speed of 2000 rev/min.

(b) The sensor and control unit provide a voltage V of $+V_s$ volts when the fuel/air equivalence ratio ϕ is less than one and a voltage of $-V_s$ volts when ϕ is greater than one. The feedback injection system provides a fuel/air ratio (F/A) given by

$$\left(\frac{F}{A}\right)=\left(\frac{F}{A}\right)_{t=0}(1+CVt)$$

where t is the time (in seconds) after the voltage signal last changed sign, (F/A) $_{t=0}$ is the fuel/air ratio at the injector at $t = 0$, and C is a constant. Develop carefully proportioned quantitative sketches of the variation in the fuel/air ratio at the injector and at the exhaust sensor, with time, showing the phase relation between the two curves. Explain briefly how you developed these graphs.

(c) Find the value of the constant C, in volts $^{-1}$ seconds^{-1} (the feedback system gain), such that (F/A) variations about the stoichiometric value do not exceed ± 10% for $V_s = 1$ V.



7.3 In many spark-ignition engines, liquid gasoline is added to the inlet air upstream of the engine. With port fuel-injection the gasoline is injected into the intake port using an electronically controlled fuel injector. The intake port and valve are warmed by the coolant to ensure that under steady-state conditions the fuel is essentially vaporized before the mixture enters the cylinder.

(a) At normal wide-open-throttle operating conditions, in a four-stroke cycle 1.6 dm^3 displacement four-cylinder engine, at 2500 rev/min, the temperature of the entering air is 40°C. The heart of vaporization of the fuel is 350 kJ/kg and the rate of heat transfer to the intake mixture if 1.4 kW. Estimate the temperature of the inlet mixture as it passes through the inlet valve, assuming that the fuel is fully vaporized. The volumetric efficiency is 0.85. The air density is 1.06 kg/m^3 and c_p for air is 1 kJ/kg·K. You may neglect the effects of the heat capacity of the liquid and vapor fuel.

(b) With port electronic fuel-injection systems, the fuel is only 90% vaporized prior to entering the cylinder. Estimate the mixture temperature as it passes through the inlet valve with the PFI system, assuming that the air temperature leaving the intake manifold is still 40°C and that the fuel that is vaporized draws the heat required from the intake port walls.

(c) With direct injection of gasoline into the cylinder during the intake stoke, all the thermal energy required to vaporize the fuel is drawn from the in-cylinder air plus residual gas. If the residual fraction at these conditions is 5% and the entering air temperature is 40°C, estimate the in-cylinder charge temperature after all the directly injected gasoline has vaporized at the end of the intake process. You will need to estimate the residual gas temperature.

(d) Briefly explain the implications of the above three fuel injection approaches on the engine's torque at these conditions.

7.4 With multipoint port fuel injection, the fuel flow rate is controlled by the injection pulse duration. If each injector operates continuously at the maximum rated power point (wide-open throttle, $A/F = 12$, 5500 rev/min) of an automobile spark-ignition engine, estimate approximately the injection pulse duration (in crank angle degrees) for the same engine at idle. Idle conditions are: 700 rev/min, 0.3 atm inlet manifold pressure, stoichiometric mixture.

7.5 The fuel-air cycle results indicate that the maximum imep is obtained with gasoline-air mixtures at equivalence ratios of about 1.0. In practice, the maximum wide-open-throttle power of a spark-ignition engine at a given speed is obtained with an air/fuel ratio of about 12. The vaporization of the additional gasoline lowers the temperature of the inlet air and the richer mixture has a lower ratio of specific heats γ_u during compression. Estimate approximately the change in mixture temperature due to vaporization of the

additional fuel used to decrease A/F from 14.6 (an equivalence ratio of 1.0) to 12.2 in the intake system, and the combined effect of vaporization and lower γ_u on the unburned mixture temperature at WOT when the cylinder pressure is at its peak of 40 atm. (The principal effect of the richer mixture is its impact on knock.)

7.6 **(a)** Plot dimensionless throttle plate open area $4A_{th}/(\pi D^2)$ as a function of throttle plate angle Ψ. Assume $\Psi_0 = 10°$, D (throttle bore diameter) = 57 mm, d (throttle shaft diameter) = 10.4 mm. What is the *throttle plate* area?

(b) Estimate the average velocity of the air flowing through the throttle plate open area for $\Psi = 26°$ at 3000 rev/min and $\Psi = 36°$ at 2000 rev/min. Use the relationship between Ψ, engine speed, and inlet manifold pressure given in Fig. 7.22. Assume a discharge coefficient $C_D = 0.8$.

(c) For the throttle of part (*a*), estimate and plot the total force on the throttle plate and shaft, and the force parallel and perpendicular to the throttle bore axis (i.e., in the mean flow direction and normal to that direction) as a function of throttle angle at 2000 rev/min. Again use Fig. 7.22 for the relationship between Ψ and inlet manifold pressure.

7.7 For the engine and intake manifold shown in Fig. 7.19, estimate the ratio of the intake manifold runner cross-sectional area to $\pi B^2/4$ (B is the cylinder bore); the ratio of the length of the flow path from the intake manifold entrance to the inlet valve seat, to the bore; the ratio of the volume of each inlet port to each cylinder's displaced volume; and the ratio of the volume of each intake manifold runner to each cylinder's displaced volume. The cylinder bore is 89 mm.

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[a] Typical value only. Most gasolines have $(A/F)_s$, in the range of 14.4 to 14.7. $(A/F)_s$ could lie between 14.1 and 15.2.

[b] A three-way catalyst system, when operated with a close-to-stoichiometric mixture, achieves substantial reductions in NO_x, CO, and HC emissions simultaneously; see Sec. 11.6.2 .

[c] A detailed discussion of carburetor operation can be found in the first edition of this text.[1]

[d] Three-way catalyst technology can reduce the three gaseous gasoline engine air pollutants HC, CO, and NO_x to extremely low levels (≈ 98% removal) when the engine is operated stoichiometric

[e] See Sec. 8.4 .

[f] Note that with port fuel injection, if droplets enter the cylinder and evaporate by taking thermal energy from the in-cylinder gases there will be some charge cooling

CHAPTER 8

Charge Motion within the Cylinder

Gas motion within the engine cylinder is one of the major factors that controls the combustion process in spark-ignition (SI) engines, and the fuel-air mixing and combustion processes in diesel engines. It also has a significant impact on engine heat transfer. Both the bulk gas motion and the turbulence characteristics of the flow are important. The initial in-cylinder flow pattern is set up by the intake process. It may then be substantially modified during compression. This chapter reviews the important features of gas motion within the cylinder set up by flows into and out of the cylinder through valves or ports, and by the motion of the piston.

8.1 INTAKE-GENERATED FLOWS

The engine intake process governs many important aspects of the flow within the cylinder. In four-stroke cycle engines, the downward motion of the piston pulls air into the cylinder. The intake flow velocities scale with piston speed. For much of the intake stroke, the annular gap between the intake valve head and valve seat is the minimum area for the flow (see Sec. 6.3) so gas velocities through this gap are the highest velocities set up during the intake process. The gas issues from the valve opening into the cylinder as an annular conical jet and the radial and axial velocities in the jet are about 10 times the mean piston speed. Figure 8.1 shows the radial and axial velocity components close to the valve exit, measured during the intake process, in a motored model engine with transparent walls and single valve located on the cylinder axis, using laser doppler anemometry (see next section).[1] The jet

separates from the valve seat and head, producing shear layers with large velocity gradients which generate turbulence. This separation of the jet sets up recirculation regions beneath the valve head and in the corner between the cylinder wall and cylinder head.

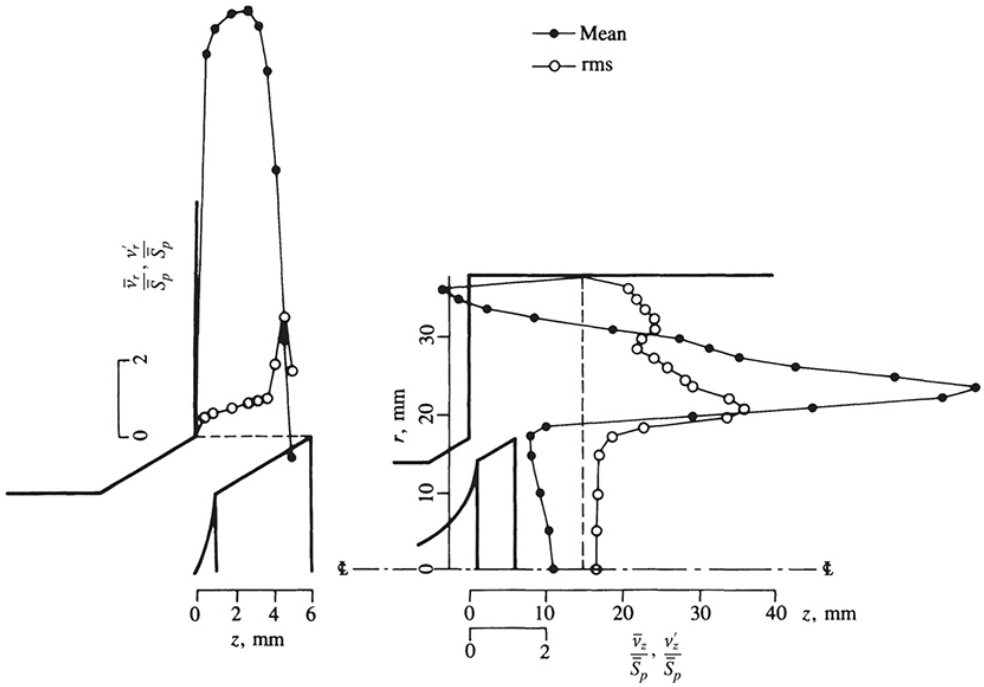


Figure 8.1 Radial mean velocity $\overline{v}_r$ and root mean square (rms) velocity fluctuations v_r' at the valve exit plane, and axial mean velocity $\overline{v}_z$ and rms velocity fluctuation v_z' 15 mm below the cylinder head, at 36° ATC in model engine operated at 200 rev/min. Valve lift = 6 mm. Velocities normalized by mean piston speed. [1]

The motion of the intake jet into the cylinder is shown in the schlieren photographs in Fig. 8.2 taken in a firing transparent engine. This engine had a square cross-section cylinder made up of two quartz walls and two steel walls, to permit easy optical access. The schlieren technique makes regions with density gradients in the flow show up as lighter or darker regions on the film. [2] The engine was throttled to one-half an atmosphere intake pressure, so the jet starts after the intake stroke has commenced, at 35° after top center (ATC), following the backflow of residual into the intake manifold. The front of the intake jet can be seen propagating from the valve toward the cylinder wall at several times the mean piston speed. Once the jet reaches the wall (θ

> 41° ATC), the wall deflects the major portion of the jet downward toward the piston; however, a substantial fraction flows upward toward the cylinder head. The highly turbulent nature of this intake jet is evident.

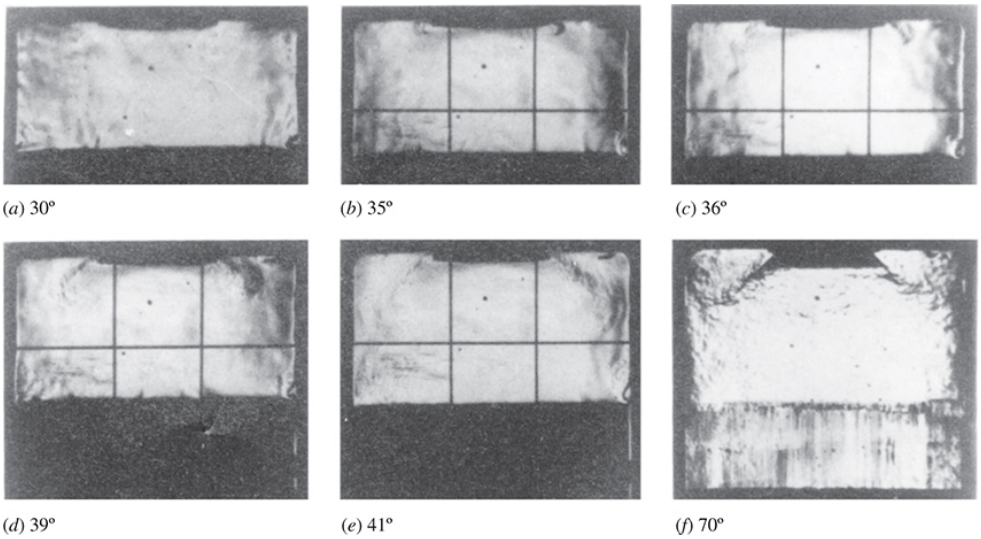


Figure 8.2 Sequence of schlieren photographs of intake jet as it develops during the intake stroke. Numbers are crank-angle degrees after TC. [2]

The interaction of jet with the wall produces large-scale rotating flow patterns within the cylinder volume. These are easiest to visualize where the engine geometry has been simplified so the flow into the cylinder is axisymmetric. The photograph in Fig. 8.3 of a water analog of an engine intake flow was taken in a transparent model of an engine cylinder and piston. The valve is located in the center of the cylinder head, and the flow into the valve is along the cylinder axis. The experimental parameters have been scaled so that the appropriate dimensionless numbers which govern the flow, the Reynolds and Strouhal numbers, were maintained at typical engine values. The photograph shows the major features of the intake-generated flow in a thin illuminated plane through the cylinder axis. The streaks are records of the paths of tracer particles in the flow during the period the camera shutter is open. The bulk of the cylinder as the piston moves down is filled with a large ring vortex, whose center moves downward and remains about halfway between the piston and the head. The upper corner of the cylinder contains a smaller vortex, rotating in the opposite direction. These

vortices persist until about the end of the intake stroke, when they became unstable and break up. [3]

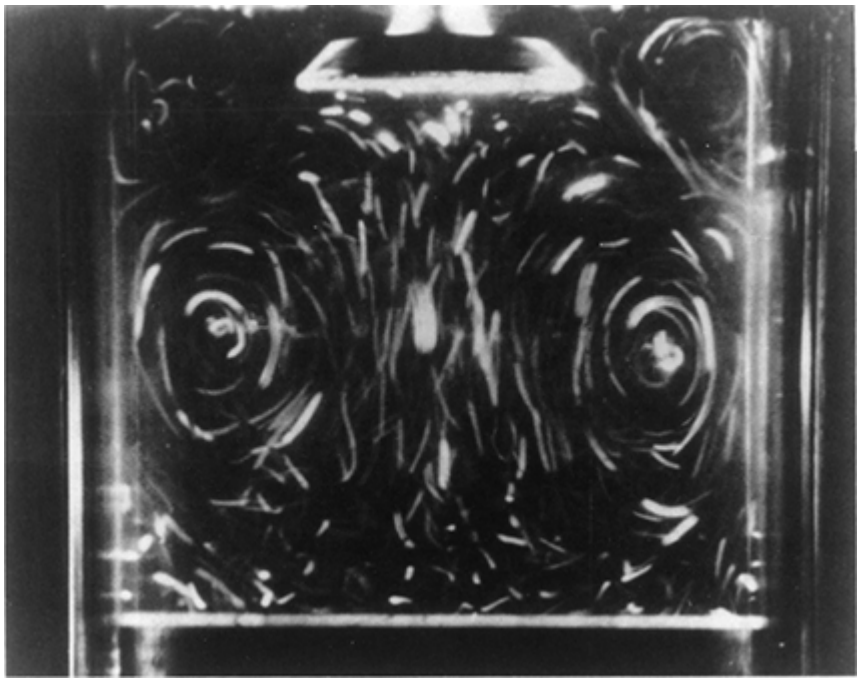


Figure 8.3 Large-scale rotating flow pattern setup within the cylinder by the intake jet. Photograph of streak lines in water flow into model engine cylinder with axisymmetric valve. [3]

With intake valve location and port geometry more typical of normal engine practice, and with two intake valves per cylinder, the intake-generated flow is more complex. However, the presence of these large-scale rotating flow patterns, produced by the airflow into the cylinder, can still be discerned. Such flows are inherent to the poppet valve geometry in the cylinder head: they bring into the cylinder significant fluid kinetic energy and create shear, both of which can generate turbulence. The intake valve(s) and port geometries can be designed to augment these jet-created flows by generating large-scale in-cylinder rotating flows. Two of these, swirl and tumble, are illustrated in Fig. 8.4 by the flow streak lines highlighted by flow tracers in water analog studies of real cylinder airflows. Swirl, Fig. 8.4 *a*, organized fluid rotation about the cylinder axis, is usually generated by bringing the air into the cylinder with net angular momentum about the cylinder axis. It is used in smaller higher speed diesel engines to promote

more rapid fuel-air mixing, and also in spark-ignition engines to increase the engine's combustion rate. Tumble, Fig. 8.4 *b*, is the name given to flow rotation about an axis orthogonal to the cylinder axis. It is generated by using closer to horizontal intake ports which direct the intake airflow across the cylinder head and down the far side of the liner as illustrated. Tumble is used to increase the kinetic energy stored in the in-cylinder charge during intake: the upward motion of the piston during compression breaks up this tumbling flow which generates additional turbulence, and as a result speeds up the spark-ignition engine combustion process. Tumble and swirl can occur together, or separately. These flows are discussed in more detail in Secs. 8.3 and 8.4.

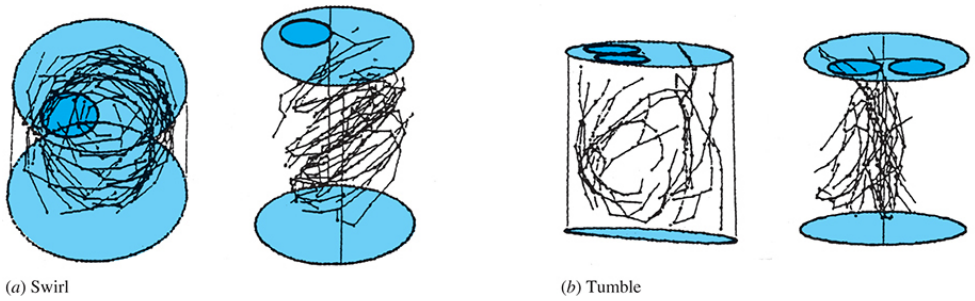


Figure 8.4 In-cylinder flow streaklines from tracers in water analog studies of (*a*) swirl and (*b*) tumble in-cylinder flows. With one intake port off axis (*a*), air enters the cylinder with net angular momentum about the cylinder axis, and a swirling in-cylinder flow is set up. With two intake ports which direct the air across the cylinder head and down the liner (*b*), a tumbling flow rotating about an axis orthogonal to the cylinder axis is created. [4,5]

In summary, the jet-like character of the four-stroke cycle intake flow, interacting with the cylinder walls and moving piston, creates large-scale rotating flow patterns within the cylinder. Additionally, large-scale flows such as swirl and tumble can be deliberately set up through use of specific port (and valve) geometries. The details of these flows are strongly dependent on the inlet port, valve, and cylinder head geometry. These flow structures are thought to become unstable, either late in the intake or during the compression process, and break down into more turbulent motions (less

so in the case of swirl). Such rotating flows are usually sensitive to variations in the flow: thus their breakup is likely to vary, cycle-by-cycle.

In two-stroke engines, the flow of air into the cylinder differs markedly from the flow generated by the four-stroke cycle engine intake process. This flow occurs while the piston is close to bottom center (BC). It is the key part of the cylinder scavenging process and overlaps the burned gas exhaust process. The entering airflow is driven through the transfer ports near the bottom of the cylinder liner which open as the piston uncovers them, by a pressure difference between the supplied air (which has been compressed prior to being fed to the transfer passages) and the cylinder. The airflow into the cylinder is not pulled by the downward motion of the piston, as occurs in the four-stroke cycle during its intake stroke. The air velocities generated in the air flowing into the two-stroke engine cylinder do not therefore scale with engine speed, except in two-stroke engines where air compression is achieved through use of a compressor or blower mechanically coupled to and driven by the engine's crankshaft. [6]

Figure 8.5 shows how the air velocity at the transfer or scavenge port varies during the scavenging process in a crankcase-compression loop-scavenged Schnürle-type single-cylinder 98 cm^3 displacement two-stroke cycle SI engine. The exhaust and scavenge port opening and closing angles are indicated. Also shown are the crankcase pressure p_c, the exhaust-port pressure p_e, their difference, and the scavenge-port pressure p_s, at wide-open-throttle conditions. The maximum air velocities at each of the three engine speeds shown are comparable in magnitude, though at the higher speeds the crank-angle extent of the blowdown process from the crankcase into the cylinder is longer. Pressure and flow oscillations due to compression and expansion waves are evident. It is the pressure difference, $p_c - p_e$, that largely drives the air into the cylinder in this type of two-stroke engine. At higher engine speeds there is less time available for the scavenging process; thus the pressure blowdown and duration of the airflow pulse occur over a longer crank-angle interval.

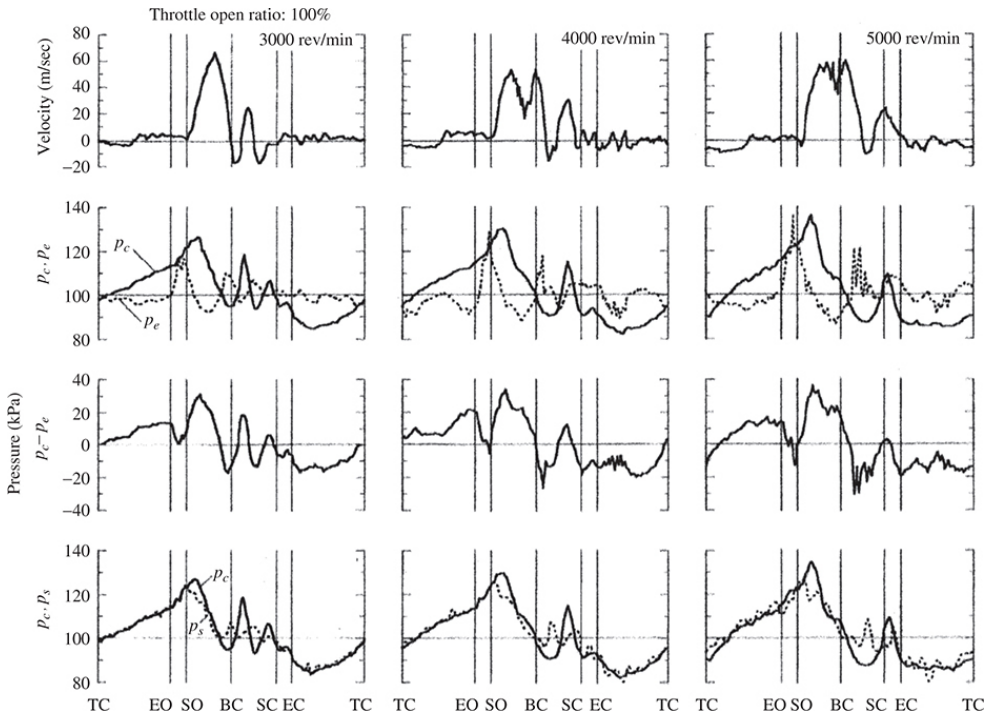


Figure 8.5 The effect of engine speed on the fresh-charge velocity at the scavenge port, and the pressure variation in the crankcase (p_c), the exhaust port (p_e), and scavenge port (p_s) at wide-open-throttle conditions. Engine details: two-stroke crankcase-compression loop-scavenged SI engine of Schnürle type, 98 cm ^{2} cylinder displaced volume, bore 5 cm, stroke 5 cm, compression ratio 6.59; EO 83° BBC; EC 82° ABC; SO 57° BBC; SC 60° ABC. [7]

At loads less than wide-open throttle in two-stroke SI engines, the amount of air entering the cylinder each cycle is reduced by throttling the airflow into the crankcase. At lower engine speeds, closing the throttle to 20% of its wide-open area does not significantly affect the mean entering air velocity. At high engine speed, the maximum velocity decreases by about 50% when the throttle is closed to 20% of its wide-open-throttle (WOT) flow area. Backflow from the cylinder into the transfer ports occurs at low engine speed during a large part of the gas exchange period, but only to a limited extent at high engine speed. At low speed, the time available for the crankcase to be filled is long enough at these throttle positions (some 20% of the fully open area) for the total cylinder trapped mass to be unaffected. The throttle open area must be further reduced to affect the trapped fuel-air mass. At high

speed, the velocity of the incoming charge through the intake system is high, and so is the pressure drop across the throttle. Consequently, at higher speeds, the throttle position affects the crankcase pressure and thus the mass of air trapped within the cylinder. [6]

These two-stroke transfer or scavenging port jet flows are usually symmetric as they enter the cylinder, and have velocities in the 50–100 m/s range and modest root-mean-square velocities (or turbulence intensities—see Sec. 8.2) or order 10% of the mean velocities. However, the interaction of these jets with each other and the combustion chamber walls produces an in-cylinder flow with strong asymmetry and high rms velocities of 30 to 40% of the mean values. As the strong inflow diminishes and the crankcase and cylinder pressure equilibrate, the scavenging-loop rotating (vortex) flow becomes evident (see Fig. 6.34), though the flow details are complex. As the rising piston compresses this scavenging-loop flow, it first speeds up and then breaks down into turbulence shortly before TC (similarly to the tumbling flow often produced in four-stroke cycle SI engines). In the crankcases-scavenged two-stroke engines described here, the gas exchange process is driven primarily by the blowdown of compressed air from the crankcase into the cylinder when the piston uncovers the transfer ports. Unlike four-stroke cycle engines where piston motion during the intake stroke pulls the air into the cylinder, and thus the velocity of the air entering the cylinder scales with piston speed, these two-stroke entering airflow velocities do not scale in any simple way with piston or engine speed. However, the charge motion produced by the piston during compression does scale with piston speed, so the mean piston speed $\overline{S}_p (= 2NS)$ is still useful as a normalizing velocity, provided we recognize that the normalized velocities will vary with engine speed and load (delivery ratio) as discussed above. At a typical part-load low-speed two-stroke SI engine operating condition (1600 rev/min, delivery ratio = 0.5), the largest velocities in the entering air jets are in the range of about 40–50 $40–50\overline{S}_p$ and the largest rms velocities are about $8–10\overline{S}_p$ and are found near the cylinder center. These values are large relative to four-stroke cycle intake-generated flow velocities and rms velocities, which typically are in the ranges of $3–10\overline{S}_p$ and $1–2\overline{S}_p$ respectively. In four-stroke engines with simple open chambers, the rms velocities (or turbulence intensities—see Sec. 8.2) at TC of the compression stroke are about $0.5\overline{S}_p$. When intake-generated tumble is used to enhance four-stroke cycle SI engine

combustion rates, the turbulence intensity (the major flow component driving the engine's flame propagation) rises to about $(0.8–1)\overline{S}_p$ at TC piston position. Such values are only slightly less than the rms velocities or turbulence intensities of about $(1–1.2)\overline{S}_p$ measured at TC in small two-stroke cycle SI engines. [8]

8.2 MEAN VELOCITY AND TURBULENCE CHARACTERISTICS

8.2.1 Definitions of Relevant Parameters

The charge motions into and within the engine cylinder are turbulent. In turbulent flows, the rates of transfer and mixing are several times greater than the rates due to molecular diffusion. This "turbulent diffusion" results from the local fluctuations in the flow field. It leads to increased rates of momentum, heat, and mass transfer, and is essential to the satisfactory operation of spark-ignition and diesel engines. Turbulent flows are always dissipative. Viscous shear stresses perform deformation work on the fluid, which increases its internal energy at the expense of its turbulent kinetic energy. Energy is required to generate turbulence: if no energy is supplied, turbulence decays. A common source of energy for turbulent velocity fluctuations is shear in the mean flow. Turbulence is rotational and is characterized by high fluctuating vorticity: these vorticity fluctuations can only persist if the velocity fluctuations are three dimensional. [9]

The character of a turbulent flow depends on its environment. In the engine cylinder, the flow involves a combination of turbulent shear layers, recirculating regions, and boundary layers. The flow is unsteady and may exhibit substantial cycle-to-cycle fluctuations. Both large-scale and small-scale turbulent motions are important factors governing the overall behavior of the flow.

An important characteristic of a turbulent flow is its irregularity or randomness. Statistical methods are therefore used to define such a flow field. The quantities normally used are: the mean velocity, the fluctuating velocity about the mean, and several length and time scales. In a steady

turbulent flow situation, the instantaneous local fluid velocity U (in a specific direction) is written:

$$U(t) = \bar{U} + u(t) \tag{8.1}$$

where $\bar{U}$, the mean velocity, is the time average of $U(t)$:

$$\bar{U} = \lim_{\tau \to \infty} \frac{1}{\tau} \int_{t_0}^{t_0+\tau} U(t)dt \tag{8.2}$$

The fluctuating velocity component u is characterized by its root mean square value, the turbulence intensity, u':

$$u' = \lim_{\tau \to \infty} \left(\frac{1}{\tau} \int_{t_0}^{t_0+\tau} u^2 dt \right)^{1/2} \tag{8.3a}$$

Alternatively,

$$u' = \lim_{\tau \to \infty} \left(\frac{1}{\tau} \int_{t_0}^{t_0+\tau} \left(U^2 - \bar{U}^2 \right) dt \right)^{1/2} \tag{8.3b}$$

since the time average of (u $\bar{U}$) is zero.

In engines, the application of these turbulence concepts is complicated by the fact that the flow pattern within the cylinder changes during the engine cycle. Also, while the overall features of the flow in each cylinder repeat each cycle, the details do not because the mean flow can vary significantly from one cycle to the next. There are both cycle-to-cycle variations in the mean or bulk flow at any point in the cycle, as well as turbulent fluctuations about that specific cycle's mean flow.

One approach used in quasi-periodic flows, such as that which occurs in the engine cylinder, is *ensemble-averaging* or *phase-averaging*. Usually, velocity measurements are made over many engine cycles, and over a range of crank angles. The instantaneous velocity at a specific crank-angle position θ in a particular cycle i can be written as

$$U(\theta, i) = \bar{U}(\theta, i) + u(\theta, i) \tag{8.4}$$

The ensemble- or phase-averaged velocity, $\bar{U}(\theta)$, is defined as the average of values at a specific phase or crank angle in the basic cycle. Figure 8.6 shows this approach applied schematically to the velocity variation during a two-stroke engine cycle, with small and large cycle-to-cycle variations. The ensemble-averaged velocity is the average over a large number of measurements taken at the same crank angle (two such points are indicated by dots):

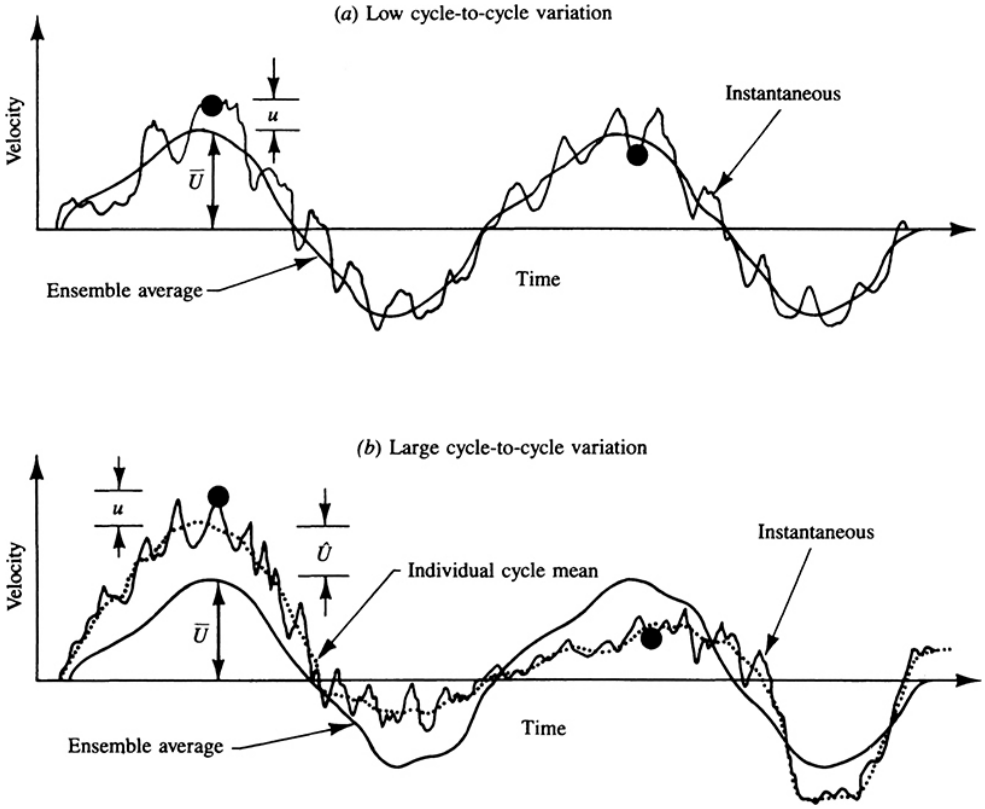


Figure 8.6 Schematic of velocity variation with crank angle at a fixed location in the cylinder during two consecutive engine cycles. Dots indicate measurements of instantaneous velocity at two specific crank angles. The ensemble- or phase-averaged velocity is obtained by averaging over a large number of such measurements shown as solid smooth line. Top graph (*a*): low cycle-to-cycle flow variations. Here the individual-cycle mean velocity and ensemble-averaged velocity are closely comparable. Bottom graph (*b*): large cycle-to-cycle variations. Here the individual-cycle mean velocity (dotted line) is different from the ensemble-averaged mean by $\hat{U}$. The turbulent fluctuation u is then defined in relation to the individual-cycle mean. [10]

$$\bar{U}_{EA}(\theta) = \frac{1}{N_c}\sum_{i=1}^{N_c} U(\theta,i) \tag{8.5}$$

where N_c is the number of cycles for which data are available. By repeating this process at many crank-angle locations, the ensemble-averaged velocity profile over the complete cycle is obtained.

The ensemble-averaged mean velocity at a given location is only a function of crank angle since the cyclic variation has been averaged out. The difference between the mean velocity in a particular cycle and the ensemble-averaged mean velocity over many cycles is defined as the cycle-by-cycle variation in mean velocity, $\hat{U}$:

$$\hat{U}(\theta,i) = \bar{U}(\theta,i) - \bar{U}_{EA}(\theta) \tag{8.6}$$

Thus the instantaneous velocity, given by Eq. (8.4) , can be split into three components:

$$U(\theta,i) = \bar{U}_{EA}(\theta) + \hat{U}(\theta,i) + u(\theta,i) \tag{8.7}$$

Figure 8.6 illustrates this breakdown of the instantaneous velocity into an ensemble-averaged component, an individual-cycle mean velocity, and a component which randomly fluctuates in time at a particular point in space in a single cycle. This last component is the conventional definition of the turbulent velocity fluctuations. Whether this differs significantly from the fluctuations about the ensemble-averaged velocity depends on whether the cycle-to-cycle fluctuations are small or large. The figure indicates these two extremes. [a]

In turbulent flows, a number of length scales exist that characterize different aspects of the flow behavior. The largest eddies in the flow are limited in size by the system boundaries. The smallest scales of the turbulent motion are limited by molecular diffusion. The important length scales are illustrated by the schematic of the jet issuing into the cylinder from the intake valve in Fig. 8.7. The eddies responsible for most of the turbulence production during intake are the large eddies in the conical inlet jet flow between the valve head and seat. These are roughly equal in size to the local jet thickness. This scale is called the *integral length scale,* l_I: it is a measure

of the largest scale structures in the turbulent flow. Velocity measurements made at two points separated by a distance x significantly less than l_I will correlate with each other; with $x >> l_I$, no correlation will exist. The integral length scale is defined, therefore, as the integral of the autocorrelation coefficient of the fluctuating velocity at two adjacent points in the flow with respect to the variable distance between the points, as shown in Fig. 8.8: that is,

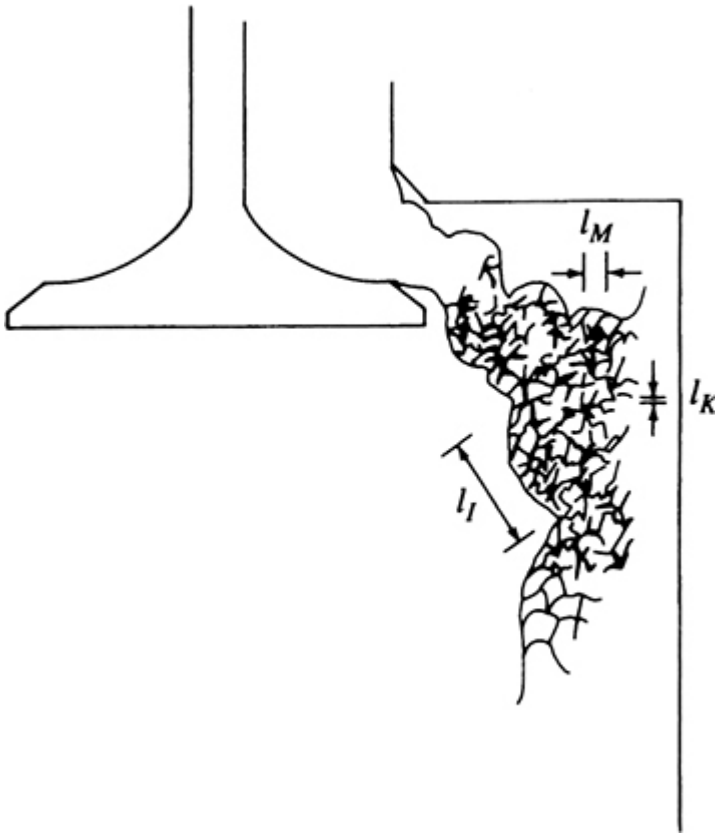


Figure 8.7 Schematic of jet created by the flow through the intake valve indicating its turbulent structure.[9, 10]

$$l_I = \int_0^\infty R_x dx \tag{8.8a}$$

where

$$R_x = \frac{1}{N_m - 1} \sum_{i=1}^{N_m} \frac{u(x_0)u(x_0 + x)}{u'(x_0)u(x_0 + x)} \tag{8.8b}$$

This technique for determining the integral scale requires simultaneous measurements at two points. Due to the difficulty of applying this technique in engines, most efforts to determine length scales have first employed correlations to determine the *integral time scale,* τI. The integral time scale of the turbulence is defined as a correlation between two velocities at a fixed point in space, but separated in time:

$$\tau_I = \int_0^\infty R_t dt \tag{8.9a}$$

where

$$R_t = \frac{1}{N_m - 1} \sum_{i=1}^{N_m} \frac{u(t_0)u(t_0 + t)}{u'(t_0)u'(t_0 + t)} \tag{8.9b}$$

and N_m is the number of measurements. Under conditions where the turbulence pattern is convected past the observation point without significant distortion and the turbulence is relatively weak, the integral length and time scales are related by

$$l_I = \bar{U}\tau_I \tag{8.10}$$

In flows where the large-scale structures are convected, τ_I is a measure of the time it takes a large eddy to pass between two points. In flows without mean motion, the integral time scale is an indication of the lifetime of an eddy. [10, 11]

Superposed on this large-scale flow is a range of eddies of smaller and smaller size, fed by the continual breakdown of larger eddies. Since the smaller eddies respond more rapidly to changes in local flow pattern, their characteristics are more likely to be isotropic (have no preferred direction) than are the large eddies, and have a structure like that of other turbulent flows. The dissipation of turbulence energy takes place in the smallest structures. At this smallest scale of the turbulent motion, called the *Kolmogorov scale,* the kinetic energy in these smallest flow structures is dissipated via molecular viscosity into thermal energy. If e is the energy dissipation rate per unit mass and v the kinematic viscosity, Kolmogorov length and time scales are defined by:

$$l_K = \left(\frac{\nu^3}{\varepsilon}\right)^{1/4} \qquad \tau_K = \left(\frac{\nu}{\varepsilon}\right)^{1/2} \tag{8.11}$$

The Kolmogorov length scale characterizes the size of the smallest eddies in the flow. The Kolmogorov time scale characterizes the momentum diffusion rate at this smallest structure level.

A third scale is useful in characterizing a turbulent flow. It is called the *microscale* (or *Taylor microscale*). The micro length scale l_M is defined by relating the fluctuating strain rate of the turbulent flow field to the turbulence intensity:

$$\frac{\partial u}{\partial x} \approx \frac{u'}{l_M} \tag{8.12}$$

It can be determined from the curvature of the spatial correlation curve at the origin, as shown in Fig. 8.8. [9, 10] More commonly, the micro time scale τ_M is determined from the temporal auto correlation function of Eq. (8.9) :

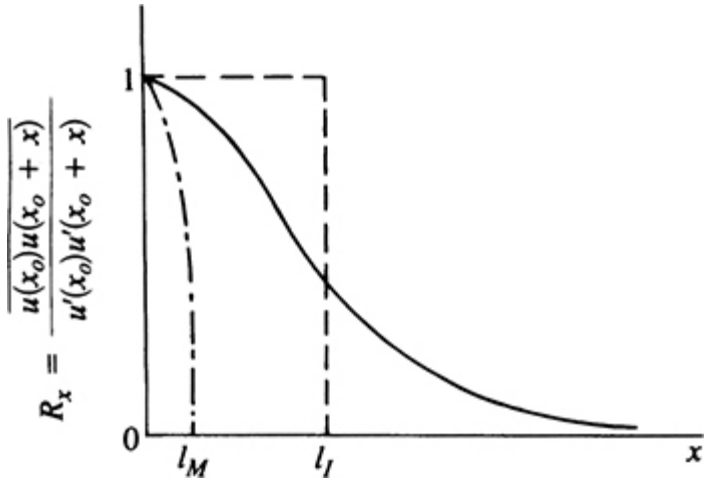


Figure 8.8 Spatial velocity autocorrelation R_x as a function of x, defining the integral length scale l_I and the micro length scale l_M.

$$\tau_M^{\,2} = -\frac{2}{(\partial^2 R_t / \partial t^2)_{t_0}}$$

For turbulence which is homogeneous (has no spatial gradients) and is isotropic (has no preferred direction), the microscales l_M and τ_M are related by

$$l_M = \bar{U}\tau_M \tag{8.13}$$

These different scales are related as follows. The turbulent kinetic energy per unit mass in the large-scale eddies is proportional to u'^2. Large eddies lose a substantial fraction of this energy in one "turnover" time l_I / u'. In an equilibrium situation the rate of energy supply equals the rate of dissipation:

$$\varepsilon \approx \frac{u'^3}{l_I}$$

Thus,

$$\frac{l_K}{l_I} \approx \left(\frac{u' l_I}{\nu}\right)^{-3/4} = \mathrm{Re}_T^{-3/4} \tag{8.14}$$

where Re_T is the turbulent Reynolds number, $u' l_I / \nu$.

Within the restrictions of homogeneous and isotropic turbulence, an energy budget can be used to relate l_I and l_M: [9]

$$\varepsilon = \frac{Au'^3}{l_I} = \frac{15\nu u'^2}{l_M^2}$$

where A is a constant of order 1. Thus,

$$\frac{l_M}{l_I} = \left(\frac{15}{A}\right)^{1/2} \mathrm{Re}_T^{-1/2} \tag{8.15}$$

These restrictions are not usually satisfied within the engine cylinder during intake. They are approximately satisfied at the end of compression.

8.2.2 Application to Engine Velocity Data

As explained above, it is necessary to analyze velocity data on an individual cycle basis as well as using ensemble-averaging techniques. The basic definitions for obtaining velocities which characterize the flow will now be developed. The ensemble-averaged velocity $\bar{U}_{EA}$ has already been defined by Eq. (8.5) . The ensemble-averaged fluctuation intensity $u'_{F,EA}$ is given by

$$u'_{F,EA}(\theta) = \left\{ \frac{1}{N_c} \sum_{i=1}^{N_c} [u(\theta,i)]^2 \right\}^{1/2} = \left\{ \frac{1}{N_c} \sum_{i=1}^{N_c} [U(\theta,i)^2 - \bar{U}_{EA}(\theta)^2] \right\}^{1/2} \tag{8.16}$$

It includes all fluctuations about the ensemble-averaged mean velocity.

Flow fields in fired engines are readily measured using two-dimensional particle-image velocimetry (PIV). Use of Eqs. (8.5) and (8.16) requires values for U and u at each specific crank angle under consideration. While some measurement techniques can provide this, a commonly used point velocity measurement method (laser Doppler velocimetry) provides an intermittent signal. With laser doppler velocimetry (LDV), interference fringes are produced within the small volume created by the intersection of two laser beams within the flow field. When a small particle passes through this volume, it scatters light at a frequency proportional to the particle velocity. By seeding the flow with particles small enough to be carried without slip by the flow and collecting this scattered light, the flow velocity can be determined. A signal is only produced when a particle moves through the measurement volume; thus one collects data as velocity crank-angle pairs. It is necessary, therefore, to perform the ensemble-averaging over a finite crank-angle window $\Delta\theta$ around the specific crank angle of interest, $\bar{\theta}$ The ensemble-averaged velocity equation becomes

$$\bar{U}_{EA}(\bar{\theta}) = \frac{1}{N_t} \sum_{i=1}^{N_c} \sum_{j=1}^{N_i} U_{i,j} \left(\bar{\theta} \pm \frac{\Delta\theta}{2} \right) \tag{8.17}$$

where N_i is the number of velocity measurements recorded in the window during the ith cycle, N_c is the number of cycles, and N_t is the total number of measurements. The corresponding equation for the ensemble-averaged root-mean-square velocity fluctuation is

$$u'_{F,\mathrm{EA}}(\bar{\theta}) = \left\{ \frac{1}{N_t} \sum_{i=1}^{N_c} \sum_{j=1}^{N_t} \left[u_{i,j} \left(\bar{\theta} \pm \frac{\Delta\theta}{2} \right) \right]^2 \right\}^{1/2} \tag{8.18}$$

where

$$u_{i,j} = U_{i,j} - \bar{U}_{\mathrm{EA}} \tag{8.19}$$

As has already been explained, this definition of fluctuation intensity [the ensemble-averaged rms velocity fluctuation, Eq. (8.18)] includes cyclic variations in the mean flow as well as the turbulent fluctuations about each cycle's mean flow. It is necessary to determine the mean and fluctuating velocities on an individual-cycle basis to characterize the in-cylinder flow field more completely. The critical part of this process is defining the mean velocity at a specific crank angle (or within a small window centered about that crank angle) in each cycle. In this individual-cycle velocity analysis, the individual-cycle time-averaged or mean velocity $u_{i,j} = U_{i,j} - \bar{U}_{\mathrm{EA}}$ is first determined. [12] The ensemble average of this mean velocity

$$\bar{U}_{\mathrm{EA}}(\bar{\theta}) = \frac{1}{N_c} \sum_{i=1}^{N_c} \bar{U} \left(\bar{\theta} \pm \frac{\Delta\theta}{2}, i \right) \tag{8.20}$$

is identical to the ensemble-averaged value given by Eq. (8.17) . The root mean square fluctuation in individual-cycle mean velocity can then be determined from

$$U_{\mathrm{RMS}}(\bar{\theta}) = \left\{ \frac{1}{N_c} \sum_{i=1}^{N_c} \left[\bar{U} \left(\bar{\theta} \pm \frac{\Delta\theta}{2}, i \right) - \bar{U}_{\mathrm{EA}}(\bar{\theta}) \right]^2 \right\}^{1/2} \tag{8.21}$$

This indicates the magnitude of the cyclic fluctuations in the mean motion. The instantaneous velocity fluctuation from the mean velocity, within a specified window $\Delta\,\overline{\theta}$ at a particular crank angle $\overline{\theta}$, is obtained from Eq. (8.4) . This instantaneous velocity fluctuation is ensemble-averaged, because it varies substantially cycle-by-cycle and because the amount of data is usually insufficient to give reliable individual-cycle results:

$$u'_{T,\mathrm{EA}}(\theta)=\left\{\frac{1}{N_c}\sum_{i=1}^{N_c}\left[U\left(\bar{\theta}\pm\frac{\Delta\theta}{2},i\right)-\bar{U}\left(\bar{\theta}\pm\frac{\Delta\theta}{2},i\right)\right]^2\right\}^{1/2} \tag{8.22}$$

This quantity is the ensemble-averaged turbulence intensity.

Results obtained with this approach will now be reviewed to illustrate the major features of the in-cylinder gas motion. While some of these results were obtained in special engines where the geometry and flow have been modified to make the experiments and their interpretation easier, they illustrate key features of the in-cylinder flow. Note, the flow within the cylinder is three dimensional in nature. It takes measurements at many points within the flow field, coupled with flow visualization, to characterize the flow.

Figure 8.9 shows ensemble-averaged velocities throughout the engine cycle at two measurement locations in a special L-head engine designed to generate a swirling flow within the cylinder. The engine was motored at 300 rev/min, giving a mean piston speed of 0.76 m/s. Figure 8.9 *b* shows the mean velocity within the intake-generated swirling flow, within the clearance volume, in the swirl direction. High velocities are generated during the intake process, rising to a maximum and then decreasing in response to the piston motion (see Fig. 2.2). During the compression stroke, the velocity continues to decrease but at a much slower rate. This is a motored engine cycle. A comparison of intake and compression velocities with an equivalent firing cycle showed close agreement. [13] The late expansion and exhaust stroke velocities are not typical of firing engine behavior, however, and the increase in velocity when the exhaust valve opens is due to the flow of gas *into* the cylinder because in this motored cycle, due primarily to heat losses, the cylinder pressure is then below 1 atm. Figure 8.9 *c* shows the mean velocity in the clearance volume in the same direction but on the cylinder axis. At this location, positive and negative flow velocities were measured. Since this location is out of the direct path of the intake-generated flow, velocities during the intake stroke are much lower. The non-homogeneous character of this particular ensemble-mean flow is evident.

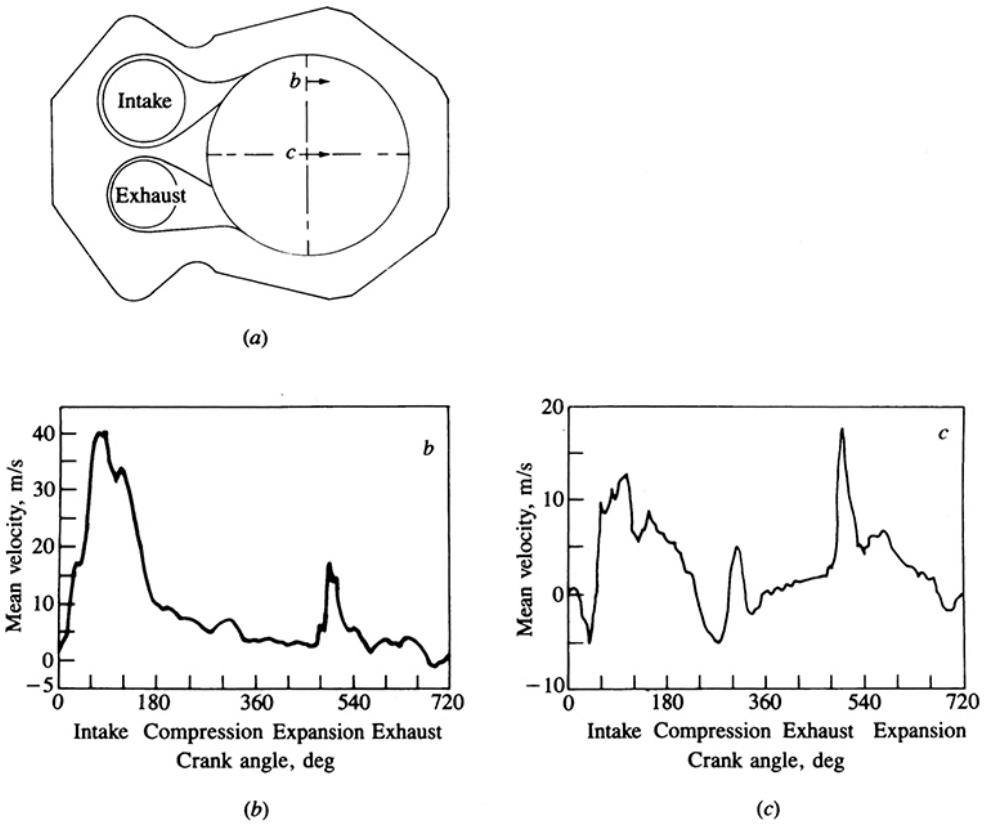


Figure 8.9 Ensemble-averaged velocities throughout the engine cycle in motored four-stroke L-head engine: 300 rev/min, mean piston speed 0.76 m/s. (*a*) Engine schematic showing measurement locations and velocity directions; (*b*) velocity at *b* in the intake flow path; (c) velocity at *c* on cylinder axis. [14]

Figure 8.10 shows the ensemble-averaged rms velocity fluctuation (which includes contributions from cycle-by-cycle variations in the mean flow and turbulence) and the ensemble-averaged individual-cycle turbulence intensity at these same two locations and directions. The difference between the two curves in each graph is an indication of the cycle-by-cycle variation in the mean flow [see Eq. (8.7)]. During the intake process, within the directed intake flow pattern, the cycle-by-cycle variation in the mean flow is small in comparison to the high turbulence levels created by the strong intake flow. Outside this directed flow region, again during intake, this cycle-by-cycle contribution is more significant relative to the turbulence. During compression, the cycle-by-cycle mean flow variation is comparable in

magnitude to the ensemble-averaged turbulence intensity. It is therefore highly significant.

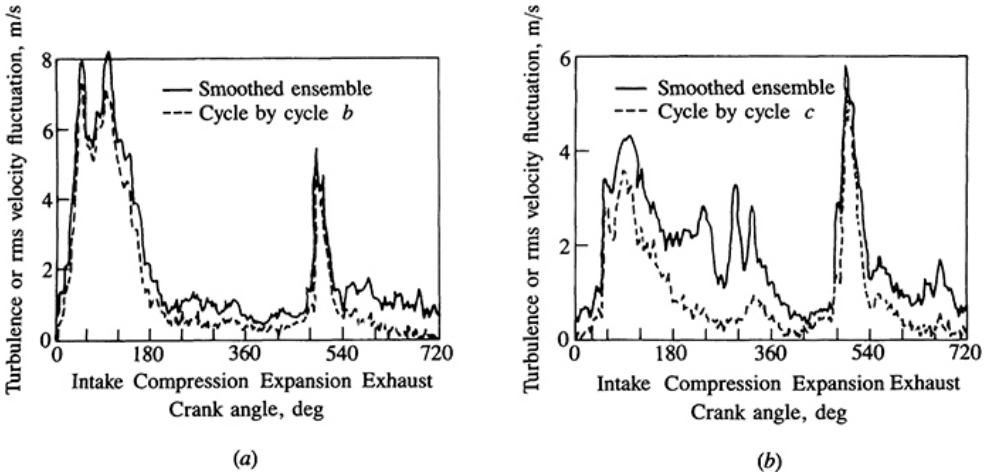


Figure 8.10 Ensemble-averaged rms velocity fluctuation and ensemble-averaged individual-cycle turbulence intensity as a function of crank angle: (*a*) at location *b* in Fig. 8.9*a*; (*b*) at location *c* in Fig. 8.9*a*.[14]

The above examples of intake-flow-generated in-cylinder motion set the stage for the subsequent flow development. Mean flow structures such as the vortex structures formed by the flow through an intake valve have usually decayed by BC. The only mean flow structures surviving into the compression stroke are swirl and tumble. The bulk of the turbulent kinetic energy observed during intake is generated, in-cylinder, by the air flowing through the valve head and seat flow restriction into the cylinder. Intake port-generated turbulence convected into the cylinder is a minor contribution. Peak turbulence levels occur around mid-intake stroke when piston speed is a maximum and decay rapidly thereafter. As the compression stroke proceeds, the spatial distribution of turbulence becomes reasonably homogeneous (uniform) and isotropic (independent of direction).

It is the character of the turbulence at the end of the compression process that is most important: that is what controls the fuel-air mixing and burning rates. These specific results support the broader conclusion that the intake flow plays the dominant role in the generation of the in-cylinder mean flow and turbulence fields during the intake stroke. However, in the absence of swirl or tumble, this intake-generated flow structure has almost disappeared by the time the compression process commences. The turbulence levels

follow this mean-flow trend, and also decay significantly by intake valve closing.

When a swirling flow is generated during intake, an almost solid-body rotating flow about the cylinder axis develops which remains stable for much longer than the intake-jet generated rotating flows illustrated in Fig. 8.3. With simple open combustion chambers, the turbulence still appears to become almost homogeneous at the end of compression. With swirl and bowl-in-piston geometry chambers, however, the flow is more complex (see Sec. 8.3). When a tumbling flow is generated during intake, it persists (and lengthens, axially) as the piston travels down the cylinder. Then, during compression, it is steadily "squashed" until mid-to-late compression stroke when it becomes unstable and breaks up. The kinetic energy stored in this tumbling motion as it breaks up is transferred to the ensuing turbulence that breakup generates.

During compression, many features of these flows affect the levels of turbulence within the cylinder. Quantifying the development of this turbulence through modeling the evolution of the turbulent kinetic energy k is an important component of quantifying the engines fuel-air mixing and combustion processes. The conservation equation for the turbulent kinetic energy is:

$$\frac{\bar{D}k}{\bar{D}t} = P - \varepsilon - (\nabla \bullet \mathbf{T}) \tag{8.23}$$

where P is the rate of production, ε the viscous dissipation rate, and the vector $\mathbf{T}$ is the transport of turbulent kinetic energy. (Turbulence modeling of engine flows is discussed fully in Sec. 14.5.2 .) There are three orthogonal components of k: for example, $k_i = (\overline{u}_i \overline{u}_i / 2)$. The turbulent transport term is not a dominant source of turbulent kinetic energy. [b] Production results from the turbulent stresses in the flow which have the form:

$$P = -(u_i u_j) S_{ij} \tag{8.24}$$

where (S_{ij}) is the mean rate of strain tensor, the terms of which depend on the gradients in the components of the turbulent fluctuating velocities. Thus one can see that the specific in-cylinder motions—swirl, tumble, and also *squish* (that results from the displacement of mixture toward the center of the cylinder as the piston approaches TC: see Sec. 8.5)—and their interactions

(see Sec. 8.6), can lead to significant turbulence production. Of course the viscous dissipation of turbulent kinetic energy (ε) continues throughout this evolution of the in-cylinder motions. [16]

The flow through the intake valve is responsible for many features of the four-stroke cycle in-cylinder motion. The flow velocity through the valve is proportional to the piston speed [see Eq. (6.10) for pseudo valve-flow velocity definition, and Eq. (2.10)]. It would be expected therefore that in-cylinder flow velocities at different engine speeds would scale with mean piston speed [Eq. (2.11)]. Figure 8.11 shows ensemble-averaged mean and rms velocity fluctuations, normalized by the mean piston speed through the cycle at three different engine speeds. The measurement location is in the path of the intake generated swirling flow (point *b* in Fig. 8.9 *a*). All the curves have approximately the same shape and magnitude, indicating the appropriateness of this velocity scaling. [c] Other results support this conclusion, though in the absence of an ordered mean motion such as swirl or tumble when the ensemble-averaged mean velocities at the end of compression are low, this scaling for the mean velocity does not always hold. [15,16]

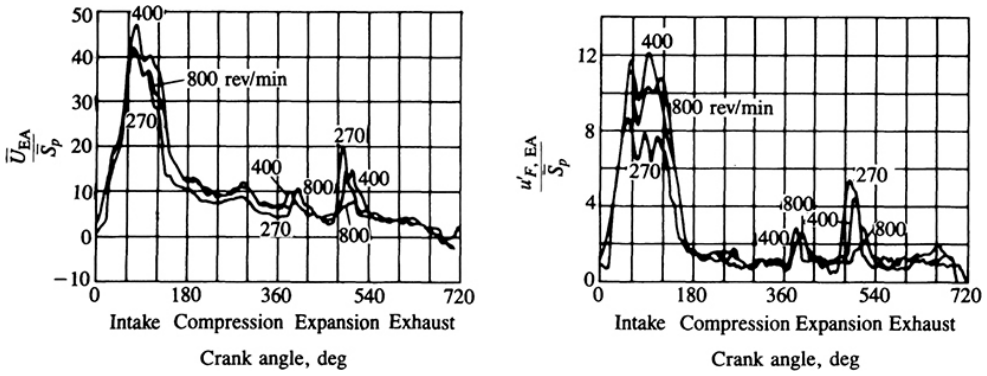


Figure 8.11 Ensemble-averaged mean and rms velocity fluctuations, normalized by mean piston speed, throughout the engine cycle for three engine speeds. Location *b* engine layout in Fig. 8.9 *a*. [14]

Figure 8.12 shows a compilation of ensemble-averaged rms fluctuation velocity or ensemble-averaged individual-cycle turbulence intensity results at TC at the end of compression, from 13 different flow configurations and combustion chamber geometries. Two of these sets of results are from two-

stroke cycle ported configurations. The measured fluctuating velocities or turbulence intensities at TC are plotted against mean piston speed. The linear relationship holds well. There is a substantial variation in the proportionality constant, in part, because in most of these studies (identified in the figure) the ensemble-averaged rms fluctuation velocity was the quantity measured. Since this includes the cycle-by-cycle fluctuation in the mean velocity, it is larger than the average turbulence intensity $u'_{T,EA}$.

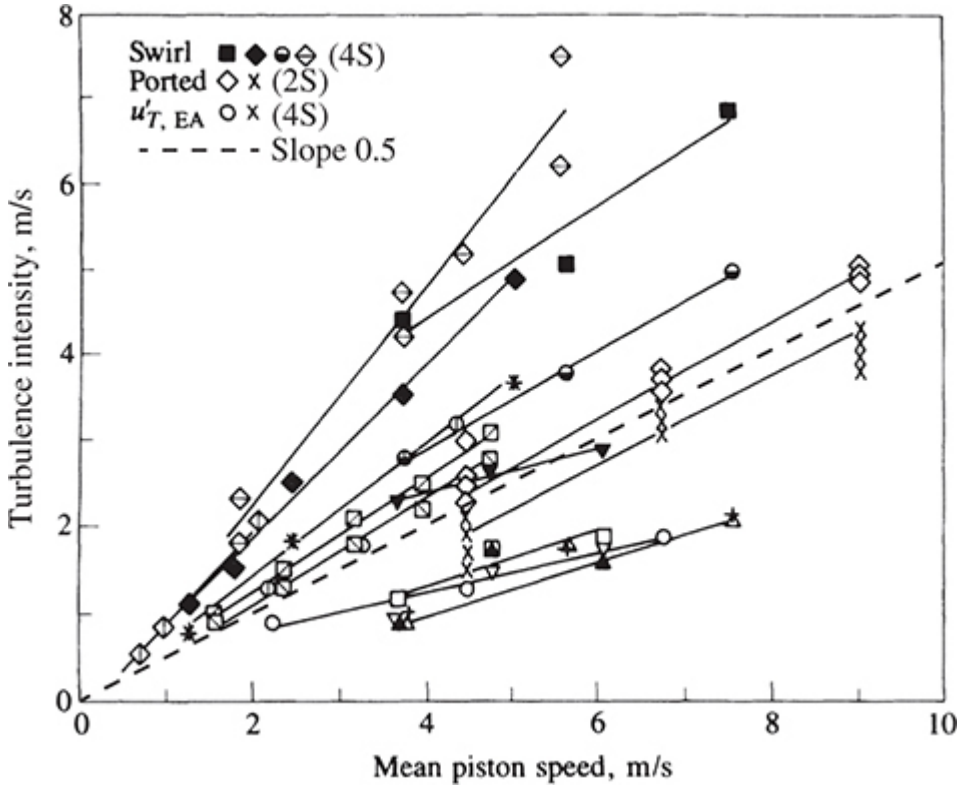


Figure 8.12 Individual-cycle turbulence intensity $u'_{T,EA}$ (o,×) and ensemble-averaged rms fluctuation velocity (remaining symbols) at TC at the end of compression, for a number of different flow configurations and chamber geometries as a function of mean piston speed. Two data sets are for two-stroke ported engines: four data sets with intake generated swirl. [17] (u′ = 0.5 $\overline{S}_p$ line also shown.)

A consensus has emerged from these studies that the turbulence intensity at around top-center of the compression process, as combustion commences, has a value equal to between half the mean piston speed and the mean piston speed, that is,

$$u'_{T,\mathrm{EA}}(\mathrm{TC}) \approx (0.5 \text{ to } 1.0)\overline{S}_p \tag{8.25}$$

Open combustion chamber shapes of four-stroke-cycle engines without significant swirl or tumble are at the lower end of this range. Small high-speed direct-injection diesel engine chambers with deep, re-entrant-shaped bowl-in-piston geometries, with high swirl, have the highest values of $u'/\overline{S}_p$. Addition of swirl, or tumble in pent-roof gasoline engine combustion chamber designs have intermediate values of $u'/\overline{S}_p$. The flow and turbulence levels in the various types of combustion chamber geometries used in gasoline SI engines, and in diesel engines, are explored in more detail in Sec. 8.6 . The ensemble-averaged cyclic variation in individual-cycle mean velocity at the end of compression also scales with mean piston speed. This quantity is normally lower in magnitude than the turbulence intensity, especially when there is a strong directed mean flow. [16]

Turbulence length and time scales in automobile engine flows have been characterized. They are important in determining fuel, air, residual gas mixing rates and, especially at TC of compression, in quantifying the scale of flame sheet wrinkling of the turbulent SI engine flame. During the intake process, the integral length scale is of the order of the intake jet diameter, which is comparable to the valve lift (≤10 mm in automobile-size engines). During compression the flow pattern relaxes to the shape of the combustion chamber. The integral length scale which defines the largest coherent structures (or eddies) in the turbulent flow can be determined using spatial correlation functions from instantaneous fluctuating velocity component measurements at multiple points in the flow via Eq. (8.8) —see Fig. 8.8. More commonly they are determined from single-point experimental data using Taylor's hypothesis via Eq. (8.10) . Taylor's hypothesis is valid when the characteristic time scale (or lifetime) of a turbulent eddy is much larger than the time required for that eddy to be convected through the measurement location by the mean flow: that is, when $u' \ll \overline{U}$. This condition is usually satisfied in engines with significant swirl, tumble, or squish-generated flows: little variation in integral length scales occurs as engine speed changes. Also, comparisons of the above two approaches for determining l_I give comparable results. These findings indicate that at around TC of compression, the in-cylinder flow turbulence is close to isotropic: that is, the fluctuating velocity components have no preferred direction. In such flows,

applying Eq. (8.8) results in two integral length scales: $l_{I,long}$ and $l_{I,lat}$ which correspond to fluctuating velocity components parallel to the separating length vector (**x**), and lateral to the separating vector, respectively.[d]

Generally, lateral integral length scales, $l_{I,lat}$ determined near TC of compression, are in the range of 1 to 3 mm while longitudinal scales range from about 2 to 8 mm, [16] thus indicating that turbulence isotropy often exists. Temporarily, integral length scales exhibit a broad minimum around TC, and the value is relatively insensitive to engine speed. Strong velocity gradients in the in-cylinder flow can lead to significant spatial variations in length scales throughout the combustion chamber. While integral scales are comparable to the largest scales characterizing in-cylinder flow, and the physical flow boundaries which the combustion chamber shape imposes limit the integral scale, there is no fundamental reason beyond the above started constraints why the integral scale should be proportional to geometric features such as the clearance height as is sometimes claimed.

The integral time scale at the end of compression decreases with increasing engine speed. It is of order 1 ms at engine speeds of about 1000 rev/min. The integral length scale at the end of compression, as explained above, is a few mm, varies little with engine speed, and decreases as the piston approaches TC. The micro time scale at the end of compression is of order 0.1 ms at 1000 rev/min, and decreases as engine speed increases (again in automobile-size engine cylinders). Micro length scales are of order 1 mm at the end of compression and vary little with engine speed. Kolmogorov length scales at the end of compression are of order 10^{-2} mm. [11, 18, 19]

8.3 SWIRL

Swirl is defined as organized rotation of the charge about the cylinder axis. Swirl is created by bringing the intake flow into the cylinder with an initial angular momentum. While some decay in swirl due to friction occurs during the engine cycle, intake-generated swirl usually persists through the compression, combustion, and expansion processes. In engine designs with bowl-in-piston combustion chambers, the rotational motion setup during intake is substantially amplified during compression. Swirl is used in diesels

and some stratified-charge engine concepts to promote more rapid mixing between the inducted air charge and the injected fuel. Swirl is also used to speed up the combustion process in spark-ignition engines. In two-stroke engines, it is used to improve scavenging. In some designs of prechamber engines, organized rotation about the prechamber axis is also characterized as swirl. In prechamber engines where swirl within the precombustion chamber is important, the flow into the prechamber during the compression process creates the rotating flow. Prechamber flows are discussed in Sec. 8.7 .

8.3.1 Swirl Measurement

The nature of the swirling flow in an actual operating engine is difficult to determine. Accordingly, steady flow tests are often used to characterize the swirl. Air is blown steadily through the inlet port and valve assembly in the cylinder head into an appropriately located equivalent of the cylinder. A common technique for characterizing the swirl within the cylinder has been to use a light paddle wheel, pivoted on the cylinder centerline (with low friction bearings), mounted between 1 and 1.5 bore diameters down the cylinder. The paddle wheel diameter is close to the cylinder bore. The rotation rate of the paddle wheel is used as a measure of the air swirl. However, this rotation rate depends on the location of the wheel and its design, as well as the details of the swirling flow. A more precise technique is the impulse swirl meter shown in Fig. 8.13. A honeycomb flow straightener replaces the paddle wheel: it measures the total torque exerted by the swirling flow. This torque equals the flux of angular momentum through the plane coinciding with the flow-straightener upstream face.

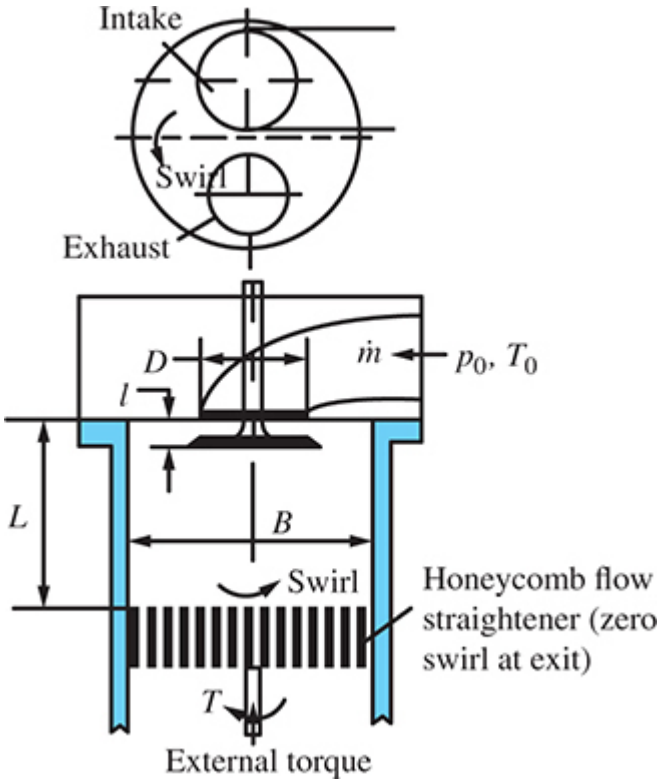


Figure 8.13 Schematic of steady-flow angular-momentum flux swirl meter.[20]

For each of these approaches, a *swirl coefficient* is defined which essentially compares the flow's angular momentum with its axial momentum. For the paddle wheel, the swirl coefficient C_s is defined by

$$C_s = \omega_p B / v_0 \tag{8.26}$$

where ω_p is the paddle wheel angular velocity ($= 2\pi N_p$, where N_p is the rotational speed) and the bore B has been used as the characteristic dimension. The characteristic velocity, v_0, is derived from the pressure drop across the valve using an incompressible flow equation:

$$v_0 = \left[\frac{2(p_0 - p_c)}{\rho}\right]^{1/2} \tag{8.27}$$

or a compressible flow equation:

$$v_0 = \left\{ \frac{2\gamma}{(\gamma-1)} \frac{p_0}{\rho_0} \left[1 - \left(\frac{p_c}{p_0} \right)^{(\gamma-1)/\gamma} \right] \right\}^{1/2} \tag{8.28}$$

where the subscripts 0 and c refer to upstream stagnation and cylinder values, respectively. The difference between Eqs. (8.27) and (8.28) is usually small. With the impulse torque meter, characteristic velocity and length scales must also be introduced. Several swirl parameters have been defined. The simplest is

$$C_s = \frac{8T}{\dot{m} v_0 B} \tag{8.29}$$

where T is the torque and $\dot{m}$ the air mass flow rate. The velocity, v_0, defined by Eq. (8.27) or Eq. (8.28) , and the bore have again been used to obtain a dimensionless coefficient. Note that for solid-body rotation of the fluid within the cylinder at the paddle wheel speed ω_p, Eqs. (8.26) and (8.29) give identical swirl coefficients. In practice, because the swirling flow is not solid-body rotation and because the paddle wheel usually lags the flow due to slip, the impulse torque meter gives higher swirl coefficients. [21] When swirl measurements are made in an operating engine, a *swirl ratio* is normally used to define the swirl. It is defined as the angular velocity of a solid-body rotating flow ω_s that has equal angular momentum to the actual flow, divided by the crankshaft angular rotational speed:

$$R_s = \frac{\omega_s}{2\pi N} \tag{8.30}$$

Typically, values for swirl ratio are in the range of 1 to 4. During the induction stroke in an engine, the flow and the valve open area, and consequently the angular momentum flux into the cylinder, vary with crank angle. Whereas in the above described rig tests, the flow and valve open area are fixed, and the angular momentum passes down the cylinder continuously, in the engine intake process the momentum produced under corresponding conditions of flow and valve lift remains in the cylinder. Steady-state impulse torque-meter flow-rig data can be used to estimate engine swirl in the following manner. [21] Assuming that the port and valve

retain the same characteristics under the transient conditions of the engine as on the steady-flow rig, the equivalent solid-body angular velocity ω_s at the end of the intake process is given by

$$\omega_s = \frac{8}{B^2}\left[\int_{\theta_1}^{\theta_2} T\,d\theta\right] \Big/ \left[\int_{\theta_1}^{\theta_2} \dot{m}\,d\theta\right]$$

where θ_1 and θ_2 are crank angles at the start and end of the intake process and the torque T and mass flow rate $\dot{m}$ are evaluated at the valve lift corresponding to the local crank angle. Using Eq. (8.29) for T, Eq. (6.11) for $\dot{m}$, assuming v_0 and ρ are constant throughout the intake process, and introducing volumetric efficiency η_v based on intake manifold conditions via Eq. (2.27) , it can be shown that

$$R_s = \frac{\omega_s}{2\pi N} = \pi \eta_v BL\left[\int_{\theta_1}^{\theta_2} (A_v C_D) C_s\,d\theta\right] \Big/ \left[\int_{\theta_1}^{\theta_2} (A_v C_D)\,d\theta\right]^2 \qquad (8.31)$$

where $A_V C_D$ is the effective valve open area at each crank angle. Note that crank angle in Eq. (8.31) should be in radians. Except for its (weak) dependence on η_v, Eq. (8.31) gives R_s independent of operating conditions directly from rig test results and engine geometry.

The relationship between steady-flow rig tests (which are used because of their simplicity) and actual engine swirl patterns is complex. Steady-flow tests adequately define the swirl generating characteristics of the intake port and valve (at fixed valve lift) and are used extensively for this purpose. However, the swirling flow set up in the cylinder during intake can change significantly during compression.

8.3.2 Swirl Generation during Induction

Two general approaches are used to create swirl during the induction process. In one, the flow is discharged into the cylinder tangentially, where it is deflected sideways and downward in a swirling motion. In the other, the swirl is largely generated within the inlet port: the flow is forced to rotate about the valve axis *before* it enters the cylinder. The former type of motion is achieved by forcing the flow distribution around the circumference of the

inlet valve to be nonuniform so that the intake flow has substantial net angular momentum about the cylinder axis. The directed port and deflector wall port in Fig. 8.14 are two common ways of achieving this result. The directed port brings the flow toward the valve opening in the desired tangential direction. Its passage is straight, which due to other cylinder head requirements restricts the flow area and results in a relatively low discharge coefficient. The deflector wall port uses the port's inner side wall to force the flow preferentially through the outer periphery of the valve opening, in a tangential direction. Since only one wall is used to obtain a directional effect, the port areas are less restrictive.

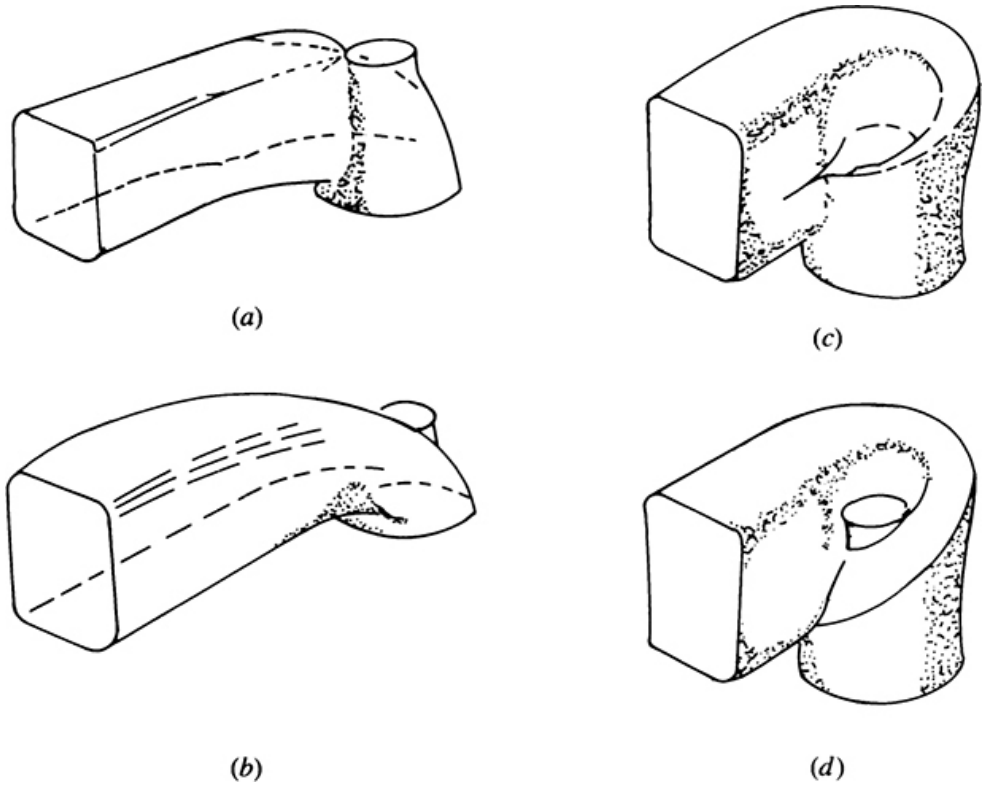


Figure 8.14 Different types of swirl-generating inlet ports: (*a*) deflector wall; (*b*) directed; (*c*) shallow ramp helical; (*d*) steep ramp helical.[22]

The second broad approach is to generate swirl within the port about the valve axis, prior to the flow entering the cylinder. Two examples of such *helical ports* are shown in Fig. 8.14. With helical ports, a higher flow discharge coefficient (lower flow resistance) at equivalent levels of swirl is usually obtained, since the whole periphery of the valve open area can be

more fully utilized. A higher volumetric efficiency—important at high loads—results. Also, helical ports are less sensitive to position displacements such as can occur in casting, since the swirl generated depends mainly on the port geometry above the valve and not the position of the port relative to the cylinder axis.

Figure 8.15 compares steady-state swirl-rig measurements of examples of the ports in Fig. 8.14. The rig swirl number increases with increasing valve lift, reflecting the increasing impact of the port shape and decreasing impact of the flow restriction between the valve head and seat. Helical ports normally impart more angular momentum at medium lifts than do directed ports. [21, 23] The swirl ratios for these ports calculated from this rig data using Eqs. (8.27) and (8.29) are: 2.5 for the directed port, 2.9 for the shallow ramp helical, and 2.6 for the steep ramp helical. Vane swirl-meter swirl ratios were about 30% less. These impulse-swirl-meter-derived engine swirl ratios are within about 20% of the solid-body rotation rate which has equal angular momentum to that of the cylinder charge determined from tangential velocity measurements made within the cylinder of an operating engine with the same port, at the end of the induction process. [23]

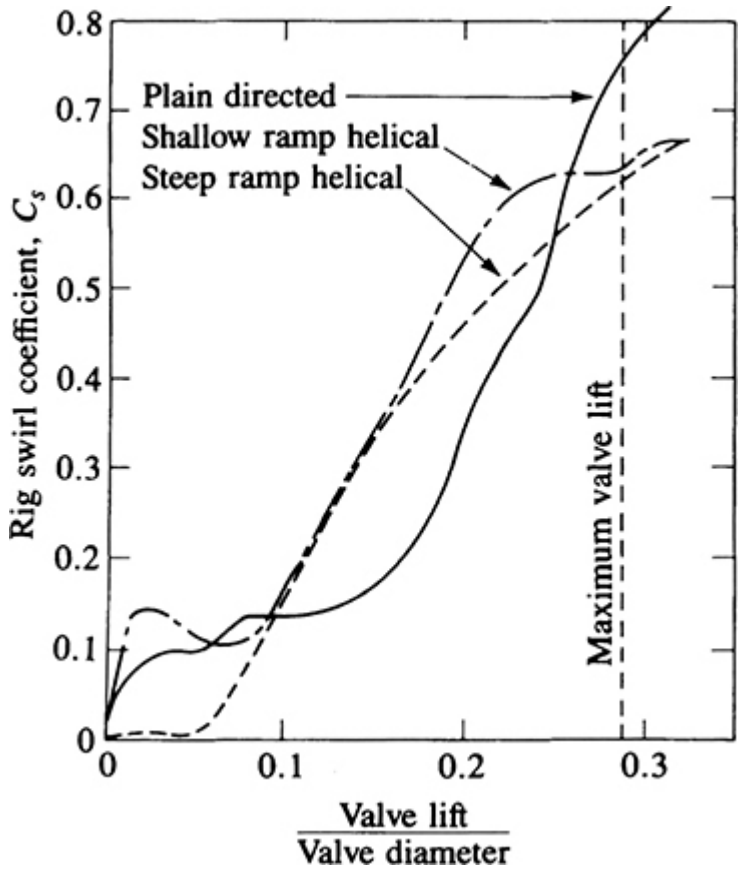


Figure 8.15 Steady-state torque meter swirl measurements of directed, shallow ramp, and steep ramp helical ports as a function of inlet valve lift/diameter ratio.[21]

Directed and deflector wall ports, and masked valve or head designs produce a tangential flow into the cylinder by increasing the flow resistance through that part of the valve open area where flow is not desired. A highly non-uniform flow through the valve periphery results, and the flow into the cylinder has a substantial v_θ velocity component in the same direction about the cylinder axis. In contrast, helical ports produce the swirl in the port upstream of the valve, and the velocity components v_r and v_z through the valve opening, and v_θ *about the valve axis* are approximately uniform around the valve open area. Figure 8.16 shows velocity data measured at the valve exit plane in steady-flow rig tests with examples of these two types of port. The valve and cylinder wall locations are shown. In Fig. 8.16 *a*, the deflector wall of the tangentially oriented port effectively prevents any significant flow around half the valve periphery. In contrast, in Fig. 8.16 *b*

with the helical port, the air flows into the cylinder around the full valve open area. The radial and axial velocities are essentially uniform around the valve periphery. The swirl velocity about the valve axis (anticlockwise when viewed from above) for this helical port is relatively uniform and is about half the magnitude of the radial and axial velocities through the valve open area.

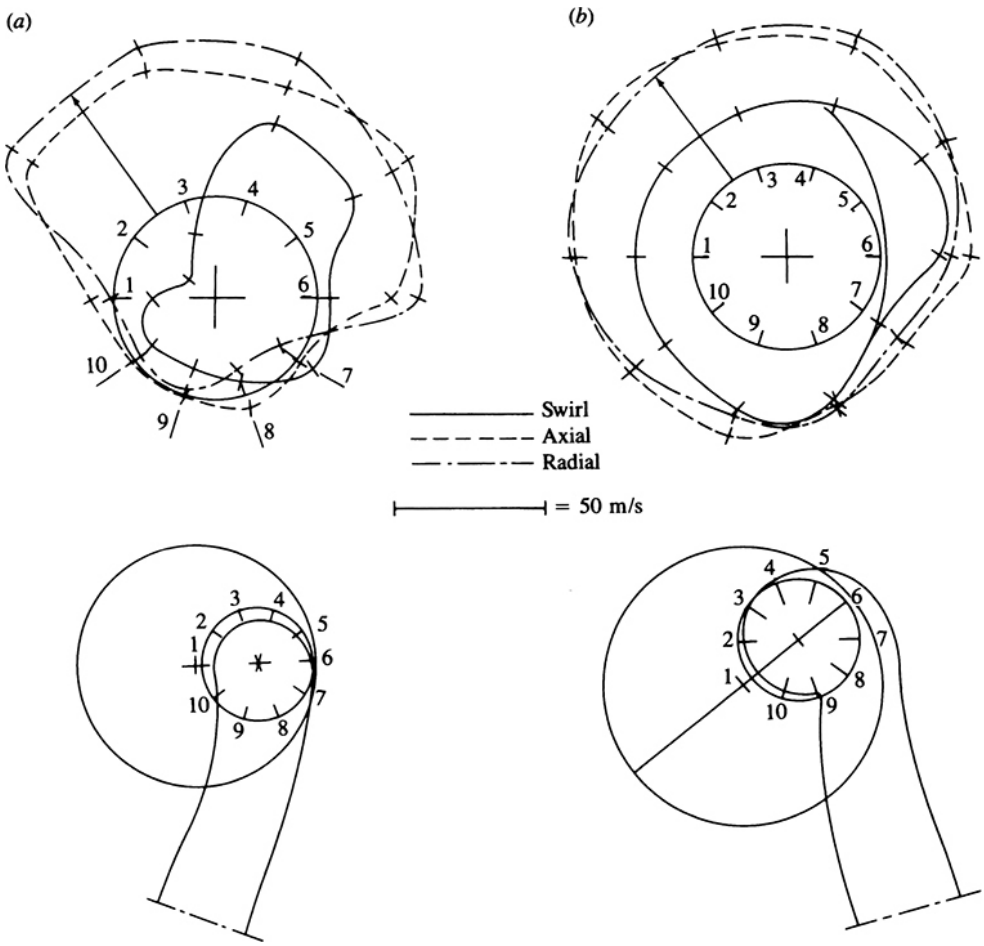


Figure 8.16 Swirl, axial, and radial velocities measured 2 mm from cylinder head around the valve circumference for (*a*) tangential deflector-wall port and (*b*) helical port; magnitude of velocity is given by the distance along a radial line (from valve axis), from valve outline to the respective curve scaled by the reference length (examples of radial velocity indicated by two arrows); valve lift = 12.8 mm.[24, 25]

The swirling airflow within the cylinder of an operating engine is not uniform. The velocities generated between the valve head and seat at each

point in the induction process depend on the valve open area and piston velocity. The velocities are highest during the first half of the intake process as indicated by the pseudo valve-flow velocity curve in Fig. 6.17. Thus, the swirl velocities generated during this portion of the induction stroke are higher than the swirl generated during the latter half of the stroke: there is swirl stratification. Also, the flow pattern close to the cylinder head during induction is comparatively disorganized, and not usually close to a solid-body rotation. It consists of a system of vortices, created by the high-velocity tangential or spiraling intake jet. Further down the cylinder, the flow pattern is closer to solid-body rotation with the swirl velocity increasing with increasing radius. [21, 22] This more ordered flow directly above the piston produces higher swirl velocities in that region of the cylinder. As the piston velocity decreases during intake, the swirl pattern redistributes, with swirl speeds close to the piston decreasing and swirl speeds in the center of the cylinder increasing. [25] Note that the axis of rotation of the swirling in-cylinder gases may rotate about the cylinder and not exactly coincide with the cylinder axis. This discussion indicates that there is a close to linear scaling of swirl with engine speed, though at higher speeds, flow fiction somewhat reduces this scaling.

8.3.3 Swirl Modification within the Cylinder

The angular momentum of the air, which enters the cylinder at each crank angle during induction, decays throughout the rest of the intake process and during the compression process due to friction at the walls and turbulent dissipation within the fluid. Typically one-quarter to one-third of the initial moment of momentum about the cylinder axis will be lost by top-center at the end of compression. However, swirl velocities in the charge can be substantially increased during compression by suitable design of the combustion chamber. In many designs of direct-injection diesel combustion systems, air swirl is used to obtain much more rapid mixing between the fuel injected into the cylinder and the air than would occur in the absence of swirl. The tangential velocity of the swirling airflow set up inside the cylinder during induction is substantially increased by forcing most of the air into a compact bowl-in-piston combustion chamber, usually centered on the cylinder axis, as the piston approaches its top-center position. Neglecting the

effects of friction, angular momentum is conserved: so as the moment of inertia of the air is decreased, its angular velocity must increase.

However, the total angular momentum of the charge within the cylinder decays due to friction at the chamber walls. Thus the angular momentum of the cylinder charge Γ_c changes with time according to the moment of momentum conservation equation:

$$\frac{d\Gamma_c}{dt} = J_i - T_f \tag{8.32}$$

where J_i is the flux of angular momentum into the cylinder and T_f is the torque due to wall friction. At each point in the intake process J_i is given by

$$J_i = \int_{A_v} \rho r \mathbf{v} \cdot \mathbf{dA}_v \tag{8.33}$$

where $\mathbf{d A}_v$ is a vector element of the valve open area, as defined in Fig. 8.17. While the angular momentum entering the cylinder during the intake process is

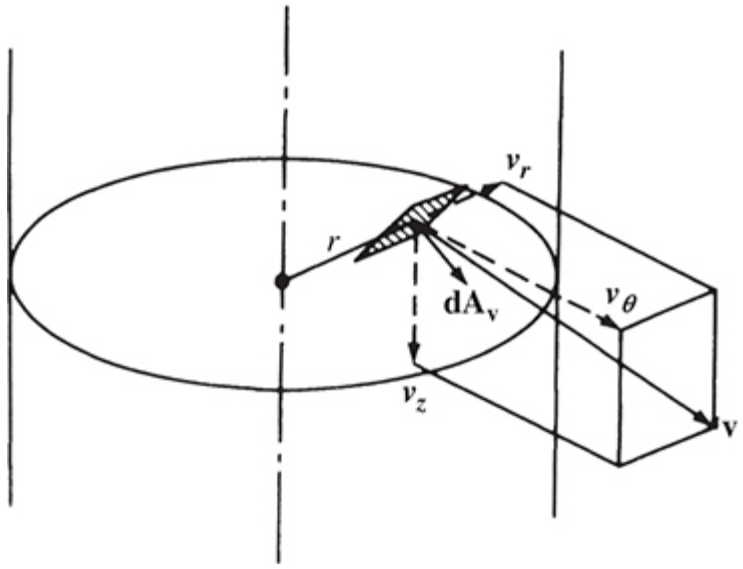


Figure 8.17 Definition of symbols in equation for angular momentum flux into the cylinder [Eq. (8.33)].

$$\gamma_{c,i} = \int_{t_{ivo}}^{t_{ivc}} \rho r v_\theta \mathbf{v} \bullet \mathbf{dA}_v dt$$

the actual angular momentum within the cylinder at the end of induction will be less, due to wall friction during the intake process. Friction continues through the compression process so the total charge angular momentum at the end of compression is further reduced.

There is friction on the cylinder liner, cylinder head, and piston crown (including any bowl or cut outs within the crown). This friction can be estimated using friction formulas developed for flow over a flat plate, with suitable definitions of characteristic length and velocity scales. Friction on the cylinder liner can be estimated from the wall shear stress:

$$\tau = \frac{1}{2}\rho\left(\frac{\omega_s B}{2}\right)^2 C_F \tag{8.34}$$

where ω_s is the equivalent solid-body swirl. The friction factor C_F is given by the flat plate formula:

$$C_F = 0.037\lambda(\mathrm{Re}_B)^{-0.2} \tag{8.35}$$

where λ is an empirical constant introduced to allow for differences between the flat plate and cylinder wall ($\lambda \approx 1.5$) [23] and Re B is the equivalent of the flat plate Reynolds number [Re $B = \rho(B_\omega s/2)(\pi B/\mu)$]. Friction on the cylindrical walls of a piston cup or bowl can be obtained from the above expressions with D_B, the bowl diameter, replacing the bore.

Friction on the cylinder head, piston crown, and piston bowl floor can be estimated from expressions similar to Eqs. (8.34) and (8.35). However, since the tangential velocity v_θ at the wall varies with radius, the shear stress should be evaluated at each radius and integrated over the surface: for example, [26]

$$\tau(r) = C_1 \frac{1}{2}\rho[v_\theta(r)]^2 \,\mathrm{Re}^{-0.2} \tag{8.36}$$

$$\mathrm{Re} = \rho v_\theta(r) r/\mu$$

with

where C_1 is an empirical constant (≈ 0.055). An alternative approximate approach is to evaluate these components of the wall shear stress at the mean radius.

Next, consider the effects on swirl of radially inward displacement of the air charge during compression. The most common example of this phenomenon occurs with the bowl-in-piston combustion chamber design of medium- and high-speed direct-injection diesels (see Sec. 10.2). However, in spark-ignition engines where swirl can be used to increase the burning rate, the shape of the combustion chamber close to top-center can also force radially inward motion of the charge. For a given swirling in-cylinder flow at the end of induction, and neglecting the effects of friction, as the moment of inertia of the air about the cylinder axis is decreased the air's angular velocity must increase to conserve angular momentum. For example, for solid-body rotation of the cylinder air charge of mass m_c, the initial angular momentum $\Gamma_{c,i}$ and solid-body rotation $\omega_{s,i}$ are related at bottom-center by

$$\Gamma_{c,i} = I_c \omega_{s,i}$$

where I_c is the moment of inertia of the charge about the cylinder axis. For a disc-shaped combustion chamber, $I_c = m\, c_B^{\,2}/8$ and is constant. For a bowl-in-piston combustion chamber,

$$I_c = \frac{m_c B^2}{8} \frac{[(z/h_B) + (D_B/B)^4]}{[(z/h_B) + (D_B/B)^2]} \tag{8.37}$$

where D_B and h_B are the diameter and depth of the bowl, respectively, and z is the distance of the piston crown from the cylinder head. At TC crank position, $z \approx 0$ and $I_c \approx m_c D_B^2/8$. At the end of induction, $I_c \approx m_c\, B^2/8$. Thus, in the absence of friction, ω_s would increase by $(B/D_B)^2$, usually a factor of about 4.

In an operating engine with this bowl-in-piston chamber design, the observed increase in swirl in the bowl is less; it is usually a factor of 2 to 3.[21, 23] This is because of wall friction, dissipation in the fluid due to turbulence and velocity gradients, and the fact that a fraction of the fluid

remains in the clearance region above the piston crown. The loss in angular momentum due to these effects will vary with geometric details, initial swirl flow pattern, and engine speed.

Swirl velocity distributions in the cylinder at the end of induction show the tangential velocity increasing with radius, except close to the cylinder wall where friction causes the velocity to decrease. While the velocity distribution is not that of a solid-body rotation, depending on port design and operating conditions, it is often close to solid-body rotation. [21, 23] Departures from the solid-body velocity distribution are greater at higher engine speeds, suggesting that the flow pattern in the cylinder at this point in the cycle is still developing with time. [21, 27] In the absence of radially inward gas displacement during compression, the flow pattern continues to develop toward a solid-body distribution throughout the compression stroke. [23] Swirl ratios of 3 to 5 at top-center can be achieved with the ports shown in Fig. 8.13, with flat-topped pistons (i.e., in the absence of any swirl amplification during compression).

8.4 TUMBLE

An in-cylinder *tumble flow* is used in spark-ignition engines to speed up the combustion process. As Fig. 8.4 indicates, tumble is charge rotation about an axis orthogonal to the cylinder axis. It is set up, usually in pent-roof combustion chambers with four valves (two intake valves) with inclined valve stems (see Fig. 8.18), by positioning the intake ports to bring the entering airflow into the cylinder through the upper portion of the open area between the valve head and valve seat. This imparts a transverse (across the cylinder) motion to the entering air. The descending piston then pulls that flow down the liner on the side opposite to the intake valves; it then moves back across the piston crown and upward along the liner to complete this flow rotation. Because the intake-valve open area is not used uniformly, these transverse flow velocities past the valve for a given piston speed are higher, thus additional kinetic energy is stored in this in-cylinder tumbling flow. Note also that, as a result of the uneven use of the valve (and port) open area, the flow resistance of the intake valve and port system is higher so the engine's maximum volumetric efficiency is lowered. During compression,

this tumble flow is “squashed” by the upward-moving piston, becomes unstable as a consequence, and breaks up into smaller-scale vortices generating turbulent kinetic energy which, following the spark discharge, increases the engine’s mixture burning rate. In spark-ignition engines, tumble is commonly used to achieve faster burn rates (important at part-load to achieve good EGR tolerance and improve part-load efficiency); the modest loss in engine breathing capacity at higher loads and speeds is accepted as a necessary consequence.

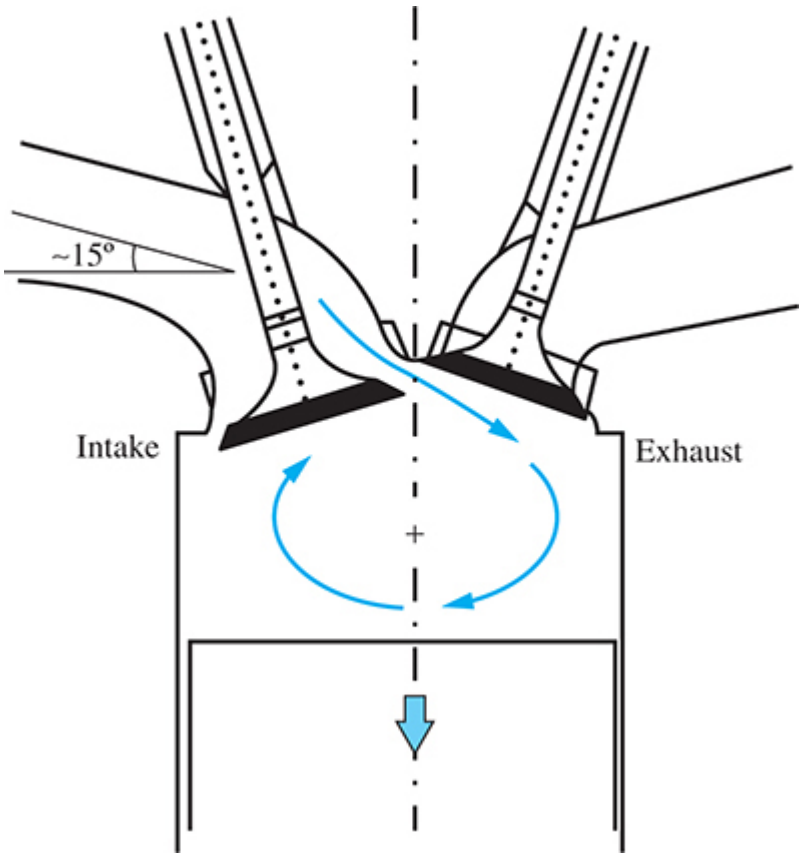


Figure 8.18 Schematic showing how the intake ports and valves (in a four-valve-per-cylinder engine), and piston motion down the cylinder, generate tumble. Key feature is low-angle intake ports that direct the flow primarily through the upper portion of the open valve gap across the cylinder head.

Measuring and characterizing tumble are done analogously to swirl. Figure 8.19 illustrates how the tumble flow angular momentum can be measured in a steady flow rig (analogous to the swirl meter rig shown in Fig. 8.15). The measured torque T, is proportional to the mass flow rate into the

cylinder through the intake ports, the in-cylinder flow's (average) angular velocity, and the mean radius of rotation about the tumble flow axis (about $B/4$). [20]

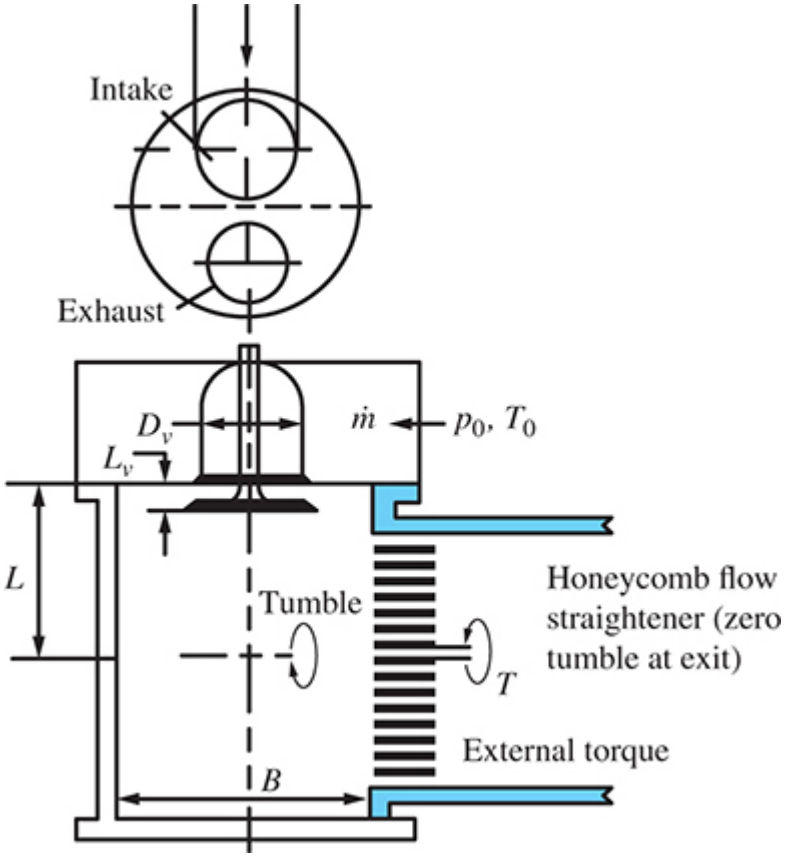


Figure 8.19 Schematic of steady-flow angular-momentum-flux tumble meter.[20]

Tumble flows are characterized by their angular momentum. Figure 8.18 shows what is often called *normal tumble* where flow rotation is about an axis (shown in the figure) that is parallel to the ridge of the pent roof cylinder head. *Reverse tumble* , where the flow would rotate in the opposite direction to that in Fig. 8.18 (counter-clockwise), is generated with the intake ports directed vertically downward toward the intake valve, rather than almost horizontal (as in Fig. 8.18). Reverse tumble is used in certain designs of stratified-charge direct-injection gasoline engines to control the motion of the evolving fuel-air mixture "cloud" the evaporating fuel spray creates (see Sec. 7.7.2). When tumble is created in the presence of swirl, the air charge

rotation about the cylinder axis can result in a tumbling flow component in the plane at 90° to the rotation plane shown in Fig. 8.18. This *cross tumble* flow, orthogonal to the normal tumble flow, should be combined with the normal tumble angular momentum to quantify the total in-cylinder tumble flow angular momentum.

These various tumbling flows are characterized by a *tumble ratio*, analogous to the swirl ratio defined by Eq. (8.30) . The tumble ratio is defined as

$$R_t = \frac{\omega_t}{2\pi N} \tag{8.38}$$

where ω_t is the total angular momentum of the charge divided by the moment of inertia of the in-cylinder fluid: i.e., the equivalent solid-body angular speed divided by the angular rotation rate of the crankshaft. [28] Thus:

$$R_t = \frac{\int_{\theta_1}^{\theta_2} \rho(\mathbf{r}\times\mathbf{v})\bullet\mathbf{i}\,d\theta}{2\pi N \int_{\theta_1}^{\theta_2} \rho(\mathbf{r}\bullet\mathbf{r})d\theta} v \tag{8.39}$$

When both normal- and cross-tumble are significant, the overall tumble ratio, $R_{t,o}$, is given by:

$$R_{t,o} = (R_{t,n}^2 + R_{t,c}^2)^{1/2} \tag{8.40}$$

For a given geometry cylinder-head design, the tumble ratio increases as the intake valve lift and the piston speed increase during the first half or so of the intake stroke. With four-valve cylinder heads, with intake ports of the geometry shown in Fig. 8.18, tumble ratios of 1 to 2 or so are realized. With additional flow-directing restrictions in the intake ports (e.g., throttles or baffles) higher tumble ratios of up to about 4 can be obtained. Then, during compression the tumbling flow is squashed by the piston moving up the cylinder. As the clearance height becomes small, this squashed tumble flow breaks up into smaller-scale rotating eddies that decay into turbulence.

Intake port baffles (which close off specific portions of the port flow area) can be used to enhance the tumble (and swirl) levels generated with the

cylinder during the intake process. These close off that portion of the intake port cross section leading to the part of the annular valve-head valve-seat open area where flow is *not* desired. For tumble, with angled valve stems, increasing the flow through the upper portion of the intake valve opening creates a stronger tumbling flow: with swirl, the airflow is directed through the valve open area closest to the cylinder liner wall. [29]

Creating more vigorous in-cylinder flows such as swirl and tumble involve higher pressure losses and flow restrictions across the intake port, and the valve and seat opening, and thus lower maximum engine airflows. The challenge is to achieve the desired faster combustion rates that these higher in-cylinder kinetic energy flows produce while minimizing the loss in airflow at the higher engine speeds and loads, where the combustion process is usually "fast enough." Thus many cylinder-head intake-port and piston-crown geometries have been explored to improve the combustion benefits and reduce the flow losses. These include fixed geometry designs, as well as use of throttles or plates in the intake ports to achieve benefits at part-load and lower speed, and avoid the penalties elsewhere. With port fuel injected engine system, the usual application is for homogeneous gasoline-air mixture SI engine concepts. With direct-injection fuel systems, both homogeneous mixture gasoline-direct-injection (GDI) engine concepts, and stratified engine concepts (at part load) are important application areas. The flow characteristics of several different tumble motion approaches (at full load) are shown in Fig. 8.20. [30] The port-fuel-injected (PFI) engine characteristics indicate the flow losses incurred in the absence of efforts to increase the in-cylinder motion to enhance the combustion rate. Some of these engines as a result of their cylinder-head design do create a modest tumbling flow during the intake process. Forward tumble is generated as indicated in Fig. 8.18 (shown in the clockwise direction). Reverse tumble rotates anti-clockwise, and uses vertically downward directed intake ports to achieve such a flow. As the in-cylinder tumble flow is intensified (with, for example, rotatable plates in the intake ports to block off the lower portion of the intake port—at lighter loads), the intake port and valve flow resistance at high loads and speeds) increases substantially. That, however, is not so important, as this partial port blocking to achieve high in-cylinder tumble would neither be needed nor utilized at these higher-load operating conditions.

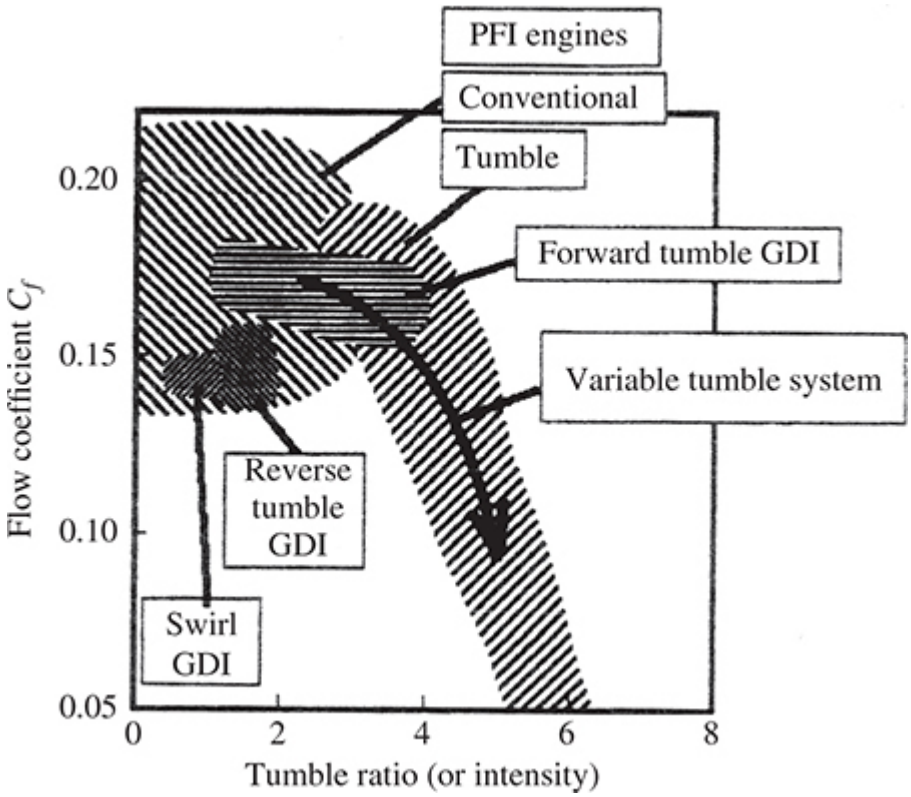


Figure 8.20 Comparison of flow characteristics—flow coefficient (resistance) based on the pressure loss across the intake ports and valves, and tumble intensity (or ratio); see Eqs. (8.38) and (8.39)—of different tumble and swirl generating concepts. GDI, gasoline-direct-injection SI engines; PFI, port-fuel-injection SI engines. [30]

In gasoline SI engines operating with a homogeneous (usually stoichiometric) mixture, moderate tumble is usually preferred over swirl as the way to increase the burning rate so that it is fast enough to provide low combustion variability (good enough combustion stability) at low- to mid-loads and speeds.

8.5 PISTON-GENERATED FLOWS: SQUISH

The movement of the piston up and down the cylinder obviously produces charge motion. In its simplest form, with a flat piston crown and cylinder head, the fluid moves axially as charge compression occurs, with the gas in

contact with the piston moving at the piston speed (see Fig. 2.2). When the valves are closed, the gas in contact with the cylinder head is at rest. In between the piston crown and cylinder head, the axial or z-component of the gas velocity will be the piston speed scaled linearly by the ratio of the axial distance of the in-cylinder location from the mean cylinder head location relative to the axial distance of the piston crown:

$$v_z(z) = \frac{z}{c_c + c} \cdot S_p \tag{8.41}$$

c is the piston displacement from its TC position and c_c is the average height of the clearance volume [see Fig. 2.1 and Eq. (2.6)], and the z-coordinate is zero at the TC piston-crown location. (Note: $z = -c_c$ at the average cylinder-head location.)

Squish

When the cylinder head or piston crown surfaces are not flat but are shaped (e.g., pent-roof cylinder heads in spark-ignition engines, and bowl-in-piston combustion chambers in diesels) this piston-generated axial velocity during compression and expansion produces transverse or radial flows due to differences in the axial distance between the cylinder head and piston crown surfaces across the cylinder cross section. *Squish* is the name given to the radial or transverse gas motion that occurs toward the end of the compression stroke (and early part of the expansion stroke) when a portion of the piston face and cylinder head approach (or separate from) each other closely. Figure 8.21 shows how gas is thereby displaced into the more open region of the combustion chamber at the end of compression. Figure 8.21 *a* shows a wedge-shaped SI engine combustion chamber and Fig. 8.21*b* shows a bowl-in-piston diesel combustion chamber. The amount of squish is often defined by the *percentage squish area:* that is, the percentage of the cylinder cross-sectional area, $\pi B^2/4$, where the cylinder head and piston (the shaded areas in Fig. 8.21) approach each other closely. Squish-generated gas motion results from using a compact combustion chamber geometry.

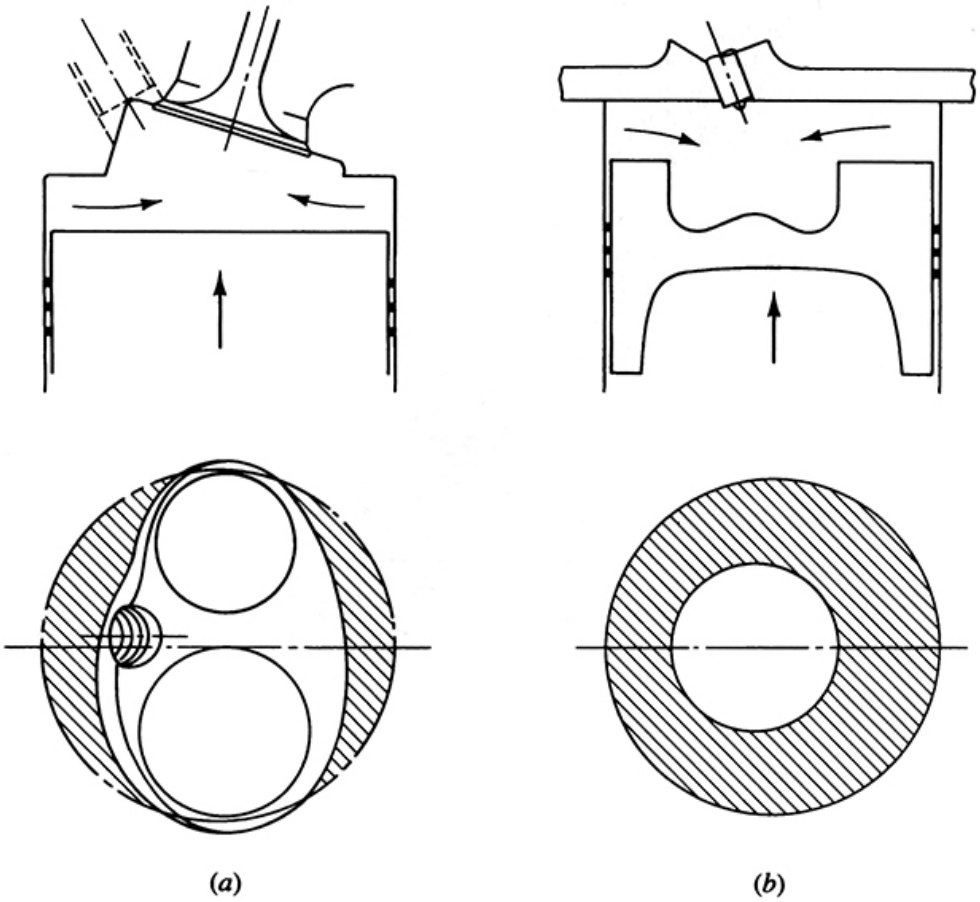


Figure 8.21 Schematics of how piston motion generates squish: (*a*) wedge-shaped SI engine combustion chamber; (*b*) bowl-in-piston direct-injection diesel combustion chamber.

A theoretical squish velocity can be calculated from the instantaneous displacement of gas across the inner edge of the squish region (across the dashed lines in the drawings in Fig. 8.22 *a* and *b*), required to satisfy mass conservation. Ignoring the effects of gas dynamics (nonuniform pressure), friction, leakage past the piston rings, and heat transfer, expressions for the squish velocity are:

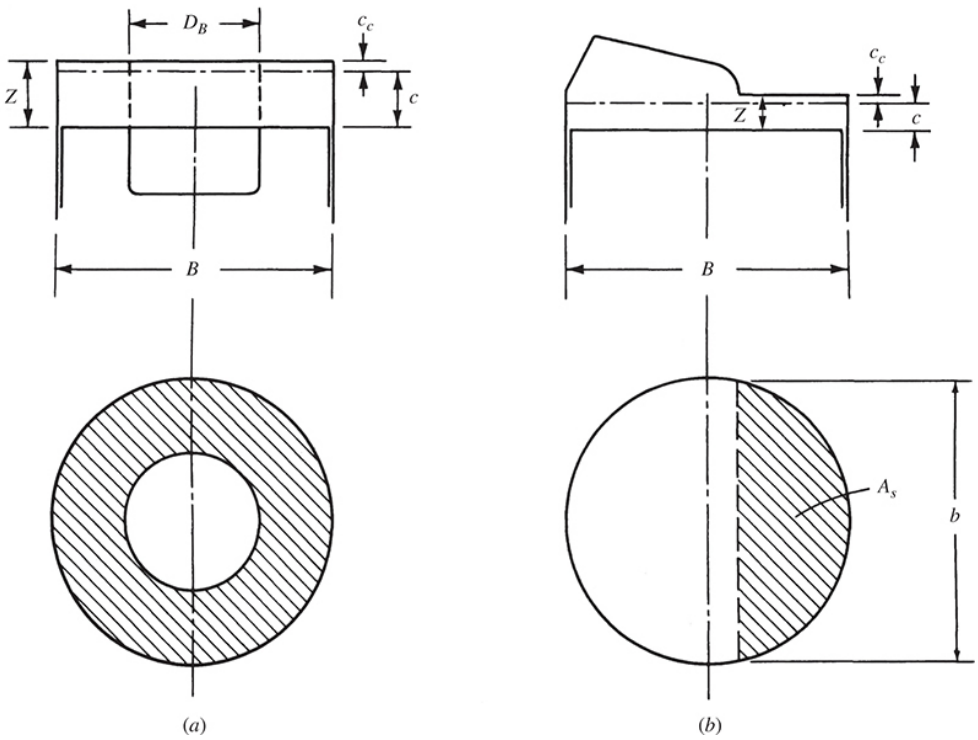


Figure 8.22 (*a*) Schematic of axisymmetric bowl-in-piston chamber for Eq. (8.42) . (*b*) Schematic of wedge chamber with transverse squish for Eq. (8.43) . Dash-dot line is piston top-center position.

1. *Bowl-in-piston chamber* (Fig. 8.22*a*):

$$\frac{v_{sq}}{S_p} = \frac{D_B}{4Z}\left[\left(\frac{B}{D_B}\right)^2 - 1\right]\frac{V_B}{A_c Z + V_B} \tag{8.42}$$

where V_B is the volume of the piston bowl, A_c is the cross-sectional area of the cylinder ($\pi B^2/4$), S_p is the instantaneous piston speed [Eq. (2.12) or (2.13)], and Z is the distance between the top of the piston crown and the cylinder head surface above the squish area ($Z = c + c_c$; $c = l + a - s$; see Fig. 2.1).

2. *Simple wedge chamber* (Fig. 8.22*b*): [31]

$$\frac{v_{sq}}{S_p} = \frac{A_s}{b(c+c_c)}\left(1 - \frac{c+c_c}{C+c}\right) \tag{8.43}$$

where A_s is the squish area, b is the width of the squish region, and C is $c/(r_c - 1)$ evaluated at the end of induction.

This theoretical squish velocity for a bowl-in-piston engine normalized by the mean piston speed $\bar{S}_p$ is shown in Fig. 8.23 for different ratios of D_B/B and clearance heights c_c. The maximum squish velocity occurs at about 10° before TC. After TC, v_{sq} is negative; a reverse squish motion occurs as gas flows out of the bowl into the clearance height region. Under motored conditions this is equal to the forward motion.

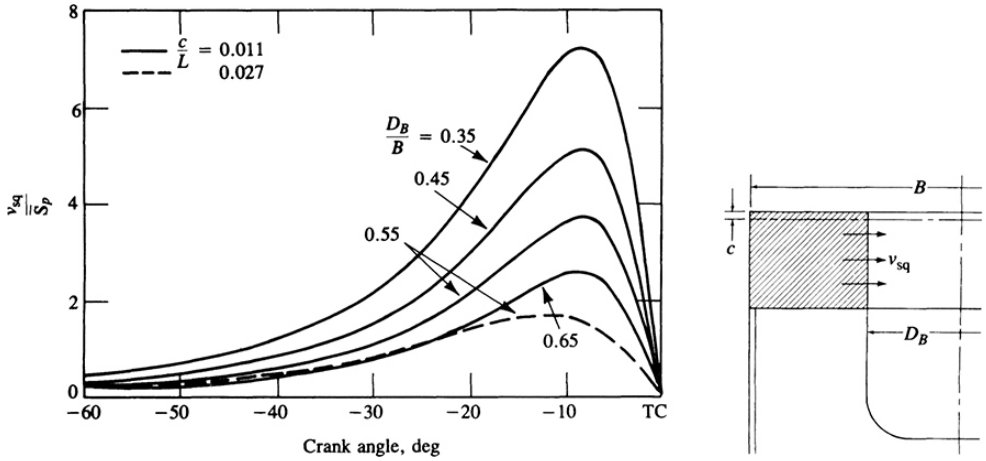


Figure 8.23 Theoretical squish velocity divided by mean piston speed for bowl-in-piston chambers for different D_B/B and c_c/L (clearance height/stroke). $B/L = 0.914$, $V_B/V_d = 0.056$, connecting rod length/crank radius = 3.76.[32]

These models omit the effects of gas inertia, friction, gas leakage past the piston rings, heat transfer, and combustion. Gas inertia and friction effects have been shown to be small. The effects of gas leakage past the piston rings and of heat transfer are more significant. The squish velocity decrement Δv_L due to leakage is proportional to the mean piston speed and a dimensionless leakage number:

$$N_L = A_{\mathrm{E,L}} \frac{\sqrt{\gamma R T_{\mathrm{IVC}}}}{N V_d} \tag{8.44}$$

where $A_{\mathrm{E,L}}$ is the effective leakage area and T_{IVC} is the temperature of the cylinder gases at inlet valve closing. Leakage was modeled as a choked flow through the effective leakage area. Values of Δv_L v_{sq} are shown in Fig. 8.24. The effect of leakage on v_{sq} is small for normal gas leakage rates. A decrement on squish velocity due to heat transfer, Δv_H has also been derived, using standard engine heat-transfer correlations (see Sec. 12.4). Values of Δv_H v_{sq} are also shown in Fig. 8.24, also. Again the effects are small in the region of maximum squish, though they become more important as the squish velocity decreases from its maximum value as the piston approaches TC. Combustion has limited impact on the squish flow at the end of compression because little of the in-cylinder charge has burned before TC. It has a much more significant effect on the "reverse squish" flow after TC, as the piston moves down the cylinder, because increasingly more of the charge has burned, and burning (usually in the more open part of the chamber) results in substantial gas expansion (by a factor of 4) which further drives flow out of the opening chamber.

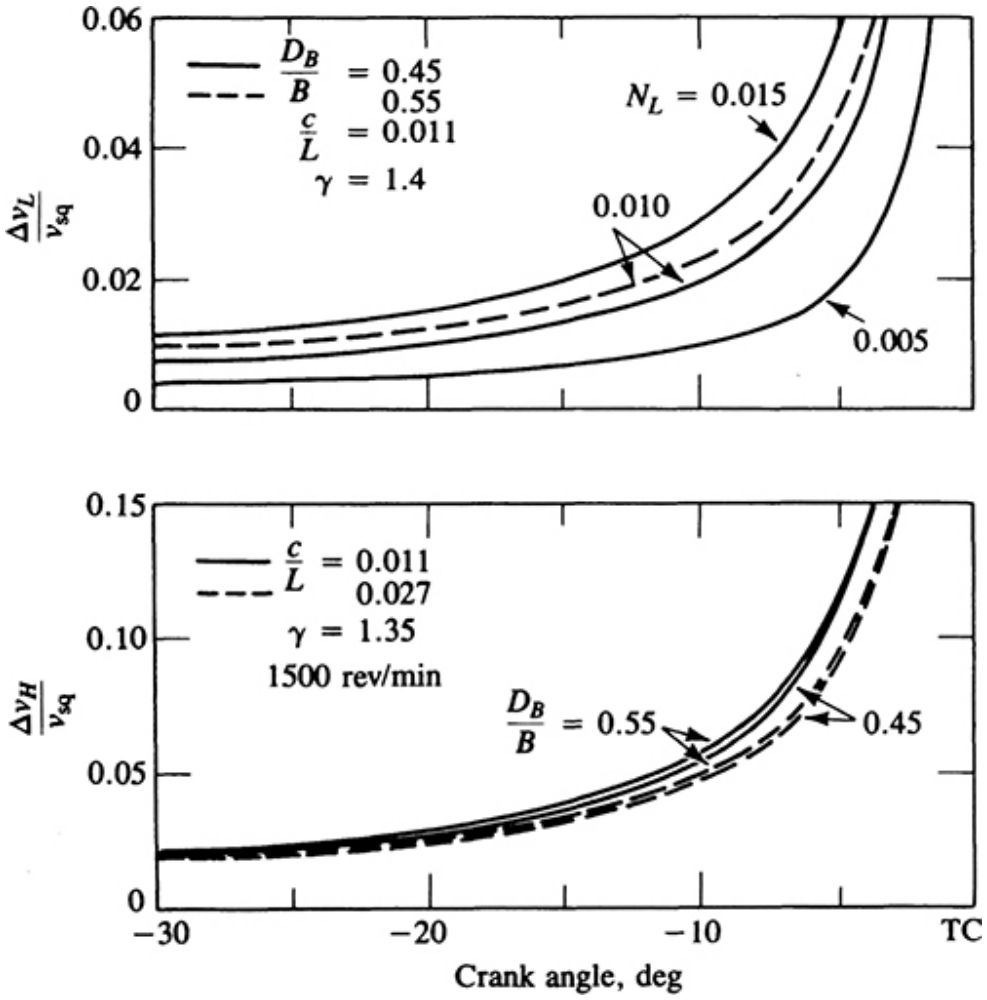


Figure 8.24 Values of squish velocity decrement due to leakage Δv_L and heat transfer Δv_H normalized by the ideal squish velocity, as a function of crank angle.[32]

Velocity measurements in engines provide good support for the above model. The ideal theory adequately predicts the dependence on engine speed. With appropriate corrections for leakage and heat-transfer effects, the above theory predicts the effects of the bowl diameter/bore ratio and clearance height on squish velocity (see Fig. 8.25). The change in direction of the radial motion as the piston moves through TC has been demonstrated under motored engine conditions. Under firing conditions, the combustion-generated gas expansion in the open portion of the combustion chamber substantially increases the magnitude of the reverse squish motion after TC.

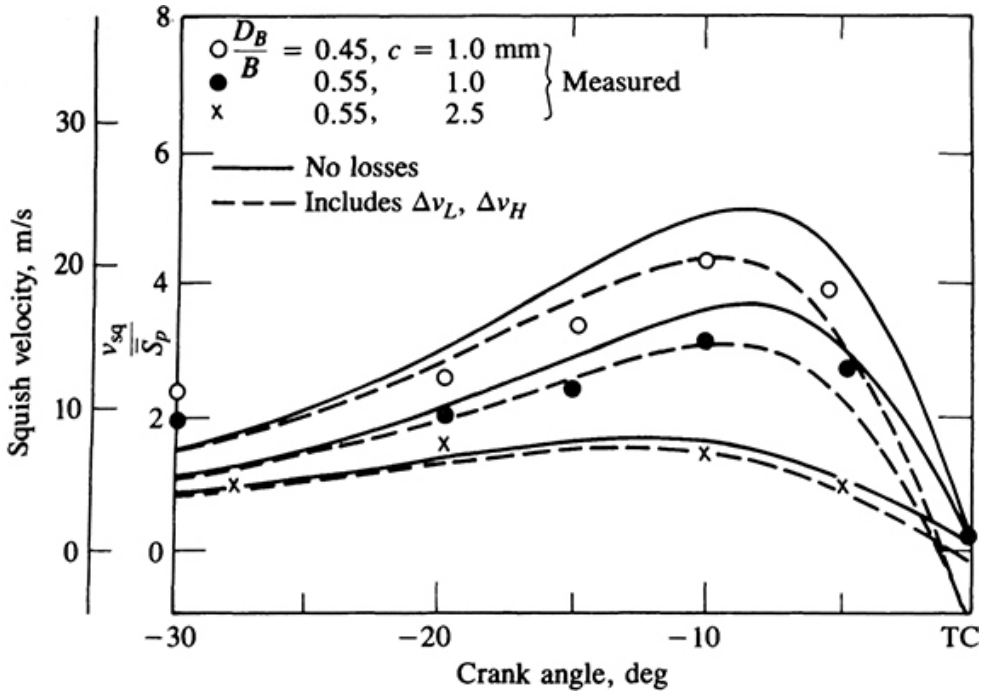


Figure 8.25 Comparison of measured squish velocities in bowl-in-piston combustion chambers, with different bowl diameter/bore ratios and clearance heights, to calculated ideal squish velocity (solid lines) and calculations corrected for leakage and heat transfer (dashed lines). Bore = 85 mm, stroke = 93 mm, 1500 rev/min. [32]

8.6 SWIRL, TUMBLE, SQUISH FLOW INTERACTIONS

The three basic in-cylinder flow patterns, discussed in Secs. 8.3 , 8.4, and 8.5, interact. For example, in spark-ignition engines, with pent-roof cylinder heads, the primary in-cylinder motion generated to increase the turbulence intensity and thus the burn rate during combustion is tumble. Often the tumble is accompanied by some swirl. Also, in diesels with deep bowl-in-piston combustion systems, the intake-generated swirl is significantly amplified by the radially-inward squish motion as the piston approaches TC, and the within-the-bowl turbulence intensity is increased by the strong swirl-squish interaction.

An important in-cylinder flow characteristic is the turbulence intensity u', or turbulent kinetic energy k [see Eq. (8.23)] around TC, since this impacts the fuel-air mixture burning rate as well as the fuel-air mixing rate in diesels. Figure 8.26 provides a summary of the TC end-of-compression turbulence intensity u' (expressed as a fraction of the mean piston speed, $\overline{S}_p$) for different SI engine combustion chamber shapes and flow configurations, and for a range of bowl-in-piston designs for diesels (shallow bowl with no or modest swirl for larger low-speed diesels; deeper bowl with swirl for medium-duty diesels; reentrant deep bowl with high swirl for smaller, high-speed, diesels). Each chamber shape matches with an engine size and speed range, an appropriate swirl or tumble level, and impact of the squish flow on the swirl. Note that smaller size cylinders can operate at higher engine speeds before intake flow choking occurs (see Sec. 6.2.5). Thus, at their maximum speed there is less time available for fuel-air mixing and for combustion than in larger engines. So mixing and burning rates must be increased as cylinder bore and stroke are decreased.

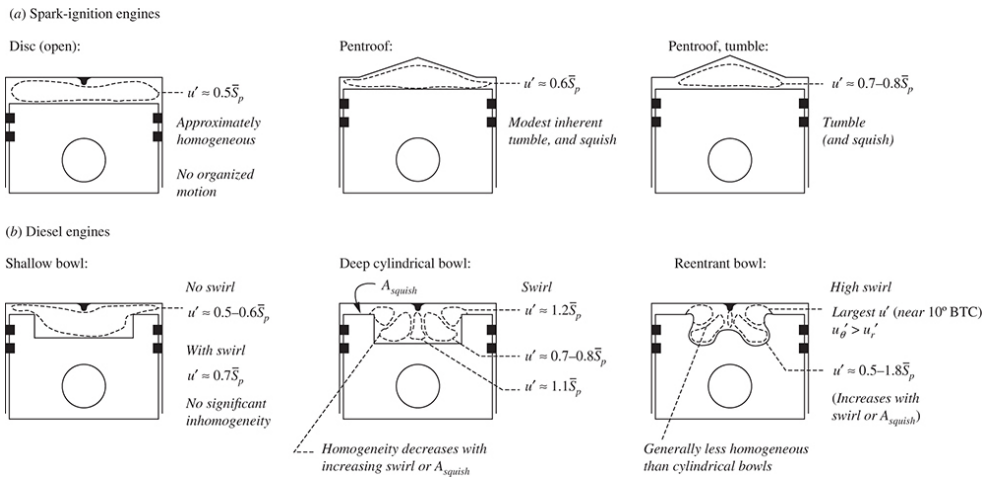


Figure 8.26 Schematics of near-TC turbulence structures and levels for a range of combustion chamber designs: (*a*) For spark-ignition engines (disc or open, and pent root chambers with increasing levels of tumble or swirling motions). (*b*) For diesel engines, with different depth and shape of bowl-in-piston chambers (shallow—larger engines, deep—medium-size engines, deep and reentrant—for small high-speed engines). [16]

With combustion chambers such as the bowl-in-piston, the tangential velocity distribution with radius will change during compression. Even if the solid-body rotation assumption is a reasonable approximation at the end of induction, the profile will distort as gas moves into the piston bowl. Neglecting the effects of friction, the angular momentum of each fluid element will remain constant as it moves radially inward. Thus the increase in tangential velocity of each fluid element as it moves radially inward is proportional to the change in the reciprocal of its radius. Measurements of the swirl velocity distribution within the cylinder of bowl-in-piston engine designs support this description. The rate of displacement of gas into the bowl depends on the bowl volume V_B, cylinder volume V, and piston speed S_p, at that particular piston position:

$$\frac{dm_B}{dt} = \frac{m_c}{L}\left(\frac{V_B}{V}\right)\left(\frac{V_d}{V}\right)S_p$$

The gas velocity into the bowl will therefore increase rapidly toward the end of the compression stroke and reach a maximum just before TC (see Sec. 8.5 where this radial "squish motion" is discussed more fully). Thus, there is a rapid increase in v_θ in the bowl as the crank angle approaches TC. The lower layers of the bowl rotate slower than the upper layers because that gas entered the bowl earlier in the compression process. [21, 23]

Velocity measurements illustrating the development of this radial distribution in tangential velocity are shown in Fig. 8.27. These measurements were made by analyzing the motion of burning carbon particles in the cylinder of an operating diesel engine from movies of the combustion process. The figure shows the engine geometry and the data compared with a model based on gas displacement and conservation of angular momentum in each element of the charge as it is displaced inward. Different swirl velocity profiles exist within and outside the bowl as the piston approaches TC. Swirl velocities within the bowl increase as TC is approached, roughly as predicted by the ideal model. Outside the bowl, the swirl velocity decreases with increasing radius due to the combined effects of friction and inward gas displacement as the clearance height decreases.

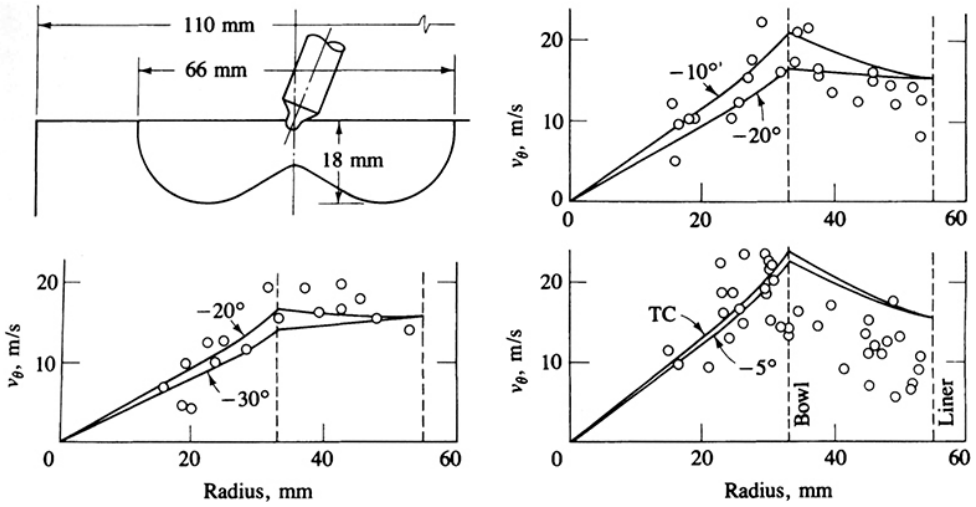


Figure 8.27 Velocity measurements as a function of radius across the combustion chamber of a firing, bowl-in-piston, direct-injection diesel engine. Schematic shows the chamber geometry. Solid lines are calculations based on the assumption of constant angular momentum for fluid elements as they move radially inward. [33]

Swirl ratios in bowl-in-piston engine designs of up to about 15 can be achieved with $D_B \approx 0.5\ B$, at top-center. Amplification factors relative to flat-topped piston swirl are typically about 2.5 to 3, some 30% lower than the ideal factor of $(B/D_B)^2$ given by Eq. (8.37) as $z \rightarrow 0$. This difference is due to the mass remaining within the clearance height which does not enter the bowl, and the effects of wall friction (enhanced by the higher gas velocities in the bowl). Sometimes the bowl axis is offset from the cylinder axis and some additional loss in swirl amplification results. [23]

The turbulence intensity with swirl was higher than without swirl (with the same chamber geometry). Integral scales of the turbulence were smaller with swirl than without. Cyclic fluctuations in the mean velocity are, apparently, reduced by swirl. There is evidence that swirl makes the turbulence intensity more homogeneous. [27]

These flow interactions become especially important when strong in-cylinder charge motion is essential—as in smaller-size high-speed direct-injection diesels—which use high swirl and deep, high-squish-area bowl-in-piston combustion systems to achieve rapid mixing and combustion. Computational fluid dynamic (CFD—see Sec. 14.5) calculations are an

essential tool for mapping out the close-to-TC end-of-compression flow field resulting from both the increase in azimuthal (tangential) velocity as the swirling charge is moved radially inward into the bowl by the strong inward squish flow generated by the piston's motion as it approaches top-center. Some general observations of this high-speed diesel example are the following. The mean flow velocities essentially scale with mean piston speed, and with swirl ratio. The flow structures within the piston bowl change as the piston moves upward, and consist of both a swirling flow (with v_θ increasing as more mass flows into the bowl) and one or more annular vortices. These are illustrated in Fig. 8.28, which shows how the spatial distribution of the high-swirl regions of the flow penetrate deeper into the bowl as the swirl ratio increases. This study also showed that the turbulent velocity fluctuations (which are not uniform) exhibit a near-linear scaling with engine speed. The turbulence intensity is about one-third of the average tangential component of velocity (the largest). Because the large-scale turbulent eddies have a time scale comparable to the mean flow deformation time, and (like the mean flow structures) these large-scale motions are constrained by the near-TC combustion chamber geometry, this near-linear scaling with engine speed is inherent. The effect of increasing flow Reynolds number is primarily to change the smaller scales of the flow and not the larger-scale structures. Since the large-scale motions are the major contributor to the turbulent fluctuations, a near-linear scaling of these velocity fluctuations would be expected. [16, 34]

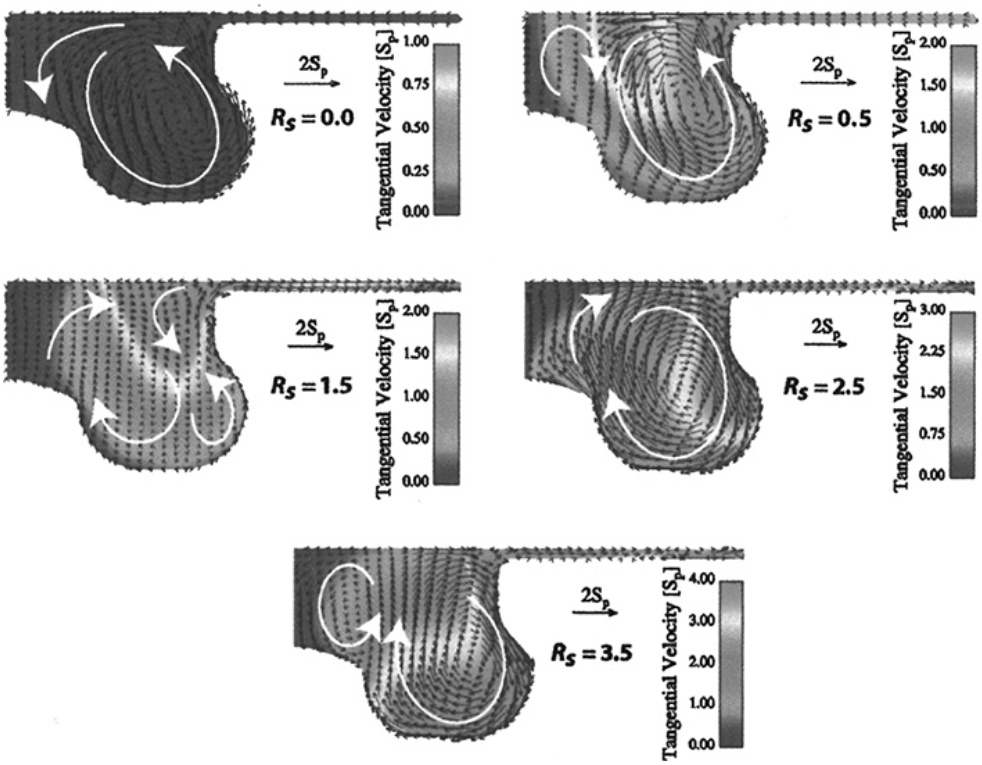


Figure 8.28 Illustration of squish-swirl interaction with reentrant bowl-in-piston high-speed diesel engine combustion system: CFD modeling results. Contours of different tangential (swirl) velocities (scaled with mean piston speed $\overline{S}_p$) in the *r-z* plane, as swirl is increased from zero to a swirl ratio of 3.5. White arrows illustrate the structure of the large-scale motions. [34]

The four-valve pent-roof gasoline SI engine provides another example of how these different types of in-cylinder flows are used. The cylinder head cross section shown in Fig. 8.18 indicates the basic geometry. The 20 ° inclination of the intake ports directs the intake flow across the cylinder head and down the liner as the piston moves down, creating a tumbling flow. Depending on intake port and cylinder head geometric details, some swirl may be generated, as well. Then, for several reasons (e.g., to create a more open chamber above much of the piston), the pent-roof chamber usually has squish regions on either side similar to the shaded areas in Fig. 8.21 *a*. As the piston moves up during the compression stroke, the tumbling vortex (which is not a natural fit inside the circular cross-section cylinder, as is a swirling flow about the cylinder axis) breaks up into several smaller vortices as its aspect ratio (width to height ratio) increases and transitions into more

irregular "turbulence" before TC. Any swirl present would persist for longer. Then, on the crank-angle scale indicated in Fig. 8.23, the inward squish flow analogous to that indicated for the two-valve open-chamber geometry of Fig. 8.21 *a* generates jet-driven shear flows and turbulence within the combustion chamber close to TC.

The effects of tumble and swirl, augmented as discussed above by the squish flow, on SI engine breathing and combustion characteristics are indicated in Fig. 8.29. The data are for four-valve per-cylinder gasoline SI engines which generate various levels of tumble: one of these has divided intake ports, and a port throttle plate in one of the two intake ports was used to generate swirl. At high airflows (at higher engine speeds and loads) with this approach, when the port throttle is opened and both ports and intake valves flow air comparably, tumble is generated. At lighter loads, the port throttle is closed, only the other intake port and valve flow air, and high levels of swirl are generated. The flow coefficient, C_f, indicates the resistance to airflow of the tumble- or swirl-generating intake ports and valves, in the cylinder head. [e]

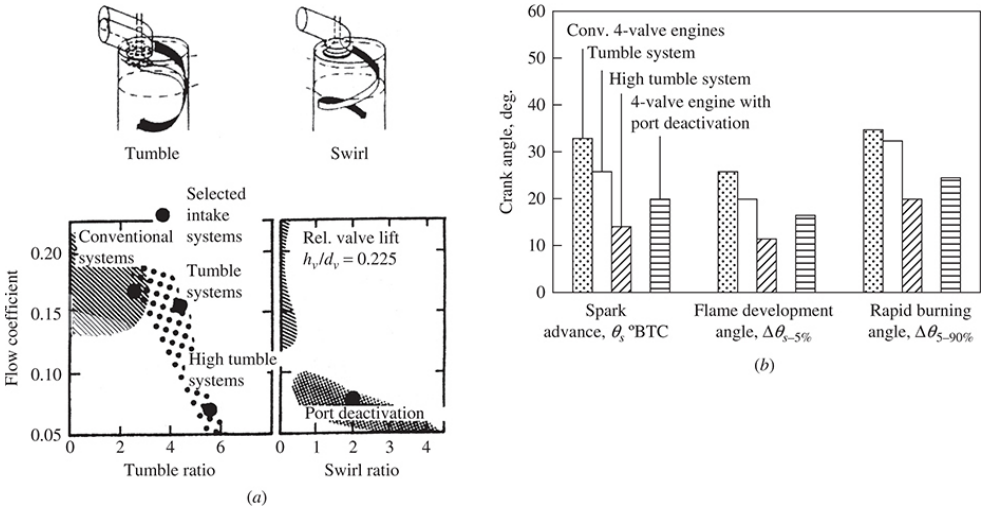


Figure 8.29 (*a*) Influence of various cylinder head designs on the resistance to airflow into the cylinder (quantified by the flow coefficient) versus tumble and swirl ratios: data points are specific examples. (*b*) Influence of these various charge motion types and levels on flame development and rapid burn durations. Engine at 3 bar bmep, 2000 rev/min, stoichiometric operation, and maximum-brake-torque (MBT) spark-timing. [35]

$$C_f = \Delta p \Big/ \left(\frac{1}{2} \rho v_a^{\,2} \right) \tag{8.45}$$

where Δp is the pressure drop across the intake port and valve, and v_a is the airflow velocity. The flow coefficients as a function of the tumble and swirl ratios for a range of cylinder head designs for multivalve engines are shown in Fig. 8.29 *a*. Increasing the charge velocities (and thus the stored kinetic energy) in these intake-generated rotating in-cylinder flows requires greater restricting of the intake flow past the valve, and thus the cylinder head has a lower flow coefficient.

The impacts of tumble and swirl on these spark-ignition engine's combustion characteristics are shown in Fig. 8.29 *b*. Increasing the tumble ratio speeds up the burning process: both the flame development angle ($\Delta \theta_{spark-5\%}$) and the burn duration ($\Delta \theta_{5-90\%}$) decrease (see Sec. 9.3.1). The port-deactivation-generated swirl is intermediate in its impacts, between medium and high tumble cases. This is in line with the broader assessment that moderate tumble is usually the appropriate choice to achieve stable SI engine combustion: it provides a fast-enough burn rate (see Sec. 9.4.1), while maintaining a relatively high flow coefficient at high speeds and loads.

Other important in-cylinder flow interactions are those between in-cylinder fuel sprays (which have significant momentum) and the charge motions within the combustion chamber at around TC in both direct-injection gasoline engines and diesel engines. These are discussed in Secs. 7.7 and 10.4 respectively. Also, in stratified DI gasoline engines of the wall-guided and flow-guided category (see Sec. 7.7.2), the DI fuel spray interacts with the flow pattern within the combustion chamber (often in a complex shape "bowl in piston"). Usually, this flow is set up by generating a swirling or tumbling flow during intake, by modifying that flow during compression through wall guiding as gas displacement into the bowl (resulting from the upward piston motion) occurs, and finally by incorporating the transverse squish-type flow as the TC piston location is approached.

8.7 PRECHAMBER ENGINE FLOWS

Historically, small high-speed diesel engines used an auxiliary combustion chamber, or pre-chamber, to achieve fast fuel-air mixing rates. In this type of diesel, the prechamber is connected to the main combustion chamber above the piston via a nozzle, passageway, or one or more orifices. Flow of air through this restriction into the prechamber during the compression process sets up high velocities in the prechamber at the time the fuel-injection process commences. This results in the required high fuel-air mixing rates. Figure 10.2 *b* shows examples of these prechamber or indirect-injection diesels. The two most common designs of auxiliary chamber are: the swirl chamber, where the flow through the passageway enters the chamber tangentially producing rapid rotation within the chamber, and the prechamber with one or more connecting orifices designed to produce a highly turbulent flow but no ordered motion within the chamber. Auxiliary chambers are sometimes used in spark-ignition engines. The torch-ignition three-valve stratified-charge engine (Fig. 1.21) is one such concept. Fuel is injected into the prechamber to create a rich mixture in the vicinity of the spark plug to promote rapid flame development. An alternative concept uses the prechamber around the spark plug to generate turbulence to enhance the early stages of combustion, but has no mixture enrichment.

The most critical phase of flow into the prechamber occurs toward the end of compression. While this flow is driven by a pressure difference between the main chamber above the piston and the auxiliary chamber, this pressure difference is small and the mass flow rate and velocity at the nozzle, orifice, or passageway can be estimated using a simple gas displacement model. Assuming that the gas density throughout the cylinder is uniform (a reasonable assumption toward the end of compression—the most critical period), the mass in the prechamber m_p is given by $m_c(\ V_p\ /V)$, where m_c is the cylinder mass, V the cylinder volume, and V_p the prechamber volume. The mass flow rate through the throat of the restriction is, therefore,

$$\dot{m} = \frac{dm_p}{dt} = -\frac{m_c V_p}{V^2}\frac{dV}{dt} \tag{8.46}$$

During compression, using the relations $dV/dt = -\ (_{\pi}\ B^{\ 2}/4)\ S_p$ where S_p is the instantaneous piston speed, $V_d = \pi_B{}^{\ 2}\ L/4$, and $\overline{S}_p = 2\ NL$, Eq. (8.46) can be written as

$$\frac{\dot{m}}{m_c N} = 2(r_c - 1)\left(\frac{V_p}{V_c}\right)\left(\frac{S_p}{\bar{S}_p}\right)\left(\frac{V_c}{V}\right) \tag{8.47}$$

where V_c is the clearance volume, $S_p/\ \bar{S}_p$ is given by Eq. (2.11) , and V/V_c is given by Eq. (2.6) . The gas velocity at the throat v_T can be obtained from $\dot{m}$ via the relation $\rho v_T A_T = \dot{m}$, the density $\rho = m_c / V$, and Eq. (8.47) becomes:

$$\frac{v_T}{\bar{S}_p} = \left(\frac{V_p}{V_c}\right)\left(\frac{\pi B^2/4}{A_T}\right)\left(\frac{S_p}{\bar{S}_p}\right)\left(\frac{V_c}{V}\right) \tag{8.48}$$

where A_T is the effective cross-sectional area of the throat. The variations of $\dot{m}/(m_c N)$ and $v_T/\ \bar{S}_p$ with crank angle during the compression process for values of r_c, V_p/V_c, and $A_T/(\pi B^2/4)$ typical of a swirl prechamber diesel are shown in Fig. 8.30. The velocity reaches its peak value about 20° before TC: very high gas velocities, an order of magnitude or more larger than the mean piston speed, can be achieved depending on the relative effective throat area. Note that as the piston approaches TC, first the nozzle velocity and then the mass flow rate decrease to zero. After TC, in the absence of combustion, an equivalent flow in the reverse direction out of prechamber would occur. Combustion in the prechamber diesel usually starts just before TC, and the pressure in the prechamber then rises significantly above the main chamber pressure. The outflow from the prechamber is then governed by the development of the combustion process, and the above simple gas displacement model no longer describes the flow.

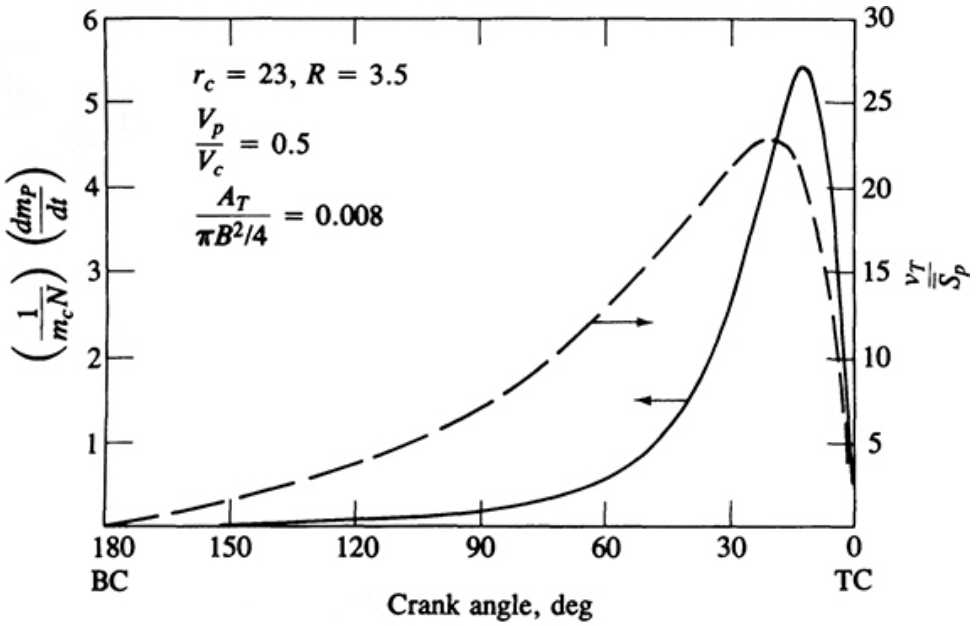


Figure 8.30 Velocity and mass flow rate at the prechamber nozzle throat, during compression, for a typical small swirl-prechamber automotive diesel.

In prechamber stratified-charge spark-ignition engines, the flow of gas into the prechamber during compression is critical to the creation of an appropriate mixture in the prechamber at the crank angle when the mixture is ignited. In the concept shown in Fig. 1.21, a very rich fuel-air mixture is fed directly to the prechamber during intake via the prechamber intake valve, while a lean mixture is fed to the main chamber via the main intake valve. During compression, the flow into the prechamber reduces the prechamber equivalence ratio to a close-to-stoichiometric value at the time of ignition. Figure 8.31 shows a gas displacement calculation of this process and relevant data; the prechamber equivalence ratio, initially greater than 3, is leaned out to unity as mass flows through the orifice into the prechamber (whose volume is 8.75% of the clearance volume). Estimates of the final equivalence ratio, based on gas displacement, for this prechamber concept are available. [36]

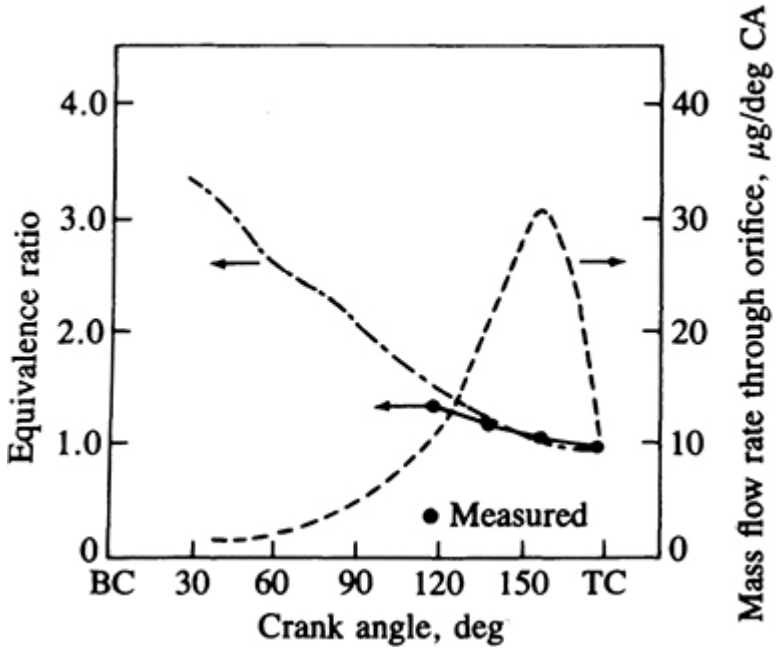


Figure 8.31 Effect of gas flow into the prechamber during compression on the prechamber equivalence ratio in a three-valve prechamber stratified-charge engine. Calculations based on gas displacement model. [37]

The velocity field set up inside the prechamber during compression is strongly dependent on the details of the nozzle and prechamber geometry. Velocities vary linearly with mean piston speed. In swirl prechambers, the nozzle flow sets up a rotating flow within the chamber. The velocities within the prechamber increase with increasing crank angle as the compression process proceeds, and reach a maximum about 20° before TC. Then, as the piston approaches TC and the flow through the passageway decreases to zero, the vortex in the swirl chamber expands to fill the entire chamber and mean velocities decay. Very high swirl rates can be achieved just before TC: local swirl ratios of up to 60 at intermediate radii and up to 20 at the outer radius have been measured. These high swirl rates produce large centrifugal accelerations.

8.8 CREVICE FLOWS AND BLOWBY

The engine combustion chamber is connected to several small volumes usually called *crevices* because of their narrow entrances. Gas flows into and out of these volumes during the engine operating cycle as the cylinder pressure and volume change.

The largest crevices are the volumes between the piston, piston rings, and cylinder wall. Some gas flows out of these regions, through the ring gaps and into the crankcase; it is called *blowby*. Other crevice volumes in production engines are the threads around the spark plug, the space between the plug center-electrode-insulator and casing, crevices between the intake and exhaust valve heads and seats in the cylinder head, and the head gasket cutout. Table 8.1 shows the size and relative importance of these crevice regions in one cylinder of a production-design spark-ignition engine determined from measurements of cold-engine components. Total crevice volume is a few percent of the clearance volume, and the piston and ring crevices are the dominant contributors. When the engine is warmed up, dimensions and crevice volumes will change some.

TABLE 8.1 Engine crevice data* [38]

	cm^3	%
Displaced volume per cylinder	465	
Clearance volume per cylinder	45	100
Compression ratio	11.2	
Piston crevice	0.782	1.74
Spark plug crevice	0.187	0.42
Valve seat crevice	0.032	0.07
Head gasket crevice	0.037	0.08
Total crevice volume	1.038	2.31

*Per cylinder; determined for cold engine.

The important crevice processes occurring during the engine cycle are the following. As the cylinder pressure rises and cylinder volume decreases during compression, unburned mixture or air is forced into each crevice region. Since these volumes are thin they have a large surface/volume ratio;

the gas flowing into the crevice cools by heat transfer to close to the wall temperature. During combustion while the pressure continues to rise, unburned mixture or air, depending on engine type, continues to flow into these crevice volumes. After flame arrival at the crevice entrance, burned gases will flow into each crevice until the cylinder pressure starts to decrease. Once the cylinder pressure starts to fall, gas flows back from each crevice into the cylinder.

The volumes between the piston, piston rings, and cylinder wall are shown schematically in Fig. 8.32. These crevices consist of a series of volumes (numbered 1 to 5) connected by flow restrictions such as the ring side clearance and ring gap. The geometry changes as each ring moves up and down in its ring groove, sealing either at the top or bottom ring surface. The gas flow, pressure distribution, and ring motion are therefore coupled. Figure 8.33 illustrates this behavior: pressure distributions, ring motion, and mass flow of gas into and out of the regions defined by planes *a*, *b, c, d,* and through the ring gap *g* are plotted versus crank angle through compression and expansion. These results come from an analysis of these regions as volumes connected by passageways, with a prescribed cylinder-pressure versus crank-angle profile coupled with a dynamic model for ring motion, and assuming that the gas temperature equals the wall temperature. [39] During compression and combustion, the rings are forced to the groove lower surfaces and mass flows *into* all the volumes in this total crevice region. The pressure above and behind the first ring is essentially the same as the cylinder pressure; there is a substantial pressure drop across each ring, however. Once the cylinder pressure starts to decrease (around 15° ATC) gas flows out of regions 1 and 2 in Fig. 8.32 into the cylinder, but continues to flow *into* regions 3, 4, and 5 until the pressure in the cylinder falls below the pressure beneath the top ring. The top ring then shifts to seal with the upper grove surface and gas flows out of regions 2, 3, and 4 (which now have the same pressure), both into the cylinder and as blowby into the crackcase. Some 5 to 10% of the total cylinder charge is trapped in these regions at the time of peak cylinder pressure. Most of this gas returns to the cylinder; about 1% goes to the crankcase as blowby. The gas flow back into the cylinder continues throughout the expansion process. In spark-ignition engines this phenomenon is a major contributor to unburned hydrocarbon emissions (see Sec. 11.4.3). It results in a loss in torque and efficiency since any fuel that does not burn during the normal combustion process does not contribute. In

diesel engines, air in these crevices during combustion is not available for combustion which additionally limits the amount of fuel that can be burned without producing soot emissions.

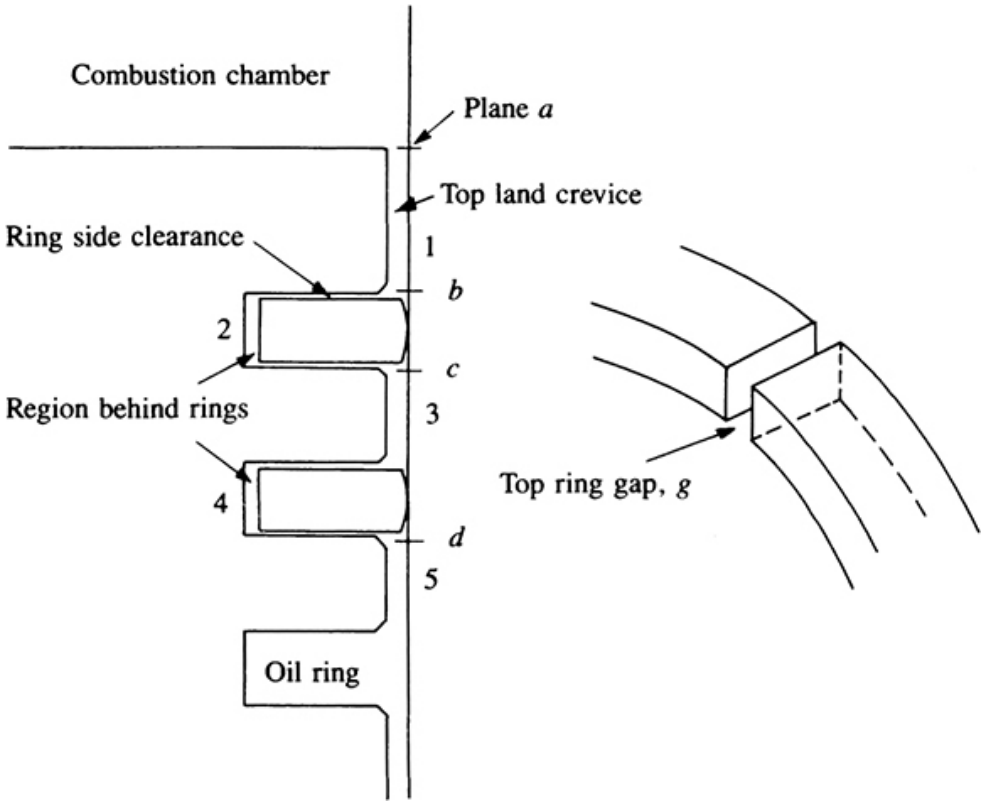


Figure 8.32 Schematic of piston and ring assembly in automotive spark-ignition engine.

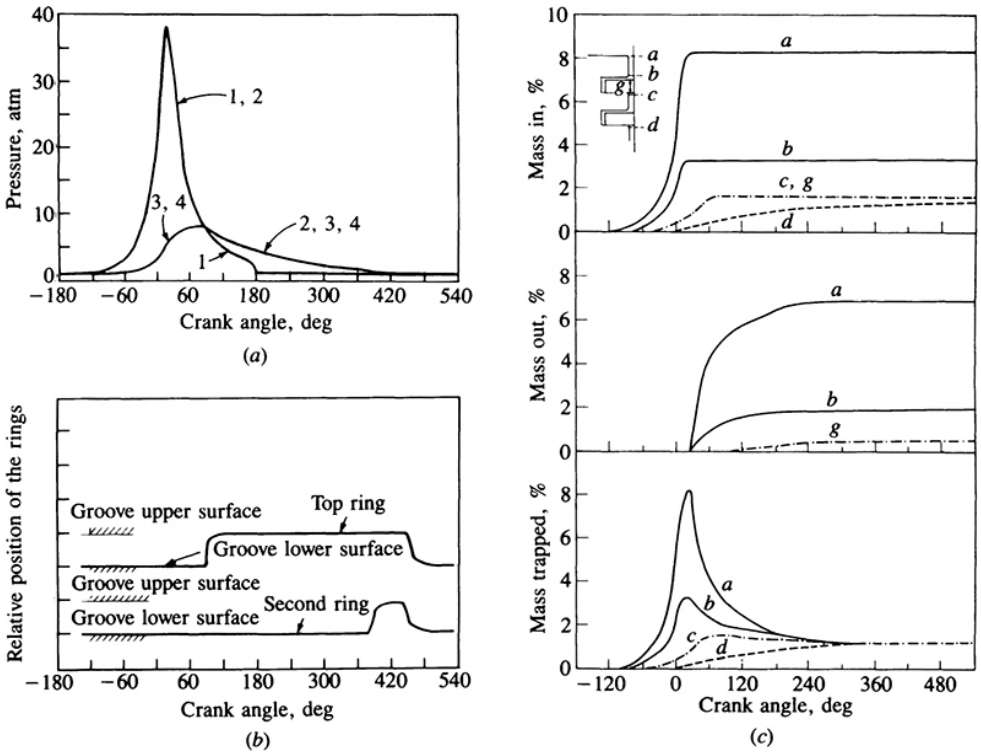


Figure 8.33 (*a*) Pressures in the combustion chamber (1), in region behind top ring (2), in region between rings (3), and behind second ring (4); (*b*) relative position of top and second rings; (*c*) percentage of total cylinder mass that flows into and out of the different crevice regions across planes *a, b, c,* and *d* and through the ring gap *g* in Fig. 8.32, and the percentage of mass trapped beneath these planes, as a function of crank angle. Automotive spark-ignition engine at wide-open throttle and 2000 rev/min. [39]

There is substantial experimental evidence to support the above description of flow in the piston ring crevice region. In a special square-cross-section flow visualization engine, both the low-velocity gas expansion out of the volume above the first ring after the time of peak pressure, and the jet-like flows through the top ring gap later in the expansion process when the pressure difference across the ring changes sign have been observed. Figure 8.34 shows these flows with explanatory schematics.

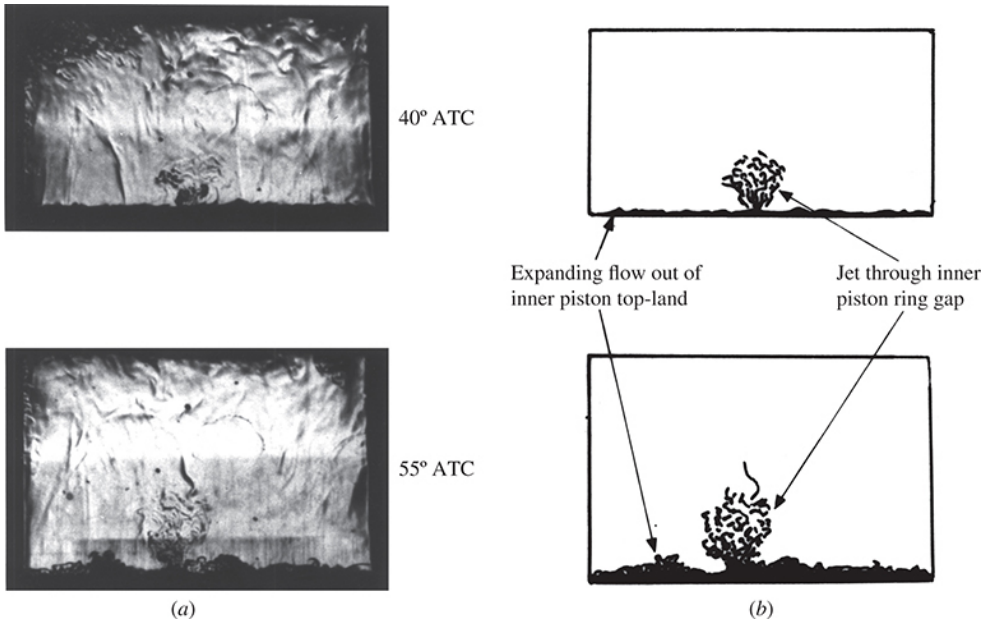


Figure 8.34 Schlieren photographs of the flow out of the piston-cylinder wall crevices during the expansion stroke. A production piston was inserted into the square cross-section piston of a special visualization engine. Gas flows, at low velocity, out of the top land-crevice entrance all around the production piston circumference once the cylinder pressure starts decreasing early in the expansion stroke. Gas flows out of the ring gap as a jet once the pressure above the ring falls below the pressure beneath the ring. [39]

Blowby is defined as the gas that flows from the combustion chamber past the piston rings and into the crankcase. It flows through any leakage paths in the piston-bore-ring assembly driven by the high combustion-chamber pressure. If there is good contact between the compression rings and the bore, and the rings and the bottom of the grooves, then the only leakage path of consequence is the ring gap. Blowby of gases from the cylinder to the crankcase removes gas from these crevice regions and thereby prevents some of the crevice gases from returning to the cylinder. Crankcase blowby gases used to be vented directly to the atmosphere and constituted a significant source of HC emissions. The crankcase is now vented to the engine intake system and the blowby gases are recycled. Blowby at a given speed and load is controlled primarily by the greatest flow resistance in the flow path between the cylinder and the crankcase. This is the smallest of the first and

second ring-gap areas. Figure 8.35 shows how measured blowby flow rates increase linearly with the smallest gap area.[40]

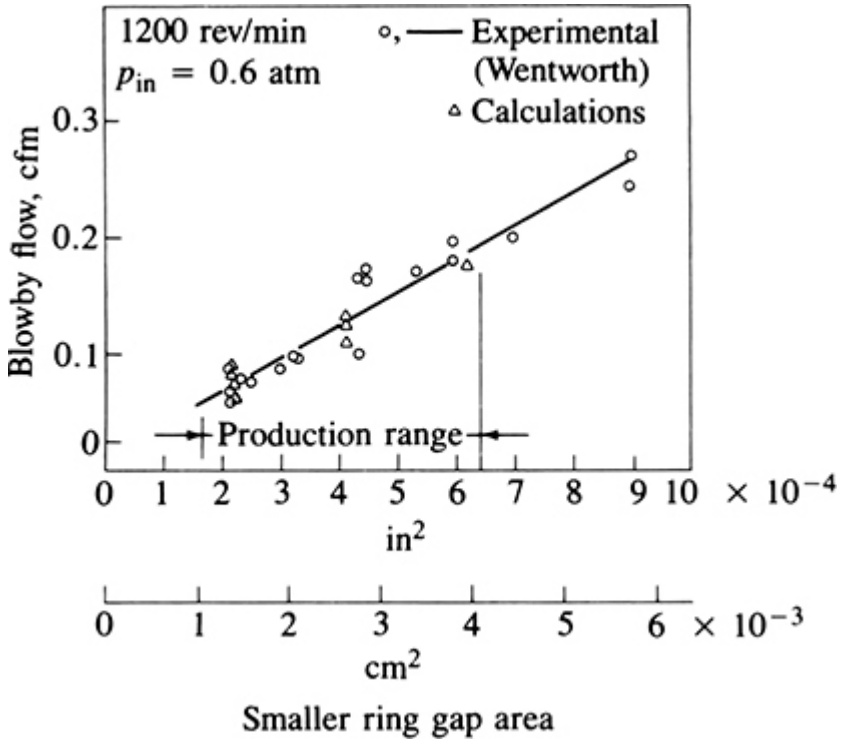


Figure 8.35 Measured blowby for one cylinder of an automobile spark-ignition engine as a function of the smallest ring gap area, compared with blowby calculations based on flow model described in text.[39, 40] Blowby of 0.1 cfm (some 0.05 liter/s) for this engine at these conditions is about 2% of the engine airflow.

Calculations of blowby based on the model described earlier are in good agreement with this data. [39] Extrapolation back to the zero gap area gives nearly zero blowby. Note, however, that if the bore finish is rough or if, through bore distortion, the rings do not contact the bore all around, or if the first or second rings lift off the bottom of their groove, this linear relationship may no longer hold. Blowby in production gasoline spark-ignition engines is typically about 1% of the engine airflow.

8.9 FLOWS GENERATED BY PISTON CYLINDER-WALL INTERACTION

Because a boundary layer exists on the cylinder wall, the motion of the piston generates unusual flow patterns in the corner formed by the cylinder wall and the piston face. When the piston is moving away from top-center a sink-type flow occurs. When the piston moves toward top-center a vortex flow is generated. Figure 8.36 shows schematics of these flows (in a coordinate frame with the piston at rest). The vortex flow has been studied because of its effect on gas motion at the time of ignition and because it is thought to be a mechanism for removing hydrocarbons off the cylinder wall during the exhaust stroke (see Sec. 11.4.3).

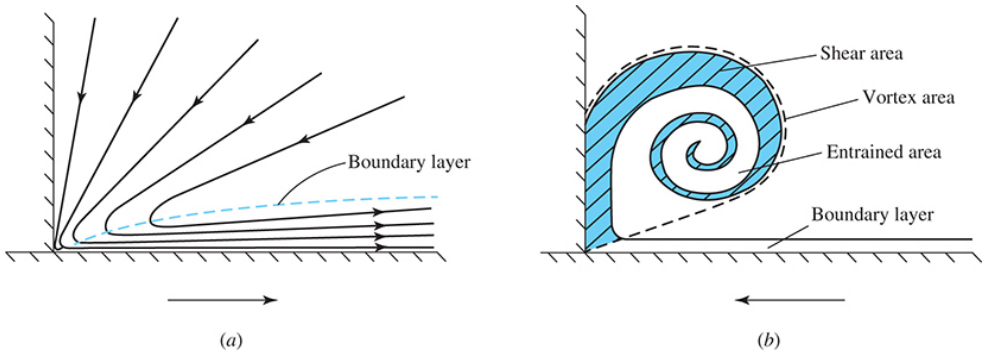


Figure 8.36 Schematics of the flow pattern set up in the piston face-cylinder wall corner, in piston-stationary coordinates, due to the boundary layer on the cylinder wall. Piston crown on left; cylinder wall at bottom. (*a*) Sink flow set up during intake and expansion; (*b*) vortex flow set up up during compression and exhaust. [41] Arrows show cylinder wall velocity relative to piston.

This vortex flow has been studied in cylinders with water as the fluid over the range of Reynolds numbers typical of engine operation. [41,4 2] Laminar, transition, and turbulent flow regimes have been identified. It has been shown that a quasi-steady flow assumption is valid and that

$$\frac{A_v}{L^2} = f\left(\frac{v_w L}{\nu}\right)$$

where A_v is the vortex area (area inside the dashed line in Fig. 8.35), L is the stroke, v_w is the wall velocity in piston stationary coordinates ($v_w = S_p$ in the engine), ν is the kinematic viscosity, and ($v_w L/\nu$) is a Reynolds number.

For the laminar flow regime, a good assumption is that A_v is proportional to the shear area in the vortex (shown cross-hatched in Fig. 8.36), which equals the boundary-layer area; this can be estimated from boundary-layer theory. In the turbulent flow regime an entrainment theory was used which assumed that the rate of change of vortex area was proportional to the product of the exposed perimeter of the vortex and the velocity difference between the vortex and the stationary fluid ($\approx v_w$). The relevant relationships are:

$$\text{For} \quad (v_w L/\nu) \leq 2\times 10^4 \qquad \frac{A_v}{L^2} = \left(\frac{v_w L}{\nu}\right)^{-1/2} \tag{8.49a}$$

$$\text{For} \quad (v_w L/\nu) \geq 2\times 10^4 \qquad \frac{A_v}{L^2} = 0.006 \tag{8.49b}$$

Figure 8.37 shows these two theories correlated against hydraulic analog data.

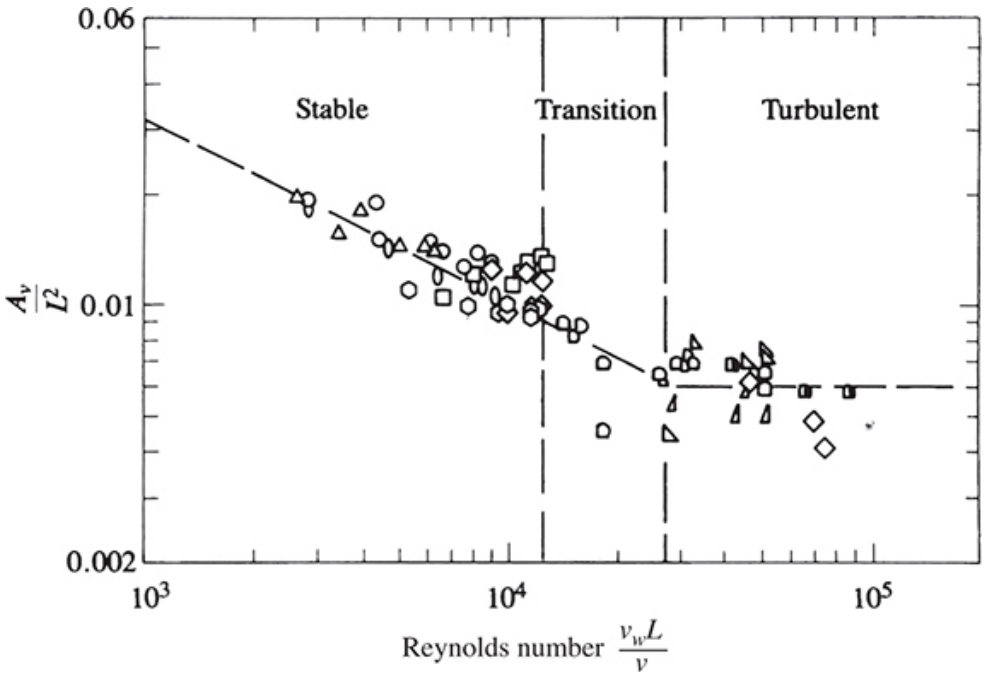


Figure 8.37 Ratio of area of vortex in piston face-cylinder wall corner to square of stroke, as a function of Reynolds number based on piston velocity, for piston moving toward the cylinder head.[41]

These theories are for constant values of kinematic viscosity, v. During compression, v decreases substantially as the gas temperature and pressure increase (v decreases by a factor of 4 for a compression ratio of 8). This will decrease the size of the vortex until the turbulent regime is reached. During the exhaust stroke following blowdown, v will remain approximately constant as the pressure and temperature do not change significantly. Typical parameter values at 1500 rev/min are: $\overline{S}_p = 5$ m/s, $L = 0.1$ m; average values of v are 1.2×10^{-5} and 1.4×10^{-4} m^2/s for compression and exhaust stroke, respectively. Hence, a Reynolds number for the compression stroke is 4×10^4, $A_v / L^2 \approx 0.006$, and the vortex diameter $d_v \approx 0.09\, L$. For the exhaust stroke, the Reynolds number is 4×10^3, $A_v / L^2 \approx 0.015$, and $d_v \approx 0.14\, L$. Thus the vortex dimensions at the end of the upward stroke of the piston are comparable to the engine clearance height.

This vortex flow has been observed in an operating engine. Figure 8.38 shows schlieren photographs taken during the exhaust stroke in a special square-cross-section flow visualization spark-ignition engine. The accompanying schematic identifies the vortex structure which is visible in the photo because the cool boundary-layer gas is being scraped off the cylinder wall by the upward-moving piston and "rolled up." The vortex diameter as the piston approaches TC is about 20% of the bore, larger than the engine's clearance height.

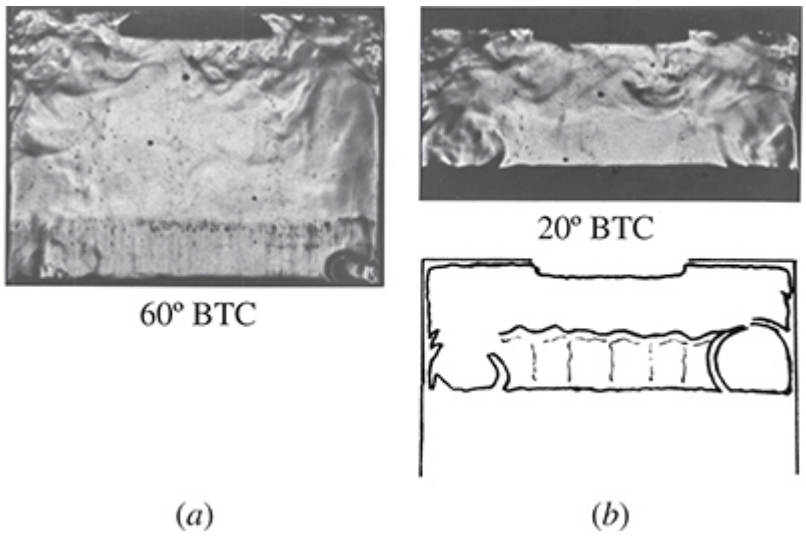


Figure 8.38 Schlieren photographs of in-cylinder flow during later stages of exhaust stroke. (a) Growing vortex in the piston face–cylinder wall corner and turbulent flow toward the exhaust valve are apparent at 60° before top center (BTC); (*b*) At 20° BTC, the vortex has grown to a diameter of about 0.2 *B*. Sketch illustrates the flow structure. [39]

PROBLEMS

8.1. **(a)** Estimate the ratio of the maximum gas velocity in the center of the hollow cone inlet jet to the mean piston speed from the data in Fig. 8.1.

(b) Compare this ratio with the ratio of inlet valve pseudo flow velocity determined from Fig. 6.15 to the mean piston speed at the same crank angle. The engine is that of Fig. 1.4.

(c) Are the engine velocity data in (*a*) consistent with the velocity calculated from the simple piston displacement model of (*b*)? Explain.

8.2. Given the relationship between turbulence intensity and mean piston speed [Eq. (8.23)] and that the turbulence integral scale is ≈ 0.2 × clearance height, use Eqs. (8.14) and (8.15) to estimate the following quantities for a spark-ignition engine with bore = stroke = 86 mm, $r_c = 9$, at 1000 and 5000 rev/min and wide-open throttle:

(a) Mean and maximum piston speed, maximum gas velocity through the inlet valve (see Prob. 8.1)

(b) Turbulence intensity, integral length scale, micro length scale, and Kolmogorov length scale, all at TC

8.3. The swirl ratio at the end of induction at 2000 rev/min in a direct-injection diesel engine of bore = stroke = 100 mm is 4.0. What is the average tangential velocity (evaluated at the inlet valve-axis radial location) required to give this swirl ratio? What is the ratio of this velocity to the mean piston speed and to the mean flow velocity through the inlet valve estimated from the average valve open area and open time?

8.4. **(a)** Derive a relationship for the depth (or height) h_B of a disc-shaped bowl-in-piston direct-injection diesel engine combustion chamber in terms of compression ratio r_c, bore B, stroke L, bowl diameter D_B, and top-center cylinder-head to piston-crown clearance c. For $B = L = 100$ mm, $r_c = 16$, $D_B = 0.5B$, $c = 1$ mm, find the fraction of the air charge within the bowl at TC.

(b) If the swirl ratio at the end of induction at 2500 rev/min is 3, find the swirl ratio and average angular velocity in the bowl-in-piston chamber of dimensions given above. Assume the swirling flow is always a solid-body rotation. Compare the tangential velocity at the bowl edge with the mean piston speed. Neglect any friction effects.

(c) What would the swirl ratio be if the top-center clearance height was zero?

8.5. Using Eq. (8.43) and Fig. 8.22*b* plot the squish velocity divided by the mean piston speed at 10° BTC (the approximate location of the maximum) as a function of squish area expressed as a percentage of the cylinder cross section, $A_s/(\pi B^2/4) \times 100$, from 50 to 0%. $r_c = 10$, $c_c/B = 0.01$, $B/L = 1$, $R = l/a = 3.5$.

8.6. The total crevice volume in an automobile spark-ignition engine is about 2% of the clearance volume. If the gas in these crevice regions is close to the wall temperature (450 K) and at the cylinder pressure, estimate the fraction of the cylinder mass within these crevice regions at these crank angles: inlet valve closing (50° ABC), spark discharge (30° BTC), maximum cylinder pressure (15° ATC), exhaust valve opening (60° BBC), TC of the exhaust stroke. Use the information in Fig. 1.9 for your input data, and assume the inlet pressure is 0.67 bar.

8.7. The piston crevice between the piston top land and cylinder wall, above the first ring, is the major source of engine-out hydrocarbon emissions. Unburned mixture in the cylinder is forced into this crevice as the cylinder pressure increases and flows out of this crevice as the cylinder pressure decreases.

Calculate the mass fraction of the cylinder gas, which is in the crevice between the piston and cylinder wall and above the first piston ring, at the time of peak pressure when the engine is cold (at ambient temperature) and fully warmed up. Assume that gas trapped in this crevice is all unburned mixture (the maximum amount).

Engine geometry and operating conditions:

Bore = 86 mm, stroke = 86 mm

Piston diameter above top ring = 85.3 mm

Distance from piston crown top to top ring = 8 mm

Volumetric efficiency = 0.85

Ambient temperature = 300 K

Piston temperature (cold) = 300 K

Piston temperature at warmed-up conditions = 460 K

Cylinder wall temperature at fully warmed-up condition = 360 K

Coefficient of linear expansion of piston = 2.41×10^{-5} m/K

Coefficient of linear expansion of cylinder wall = 1.29×10^{-5} m/K

$\varphi = 1.0$, residual fraction = 0.08, fuel = isooctane

Max. cylinder pressure = 3.5 MPa

The gas temperature in the crevice can be assumed to be the average between piston and cylinder wall temperature.

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[a] There is considerable debate as to whether the fluctuating components of the velocity $U(\theta,i)$ defined by Eq. (8.7) , cycle-by-cycle fluctuations in the mean velocity and fluctuations in time about the individual cycle mean, are physically distinct phenomena. The high-frequency fluctuations in velocity about the individual cycle mean are usually defined as "turbulence." The low-frequency fluctuations are generally attributed to the variations in the mean flow between individual cycles, a phenomenon that is well established.

[b] $\overline{D}/\overline{D}t$ is the substantive or convective derivative operator $\partial/\partial t + \nabla\bullet$, following a fluid element along its streamline.

[c] Note that because of the valve and combustion chamber of this particular engine, the ratio of $\overline{U}$ to $\overline{S}_p$ is higher than is typical of normal engine geometries.

[d] Note that the turbulence is approximately isotropic since the fluctuating velocities can be thought of as having three orthogonal components (one longitudinal and two lateral), with $l_{I,long}$ often about $2l_{I,lat}$.

[e] C_f is determined in steady airflow measurements through the cylinder head from the pressure drop across the cylinder head, and the air mass flow rate.

CHAPTER 9

Combustion in Spark-Ignition Engines

9.1 ESSENTIAL FEATURES OF PROCESS

The combustion process is an especially important part of the internal combustion engine's operating cycle. Combustion must release the chemical energy of the fuel—the primary source of energy for the engine—in a relatively short time period between the compression and expansion processes, thereby producing the high-pressure high-temperature burned gases that then expand within the cylinder transferring work to the piston. A robust combustion process is important for smooth and reliable engine operation. The combustion process must be fast, that is, it must occupy a small fraction of the total cycle time, so that the engine's energy conversion process is efficient, and highly repeatable so that variations from one cycle to the next are small enough to be barely noticeable.

Combustion is also important for its impact on several other engine characteristics or requirements. The processes by which the engine's emissions form within the cylinder are closely linked with the details of the combustion process. The antiknock and volatility requirements of fuels for spark-ignition (SI) engines and the ignition quality of diesel fuels are dictated by each engine's combustion process. Even the gas exchange process of the engine, and hence its power, are affected by what the designer must do to achieve good combustion.

9.1.1 Combustion Fundamentals

Combustion in engines takes place in a *flame*. A flame is the (confined) region within which the fuel oxidizing reactions and chemical energy release occur. Engine flames are normally thin: the actual reaction region is a fraction of a millimeter thick. In the spark-ignition engine, the fuel-vapor air charge is well mixed so the flame is a *premixed flame.* The speed at which such a premixed flame propagates into the unburned gas ahead of it is called the *flame speed* or burning velocity. In a laminar or well-ordered flow, this velocity S_L is a characteristic property of the unburned mixture ahead of the flame: it depends on the composition of the mixture, the properties of the fuel, and the mixture temperature and pressure. The values of S_L are usually less than about 1 m/s. Figure 9.1 shows the structure of a laminar premixed flame. In flame-stationary coordinates, the unburned mixture—fuel vapor and air—enters the flame from the left at temperature T_u. (In unburned-mixture stationary coordinates, the flame moves toward the left into that mixture.) The unburned mixture temperature starts to rise in the preheat zone, and steadily increases due to the thermal energy conducted through the gas moving through the conduction zone, that thermal energy coming from the reaction zone where the fuel oxidation chemical reactions occur (after the unburned mixture reaches its ignition temperature T_i). It is in this reaction zone that the bulk of the chemical energy release occurs: the thermal energy produced is conducted into the unburned mixture moving through the conduction zone as described above, and convected out of the flame in the (hot) burned gases (at temperature, T_b).

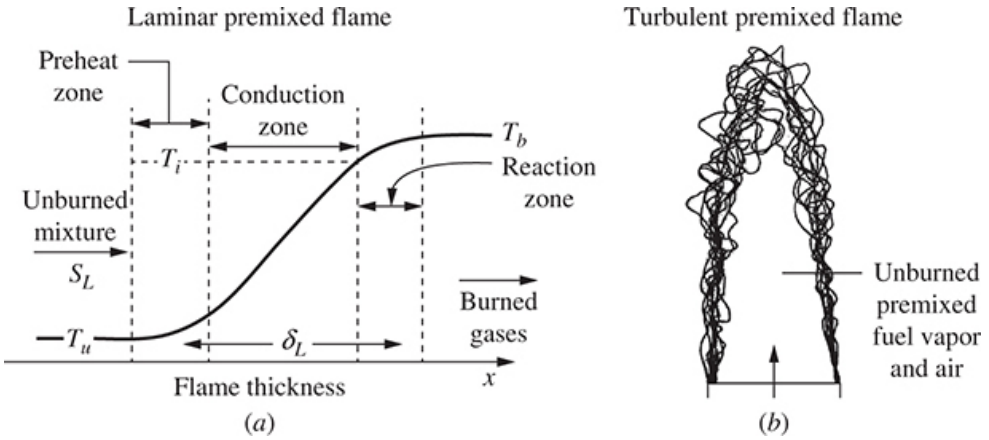


Figure 9.1 Structure of a premixed flame: (*a*) temperature profile within the different regions of a laminar premixed flame; (*b*) several realizations of

a turbulent premixed flame attached to a tube supplying premixed fuel vapor and air.

This conduction of thermal energy that steadily raises the unburned mixture temperature, along with the diffusion of reactive species within the flame, is what causes the flame to propagate. The flame speed thus scales with the temperature gradient in this conduction zone, $(T_b - T_u)/\delta_f$. A premixed flame with a stoichiometric or slightly fuel-rich mixture (which produces the highest T_b for a given T_u) has the highest laminar flame speed. Lean mixtures, and mixtures diluted with burned residual gas and exhaust-gas recirculation (EGR), decrease T_b and lengthen δ_f, so their flame speeds are lower (see Sec. 9.3.3 where the dependence of S_L on mixture composition, temperature, and pressure, is described).

In spark-ignition engines, the fuel and air are mixed together either in the intake system or in the cylinder where mixing with residual gas takes place, and are then compressed. Combustion is initiated toward the end of the compression stroke at the spark plug by an electrical discharge. Following inflammation, a *turbulent flame* develops, propagates through this essentially premixed fuel vapor, air, burned gas mixture until it reaches the combustion chamber walls, and then extinguishes. Photographs of this process taken in operating engines illustrate its essential features. Figure 9.2 shows a sequence of frames from a high-speed movie of the combustion process in a special single-cylinder engine with a glass piston crown.[1] The spark discharge is at 30° before top center (BTC). The flame first becomes visible in the photos at about 20° BTC. The flame, approximately circular in outline in this view through the piston, propagates outward from the spark plug location. The blue light from the flame is emitted most strongly from the flame front. The irregular shape of the turbulent flame front is apparent. At TC the flame diameter is about two-thirds of the cylinder bore. The flame reaches the cylinder wall farthest from the spark plug about 15° ATC, but combustion continues around parts of the chamber periphery for another 10 degrees. At about 10° ATC, additional radiation—initially white, turning to pinky-orange—centered at the spark plug location is evident. This afterglow comes from the gases behind the flame which burned earlier in the combustion process as they are compressed to the highest temperatures

attained within the cylinder (at about 15° ATC), while the rest of the charge burns. [2, 3]

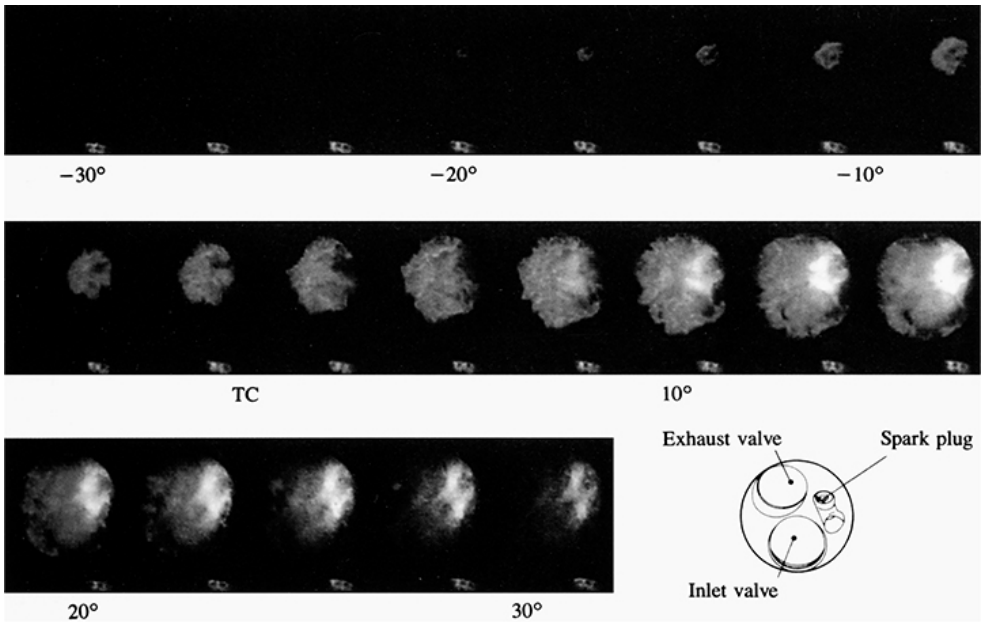


Figure 9.2 Sequence of images of flame and burned-gas radiation from high-speed movie of spark-ignition engine combustion process, taken through glass piston crown. Ignition timing 30° BTC, light load, 1430 rev/min, (*A/F*) = 19. [1]

Premixed SI engine flames become turbulent as they develop from the spark discharge, and their propagation rate is then defined by their *turbulent flame speed*. Figure 9.1 *b* shows several realizations of a turbulent premixed flame, attached to a tube through which premixed fuel vapor and air enter the unburned-mixture region inside the conical flame. Each line is a short-time-exposure realization of the irregular randomly moving thin flame sheet above the burner tube. The burned gases emerge, moving upward and outward from the flame contour. These flame profiles indicate that this turbulent flame is a *wrinkled laminar flame.*

The turbulent flow inside the engine cylinder during combustion both convects and distorts or wrinkles the thin reaction sheet flame through its combination of bulk or mean flow pattern and local turbulence or random velocity fluctuations. The higher the local turbulence, the faster the flame propagates and becomes wrinkled. If engine flames were not turbulent, with

a turbulent flame speed several times the laminar flame speed, engines could not operate satisfactorily because combustion, and the pressure rise it produces, would be much too slow. Because engine turbulence scales with piston speed, the combustion rate also scales with piston speed and thus is fast enough throughout the full crankshaft speed range desired for satisfactory engine operation (see Sec. 8.2.2). However, the turbulent flow primarily affects the gross or larger-scale characteristics of the flame (of order 1 mm) such as its average burning velocity. Locally, the thin reaction sheet flame propagates forward into the adjacent unburned mixture at the laminar flame speed.

Combustion in the compression-ignition or diesel engine occurs through a different sequence of processes. Fuel is injected directly into the engine cylinder toward the end of the compression process just before the desired start of combustion. The injected liquid fuel jets atomize, the fuel droplets move through the high-temperature high-pressure air in the combustion chamber, vaporize, and the fuel vapor mixes with air. The high temperature fuel-vapor air mixture thus produced then ignites spontaneously and burns rapidly. Combustion continues as combustible mixture is prepared by the above mixing process. Thus, for much of the diesel's combustion process, the fuel-burning rate is controlled by the rate at which fuel mixes with air. Such "non-premixed flames" are usually termed *diffusion flames* because the (gaseous) fuel and the air must diffuse into each other and then react in a thin reaction sheet close to where they reach the stoichiometric mixture proportions (see Fig. 9.3). A candle flame, also illustrated, is a common example of a diffusion flame. Because the in-cylinder engine flow is turbulent, the diesel combustion process occurs primarily as a turbulent (unsteady) diffusion flame. The spontaneous ignition due to the high air temperature in the cylinder when injection commences, and that fuel-air mixing controls the burning rate, are the distinguishing features of the diesel combustion process (the theme of Chap. 10).

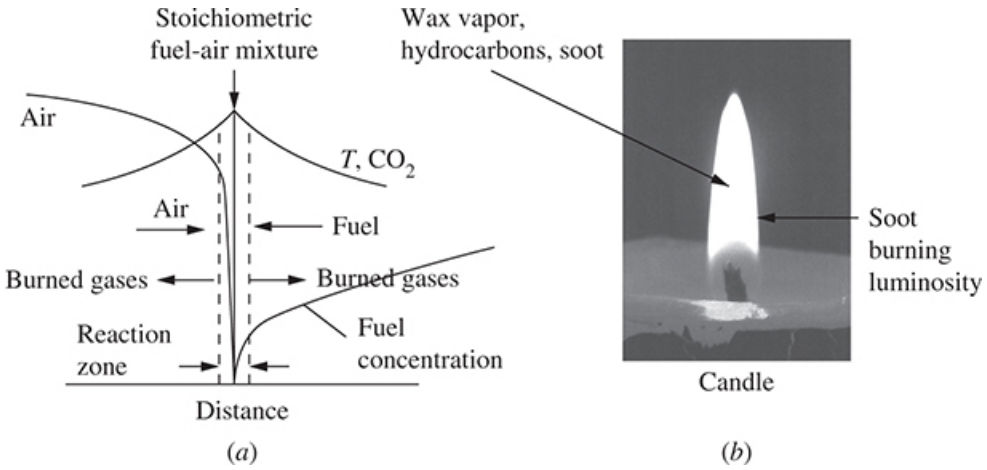


Figure 9.3 Non-premixed or diffusion flame, the basic diesel combustion process. (*a*) Diffusion flame structure. (*b*) Example of diffusion flame (candle flame): fuel-rich mixture (wax vapor, partly reacted hydrocarbons, some air, soot particles) inside the luminous flame; flame luminosity results from soot particles burning up in the thin flame reaction zone.

9.1.2 SI Engine Combustion Process

Additional features of the SI engine combustion process are evident from the data in Fig. 9.4, taken from several consecutive cycles of an operating spark-ignition engine. The cylinder pressure, fraction of the charge mass which has burned (determined from the pressure data), and fraction of the cylinder volume enflamed by the front (determined from photographs like Fig. 9.2) are shown, all as a function of crank angle. [4] Following spark discharge, there is a period during which the energy release from the developing flame is too small for the pressure rise due to combustion to be discerned. Then, as the flame continues to grow and propagate across the combustion chamber, the pressure steadily rises above the value it would have in the absence of combustion. The pressure reaches a maximum after TC but before the cylinder charge is fully burned, and then decreases as the cylinder volume continues to increase during the remainder of the expansion stroke.



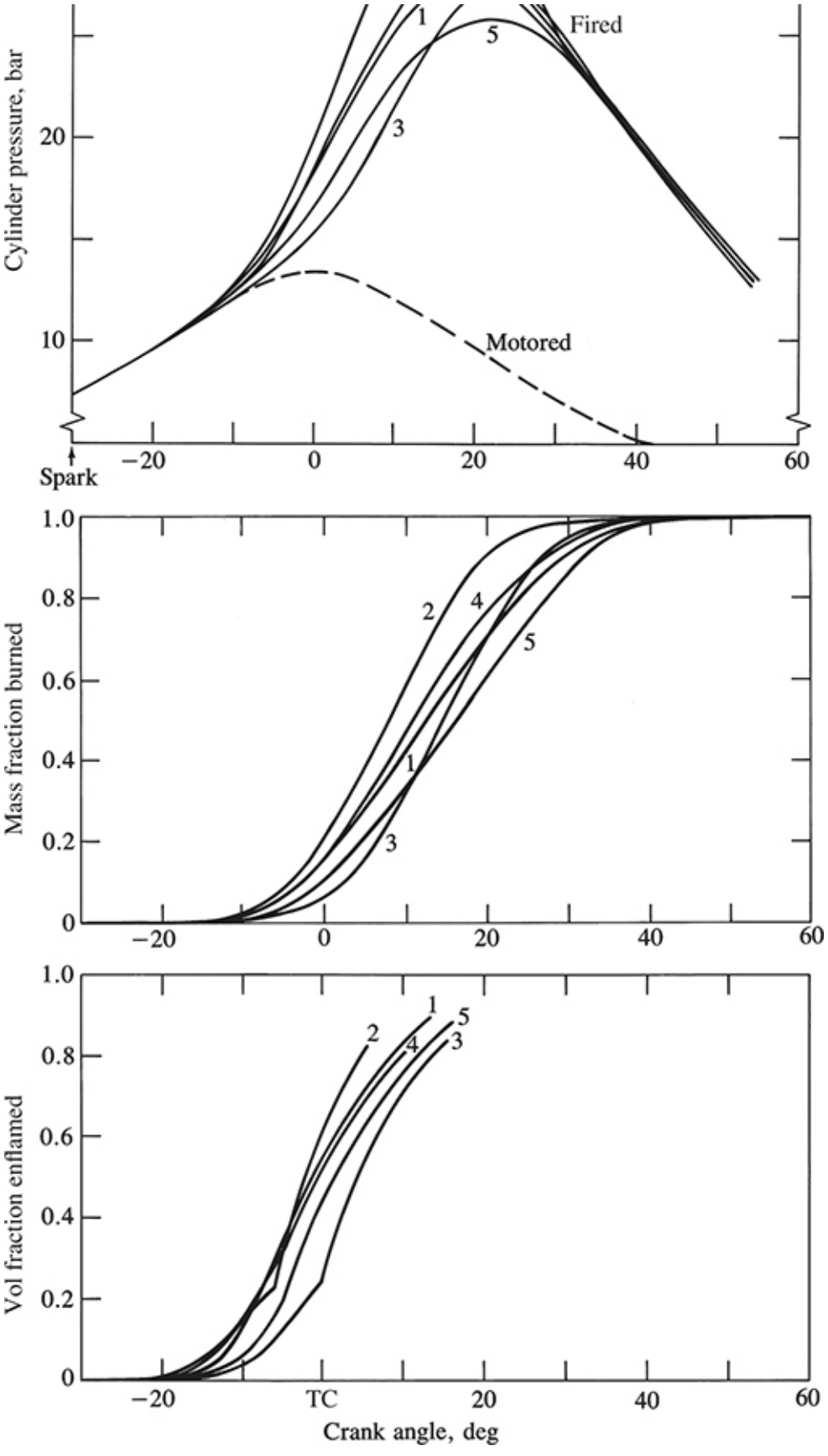

Cylinder pressure, bar
20
10
1
5
3
Fired
Motored
Spark
−20
0
20
40
60
Mass fraction burned
1.0
0.8
0.6
0.4
0.2
0
2
4
5
1
3
Vol fraction enflamed
1
2
4
5
3
TC
Crank angle, deg

Figure 9.4 Cylinder pressure, mass fraction burned, and volume fraction enflamed for five consecutive cycles in a spark-ignition engine as a function of crank angle. Ignition timing 30° BTC, wide-open throttle, 1044 rev/min, $\varphi = 0.98$. [4]

The flame development and subsequent propagation obviously vary, cycle-by-cycle, since the shape of the pressure, volume fraction enflamed, and mass fraction burned curves for each cycle differ significantly. This is because flame growth depends on local mixture motion and composition. These quantities vary in successive cycles in any given cylinder and may vary cylinder-to-cylinder. Especially significant are mixture motion and composition in the vicinity of the spark plug at the time of spark discharge since these govern the early stages of flame development. Cycle-by-cycle and cylinder-to-cylinder variations in this combustion process are important: see Sec. 9.4.1 .

Note that the volume fraction enflamed curves rise more steeply than the mass fraction burned curves. In large part, this is because the density of the unburned mixture ahead of the flame is about four times the density of the burned gases behind the flame. Also, there is some unburned mixture behind the visible front to the flame: even when the entire combustion chamber is fully enflamed, some 25% of the mass has still to burn. From this description it is plausible to divide the combustion process into four distinct phases: (1) spark ignition; (2) early flame development; (3) flame propagation; and (4) flame termination. Our understanding of each of these phases will be developed in this chapter.

The combustion event must be properly phased relative to top-center piston position to obtain the maximum power or torque. The combined duration of the flame development and propagation process is typically between 40 and 60 crank-angle degrees. Combustion starts before the end of the compression stroke, continues through the early part of the expansion stroke, and ends after the point in the cycle at which the peak cylinder pressure occurs. The pressure-versus-crank-angle curves shown in Fig. 9.5 *a* allow us to understand why engine torque (at given engine speed and intake manifold conditions) varies as spark timing is varied relative to piston position, and thus crank angle. If the start of the combustion process is progressively advanced before TC, the compression stroke work transfer

(which is *from* the piston *to* the cylinder gases) increases, as does heat transfer to the combustion chamber walls. If the end of the combustion process is progressively delayed by retarding the spark timing, the peak cylinder pressure occurs later in the expansion stroke and is reduced in magnitude. These changes reduce the expansion stroke work transfer *from* the in-cylinder gases *to* the piston. The optimum timing, which gives the maximum brake torque—called *maximum brake torque, or MBT, timing*—occurs when the magnitudes of these two opposing trends just offset each other. Timing which is advanced or retarded from this optimum gives lower torque. The optimum spark setting will depend on the rate of flame development and propagation, the length of the flame travel path across the combustion chamber, and the details of the flame termination process after it reaches the wall. These depend on engine design and operating conditions, and the properties of the fuel, air, burned gas mixture. Figure 9.5 *b* shows the effect of variations in spark timing on brake torque for a typical naturally-aspirated spark-ignition engine. MBT timing is the appropriate reference point for the actual spark timing, which should be defined relative to MBT timing (e.g., 10 degrees retard from MBT).

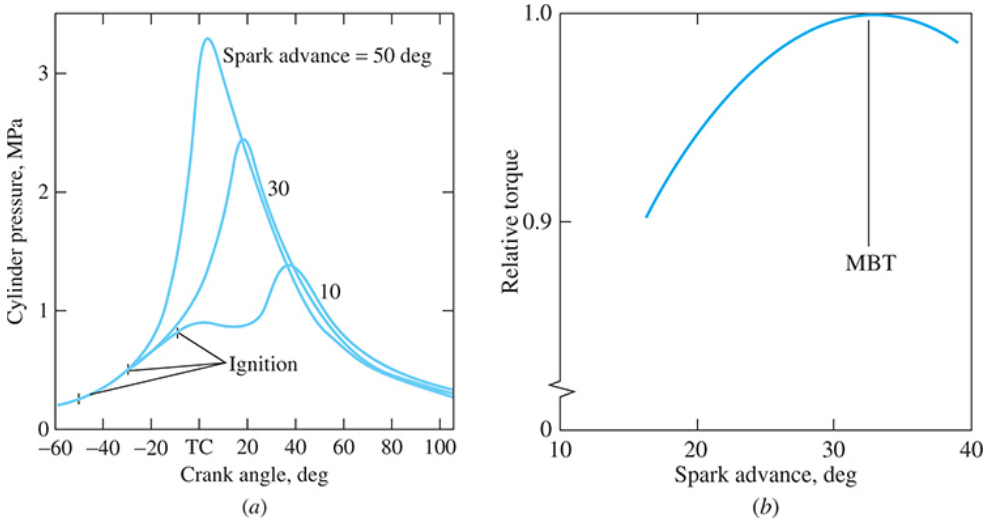


Figure 9.5 (*a*) Cylinder pressure versus crank angle for over-advanced spark timing (50°), MBT timing (30°), and retarded timing (10°). (*b*) Effect of spark advance (before TC) on brake torque at constant speed and (*A/F*), at wide-open throttle. MBT is maximum brake torque timing. [5]

Empirical rules for relating the mass burning profile and maximum cylinder pressure to crank angle at MBT timing or 50% mass burned fraction are often used. For example, with optimum spark timing: (1) the maximum pressure occurs at some 13 to 15° after TC; (2) half the charge is burned at about 8° after TC. In practice, the spark is often retarded to give a 1 or 2% reduction in brake torque from the maximum value, to lower peak cylinder pressures and burned gas temperatures. Substantial spark retard is used at times to achieve certain practical benefits (e.g., during cold engine starting to achieve fast catalyst light off, or at high load in turbocharged engines to hold peak pressure to acceptable levels).

So far we have described normal combustion in which the spark-ignited flame moves *steadily* across the combustion chamber until the charge is fully consumed. However, several factors—for example, fuel composition, certain engine design, and operating parameters—may prevent this normal combustion process from occurring. Two types of abnormal combustion have been identified: spontaneous ignition and knock.

Knock is the most important abnormal combustion phenomenon. Its name comes from the noise that results from the spontaneous autoignition of a portion of the fuel, air, residual gas mixture ahead of the advancing flame. As the flame propagates across the combustion chamber, the unburned mixture ahead of the flame—called the *end gas*—is compressed, causing its pressure, temperature, and density to increase. Some of the end-gas fuel air mixture may undergo chemical reactions prior to normal combustion. The products of these reactions may then *autoignite*: that is, spontaneously and rapidly release their chemical energy. When this happens, the end gas burns very rapidly, releasing its energy at a rate 5 to 25 times that characteristic of normal combustion. This causes high-frequency pressure oscillations inside the cylinder that produce the sharp metallic ringing noise called knock.

The presence or absence of knock reflects the outcome of a race between the advancing flame front and the autoignition chemical reactions in the unburned end gas. Knock will not occur if the flame front consumes the end gas before these reactions have time to cause the fuel-air mixture to autoignite. Knock will occur if the precombustion reactions produce autoignition before the flame front arrives. Thus, *spontaneous ignition* is an important abnormal combustion phenomenon.

Another form, *surface ignition*, is ignition of the fuel-air charge by a hot surface such as overheated valves or spark plugs, by glowing combustion-

chamber deposits, or by any other hot spot in the engine combustion chamber. This may occur before the spark plug ignites the charge (preignition) or after normal ignition (postignition). Uncontrolled combustion is most evident and its effects are most severe when it results from preignition, since this can lead to heavy knock. However, even when surface ignition occurs after the spark plug fires (postignition), the spark discharge no longer has complete control of the combustion process. Thus, surface ignition may result in knock. Knock which occurs following normal spark ignition is called *spark knock* to distinguish it from knock which has been preceded by surface ignition.

As spark-ignition engine peak in-cylinder pressures have increased, a new form of preignition and knock has been observed: a sporadic preignition process leading to very high peak gas pressures. This "super, or mega, knock" process occurs sporadically (every 30,000 or so cycles), and lasts for one or a few cycles. Initiated by local autoignition, well before the spark discharge occurs, it generates these high pressures because it initiates an overly advanced combustion process leading to very high unburned-mixture pressures and temperatures that cause knock.

These abnormal combustion phenomena are reviewed in more detail in Sec. 9.6 .

9.2 THERMODYNAMICS OF SI ENGINE COMBUSTION

9.2.1 Burned and Unburned Mixture States

Consider the schematic of the engine cylinder while combustion is in progress shown in Fig. 9.6. Work transfer occurs between the cylinder gases and the piston (*to the gas* before TC; *to the piston* after TC). Heat transfer occurs to the chamber walls, primarily from the hot burned gases. At the temperatures and pressures typical of spark-ignition engines, it is a reasonable approximation to assume that the volume of the reaction zone *where combustion is actually occurring* is a negligible fraction of the chamber volume even though the thickness of the turbulent flame may not be negligible compared with the chamber dimensions (see Sec. 9.3.2). With

normal engine operation, at any point in time or crank angle, the pressure throughout the cylinder is essentially uniform.

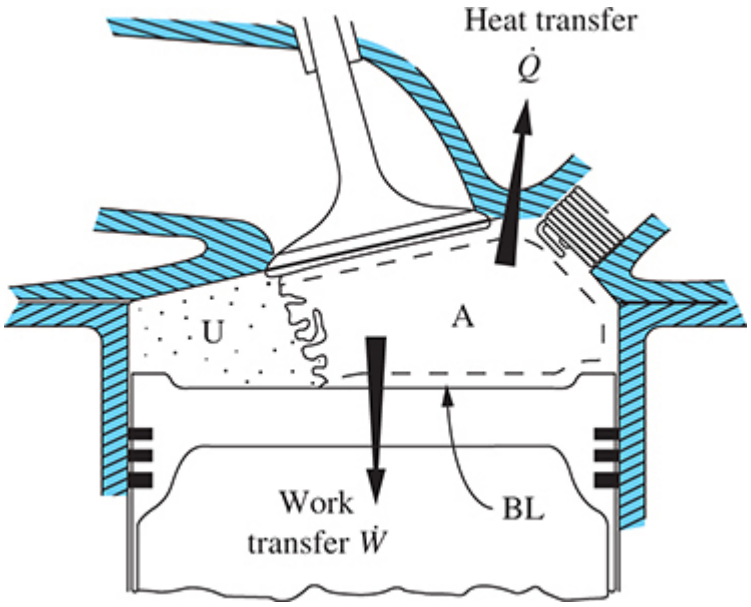


Figure 9.6 Schematic of turbulent premixed flame in the engine cylinder during combustion: unburned gas (U) to left of flame, burned gas to right. A denotes adiabatic burned-gas core, BL denotes thermal boundary layer in burned gas, $\dot{W}$ is heat-transfer rate to chamber walls.

Because combustion occurs through a flame propagation process, the changes in state and the motion of the unburned and burned gas are much more complex than the ideal cycle analysis in Chap. 5 suggests. The gas pressure, temperature, and density change as a result of changes in volume due to piston motion. Then, during combustion, the cylinder pressure increases due to the release of the fuel's chemical energy. As shown in Fig. 9.7 *a*, as each element of fuel-air mixture burns and its temperature increases, its density decreases by about a factor of four. This combustion-produced gas expansion compresses the unburned mixture ahead of the flame and displaces it toward the combustion chamber walls. The combustion-produced gas expansion also compresses those parts of the charge which have already burned, and displaces them back toward the spark plug. During the combustion process, the unburned gas elements move away from the spark plug and the burned gas elements move back toward the spark plug. Further, elements of the unburned mixture which burned at different times have

different pressures and temperatures just prior to combustion, and therefore end up at different states after combustion. The thermodynamic state and composition of the burned gas is, therefore, non-uniform. A first law analysis of the spark-ignition engine combustion process enables us to quantify these gas states.

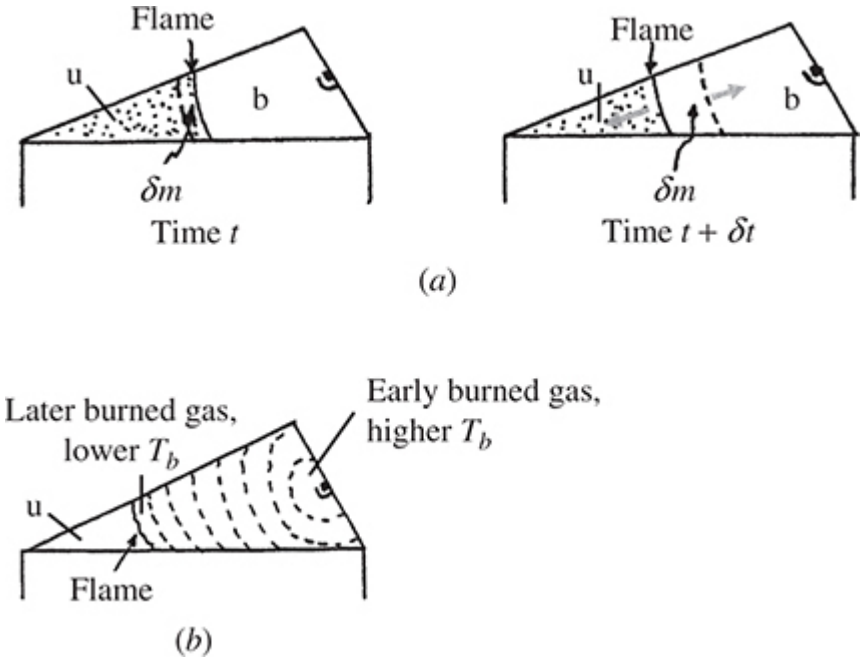


Figure 9.7 Schematic of combustion- produced pressure rise and gas motion: (*a*) burning of mass element δm causes that element to expand, compressing and displacing unburned mixture (u) ahead of the flame and burned gas (b) behind the flame; (*b*) unmixed model for burned gas temperature distribution resulting from this sequential layer-by-layer burning process.

The conditions in the burned and unburned gas are then determined by conservation of mass:

$$\frac{V}{m} = \int_0^{x_b} v_b dx + \int_{x_b}^1 v_u dx \tag{9.1}$$

and conservation of energy:

$$\frac{U_0 - W - Q}{m} = \int_0^{x_b} u_b dx + \int_{x_b}^1 u_u dx \tag{9.2}$$

where V is the cylinder volume, m is the mass of the cylinder contents, v is the specific volume, x_b is the mass fraction burned, U_0 is the internal energy of the cylinder contents at some reference point θ_0, u is the specific internal energy, W is the work done on the piston, and Q is the heat transfer to the walls. The subscripts u and b denote unburned and burned gas properties, respectively. The work and heat transfers are

$$W = \int_{V_0}^{V} p dV \quad Q = \int_{\theta_0}^{\theta} \left(\frac{\dot{Q}}{360N} \right) d\theta \tag{9.3}$$

where $\dot{Q}$ is the instantaneous heat-transfer rate to the chamber walls.

To proceed further, models for the thermodynamic properties of the burned and unburned gases are required. Several categories of models are described in Chap. 4 . Accurate calculations of the state of the cylinder gases require an equilibrium model (or good approximation to it) for the burned gas and an ideal gas mixture model (of frozen composition) for the unburned gas (see Table 4.2). However, useful illustrative results can be obtained by assuming that the burned and unburned gases are different ideal gases, each with constant specific heats, [6] that is,

$$pv_b = R_b T_b \quad u_b = c_{v,b} T_b + h_{f,b} \tag{9.4}$$

$$pv_u = R_u T_u \quad u_u = c_{v,u} T_u + h_{f,u} \tag{9.5}$$

Combining Eqs. (9.1) to (9.5) gives

$$\frac{pV}{m} = x_b R_b \bar{T}_b + (1 - x_b) R_u \bar{T}_u \tag{9.6}$$

and

$$\frac{U_0 - W - Q}{m} = x_b (c_{v,b} \bar{T}_b + h_{f,b}) + (1 - x_b)(c_{v,u} \bar{T}_u + h_{f,u}) \tag{9.7}$$

where

$$\bar{T} = \frac{1}{x_b}\int_0^{x_b} T_b dx \quad \text{and} \quad \bar{T}_u = \frac{1}{1-x_b}\int_{x_b}^{1} T_u dx$$

are the mean temperatures of the burned and unburned gases. Equations (9.6) and (9.7) may now be solved to obtain

$$x_b = \frac{pV - p_0V_0 + (\gamma_b - 1)(W+Q) + (\gamma_b - \gamma_u)mc_{v,u}(\bar{T}_u - T_0)}{m[(\gamma_b - 1)(h_{f,u} - h_{f,b}) + (\gamma_b - \gamma_u)c_{v,u}\bar{T}_u]} \tag{9.8}$$

and

$$\bar{T}_b = \frac{R_u}{R_b}\bar{T}_u + \frac{pV - mR_u\bar{T}_u}{mR_b x_b} \tag{9.9}$$

If we now assume that the unburned gas is initially uniform and undergoes isentropic compression, then

$$\frac{\bar{T}_u}{T_0} = \left(\frac{p}{p_0}\right)^{(\gamma_u - 1)/\gamma_u} \tag{9.10}$$

This equation, with Eqs. (9.8) and (9.9), enables determination of both x_b and $\bar{T}_b$ from the thermodynamic properties of the burned and unburned gases, and known values of p, V, m, and $\dot{Q}$. Alternatively, if x_b is known then p can be determined. Mass fraction burned and cylinder gas pressure are uniquely related.

While Eq. (9.9) defines a mean burned gas temperature, the burned gas is not uniform. As illustrated in Fig. 9.7, mixture which burns early in the combustion process is further compressed after combustion as the remainder of the charge is burned. Mixture, which burns late in the combustion process, is compressed prior to combustion and, therefore, ends up at a different final state. A temperature gradient exists across the burned gas with the earlier burning portions at the higher temperature. [7, 8] Two limiting models bracket what occurs in practice: (1) a *fully mixed* model, where it is assumed that each element of mixture which burns mixes instantaneously with the already

burned gases (which therefore have a uniform temperature), and (2) an *unmixed* model, where it is assumed that no mixing occurs between gas elements which burn at different times.

In the fully mixed model, the burned gas is assumed to be uniform, $T_b = \overline{T}_b$, and the equations given above fully define the state of the cylinder contents. In the unmixed model, shown in Fig. 9.7 *b*, the assumption is made that no mixing occurs between gas elements that burn at different times, and each burned gas element is therefore isentropically compressed (and then expanded) after it combusts. [a] Thus:

$$\frac{T_b(x_b', x_b)}{T_b(x_b')} = \left[\frac{p(x_b)}{p(x_b')}\right]^{(\gamma_b - 1)/\gamma_b} \tag{9.11}$$

where $T_b(x_b', x_b)$ is the temperature of the element which burned at the pressure $p(x_b')$, when the pressure is $p(x_b)$, and

$$T_b(x_b') = \frac{h_{f,u} - h_{f,b} + c_{p,u} T_u(x_b')}{c_{p,b}} \tag{9.12}$$

is the temperature resulting from isenthalpic combustion of the unburned gas at $T_u(x_b')$, $p(x_b')$. An example of the temperature distribution computed with this model is shown in Fig. 9.8. A mixture element that burns right at the start of the combustion process reaches, in the absence of mixing, a peak temperature after combustion about 400 K higher than an element that burns toward the end of the combustion process. The mean burned gas temperature is closer to the lower of these temperatures. These two models approximate, respectively, to situations where the time scale that characterizes the turbulent mixing process in the burned gases is (1) much less than the overall burning time (for the fully mixed model) or (2) much longer than the overall burning time (for the unmixed model). The real situation, based on burned gas temperature measurements, [8] lies in between.

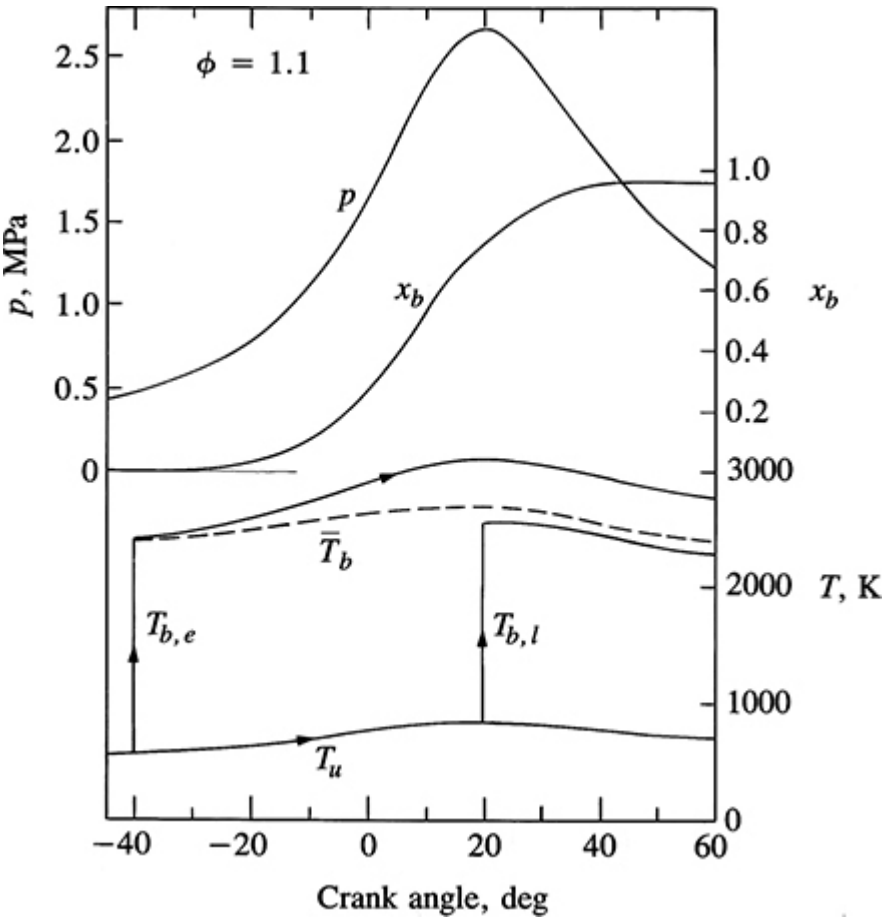


Figure 9.8 Cylinder pressure, mass fraction burned, and gas temperatures as functions of crank angle during combustion. T_u is unburned gas temperature, τ_b is burned gas temperature, the subscripts e and l denote early and late burning gas elements, and $\bar{T}_b$ is the mean burned gas temperature.[6]

As has already been mentioned, the expansion of a gas element which occurs during combustion compresses the gas ahead of the flame and moves it away from the spark plug. At the same time, previously burned gas is compressed and moved back toward the spark plug. Defining this motion in an engine requires sophisticated flow models because the combustion chamber shape is rarely symmetrical and the spark plug is often not centrally located. However, the gas motion in a spherical or cylindrical combustion bomb with central ignition, which can readily be computed, illustrates the features of the combustion-induced motion in an engine. Figure 9.9 shows calculated gas element trajectories for a stoichiometric methane-air mixture, initially at ambient conditions, as a laminar flame with a constant burning

velocity propagates outward from the center of a spherical container. Applying this gas motion model to an engine, it can be concluded that an instrument at a given location, initially views earlier burned gas (of higher temperature) and that as more of the charge burns, the instrument will view later burned gas of progressively lower temperature.

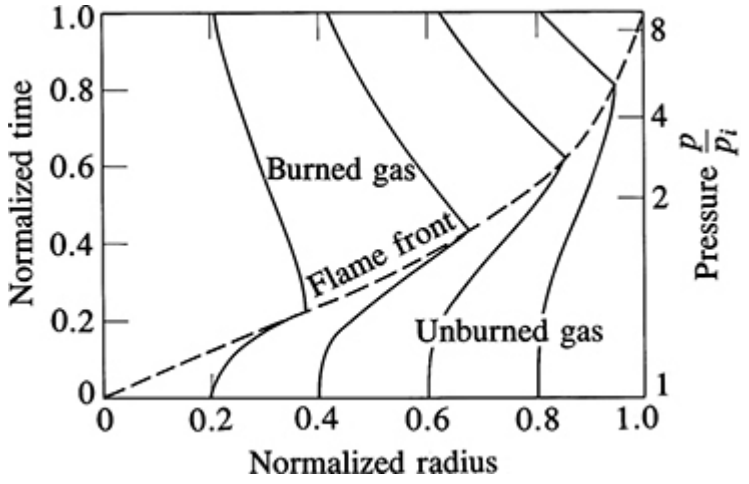


Figure 9.9 Particle trajectories in unburned and burned gas as flame propagate outward at constant velocity from the center of a spherical combustion bomb. Stoichiometric methane-air mixture initially at 1 bar and 300 K.

More accurate calculations relating the mass fraction burned, gas pressure, and gas temperature distribution are often required. The accuracy of such calculations depends on the accuracy with which the time-varying heat loss to the chamber walls can be estimated (see Sec. 12.4.3), also whether flows into and out of crevice regions are significant (see Sec. 8.6), as well as the accuracy of the models used to describe the thermodynamic properties of the gases. More accurate models for the thermodynamic properties are: an equilibrium model for the burned gas, and specific heat models, which vary with temperature for each of the components of the unburned mixture (see Sec. 4.1 and 4.7). In the absence of significant crevice effects, Eqs. (9.1) and (9.2) can be written as

$$\frac{V}{m} = \bar{v}_b x_b + \bar{v}_u (1 - x_b) \tag{9.13}$$

$$\frac{U_0 - W - Q}{m} = \bar{u}_b x_b + \bar{u}_u (1 - x_b) \tag{9.14}$$

where

$$\bar{v}_b = \frac{1}{x_b}\int_0^{x_b} v_b dx \quad \text{and} \quad \bar{v}_u = \frac{1}{1 - x_b}\int_{x_b}^{1} v_u dx$$

and similar definitions hold for $\bar{u}_b$ and $\bar{u}_u$. For a given equivalence ratio, fuel and burned gas fraction:

$$h_u = h_u(T_u) \quad h_b = h_b(T_b, p) \tag{9.15a,b}$$

$$pv_u = \left(\frac{\tilde{R}}{M_u}\right)T_u \quad pv_b = \left(\frac{\tilde{R}}{M_b}\right)T_b \tag{9.16a,b}$$

and

$$\bar{u}_u = \bar{h}_u - p\bar{v}_u \quad \bar{u}_b = \bar{h}_b - p\bar{v}_b \tag{9.17a,b}$$

To simplify the calculations it is convenient to assume that, for the burned gas, $\bar{u}_b = u_b(\bar{T}_b, p)$ and $\bar{v}_b = v_b(\bar{T}_b, p)$. This corresponds to the fully mixed assumption described above. The effect of neglecting the temperature distribution in the calculation of mass fraction burned is small. In addition, the heat losses from the unburned gas can usually be neglected. The unburned gas is then compressed isentropically. $\bar{T}_u$ is specified for some initial state of the unburned gas (where $x_b = 0$) by p_0, V_0, M_u, and the mass of charge m. Then, since for any isentropic process,

$$\left(\frac{\partial T}{\partial p}\right)_s = \frac{v - (\partial h/\partial p)_T}{(\partial h/\partial T)_p} \tag{9.18}$$

and $\bar{T}_u$ can be determined.

Equations (9.13) to (9.18) constitute a set of nine equations for the nine unknowns $\bar{v}_u$, $\bar{v}_b$, $\bar{u}_u$, $\bar{u}_b$, $\bar{h}_u$, $\bar{h}_b$, $\bar{T}_u$, $\bar{T}_b$, and x_b or p. One convenient solution method is to eliminate x_b from Eqs. (9.13) and (9.14) to obtain

$$\frac{(V/m)-\bar{v}_u}{\bar{v}_b-\bar{v}_u}-\frac{(U/m)-\bar{u}_u}{\bar{u}_b-\bar{u}_u}=f(\bar{T}_b,\bar{T}_u)=0 \tag{9.19}$$

where $U = U_0 - W - Q$. $\bar{T}_u$ can be determined from Eq. (9.18) . Equation (9.19) can then be solved using an appropriate iterative technique for $\bar{T}_b$, and x_b can be obtained from Eq. (9.13) . Some examples of mass fraction burned curves obtained from measured pressure data, with gasoline and methanol fuels, are shown in Fig. 9.10. With accurate pressure versus crank-angle records, values of final mass fraction burned should be close to but lower than unity, usually by about 5% due to incomplete combustion of the fuel within the cylinder: see Fig. 3.9 and Fig. 11.35.

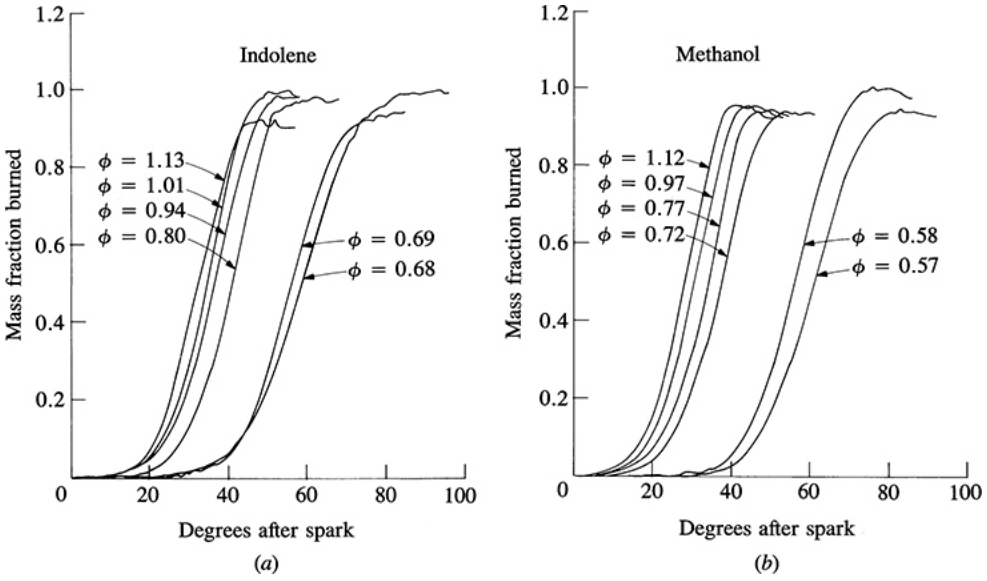


Figure 9.10 Mass fraction burned curves determined from measured cylinder pressure data using two-zone combustion model: (*a*) gasoline; (*b*) methanol. φ = fuel/air equivalence ratio.[9]

More accurate burned gas temperature calculations need to account for the presence of a thermal boundary layer (of order 1-mm thick) around the combustion chamber walls (see Sec. 12.6.5). The burned gas region in Fig. 9.6 can be divided into an adiabatic core and a boundary layer that grows in thickness with time. In the adiabatic core, in the absence of mixing between gas elements that burn at different times, burned gas is compressed and then expanded isentropically. The burned gas temperature distribution can be

calculated as follows. Given the pressure versus crank-angle data, the unburned mixture state can be determined using Eq. (9.18) above. Each small element of unburned mixture burns in an essentially constant-enthalpy constant-pressure process. So the burned state of an element of unburned charge, which burns at $p = pi$, can be obtained from the relation

$$h_b(T_{b,i}, p_i) = h_u(T_{u,i}, p_i)$$

After combustion, this element which burned at $p = p_i$ is compressed and expanded along the isentropic:

$$s_b(T_b, p) = s_b(T_{b,i}, p_i)$$

An example of the temperature distribution computed in this manner for this unmixed model in the burned gas adiabatic core is shown in Fig. 9.11. The element ignited by the spark is compressed to the highest peak temperature at P_{max}. The temperature difference across the bulk of the charge ($0.05 < x_b < 0.95$) is about 200 K.

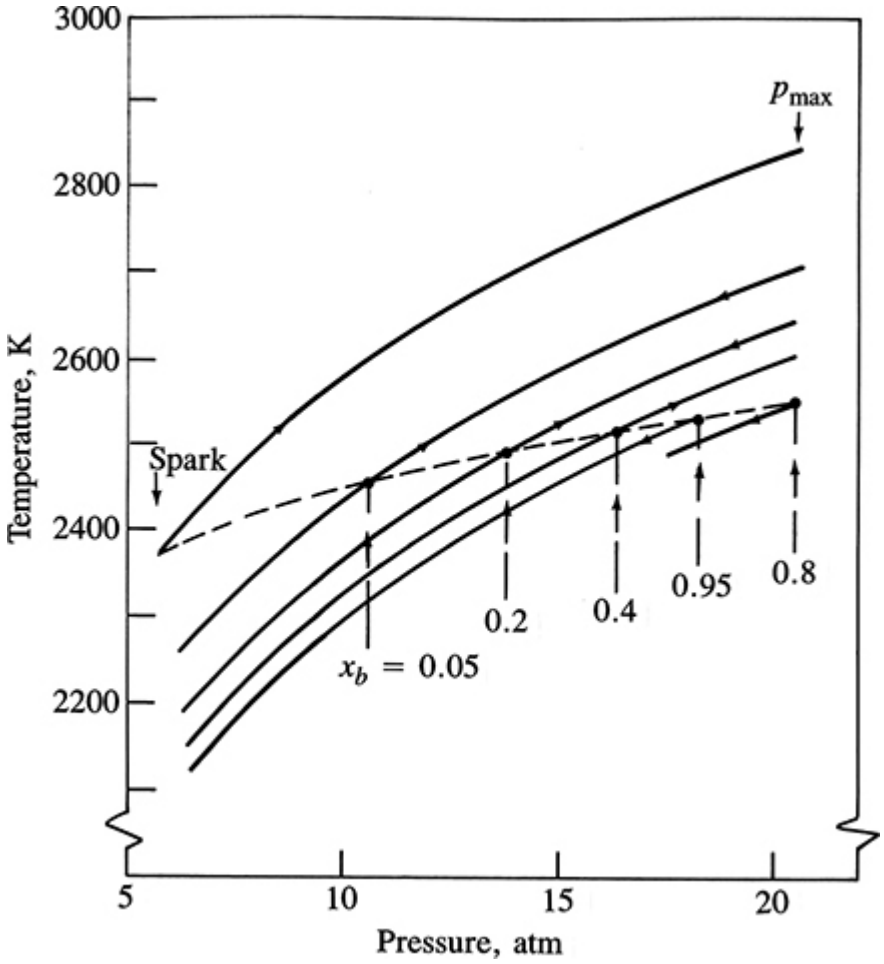


Figure 9.11 Calculated temperature distribution in the adiabatic core of the burned gas zone for the unmixed model assuming thermodynamic equilibrium. $\varphi = 1.0$. Dashed line is temperature of each element just after it burns.

9.2.2 Analysis of Cylinder Pressure Data

Cylinder pressure changes with crank angle as a result of cylinder volume change, combustion, heat transfer to the chamber walls, flow into and out of crevice regions, and leakage. The first two of these effects are the largest. The effect of volume change on the pressure can readily be accounted for. Thus, combustion rate information can be obtained from accurate pressure data provided models for the remaining phenomena can be developed at an appropriate level of approximation. The previous section has developed the fundamental basis for such calculations.

Cylinder pressure is usually measured with piezoelectric pressure transducers. This type of transducer contains a quartz crystal. One end of the crystal is exposed through a diaphragm to the cylinder pressure; as the cylinder pressure increases, the crystal is compressed and generates an electric charge which is proportional to the pressure. A charge amplifier is then used to produce an output voltage proportional to this charge. Accurate cylinder pressure versus crank-angle data can be obtained with these systems provided the following steps are carried out: (1) the correct reference pressure is used to convert the measured pressure signals to absolute pressures; (2) the pressure versus crank-angle (or volume) phasing is accurate to within about 0.2°; (3) the clearance volume is estimated with sufficient accuracy; (4) transducer temperature variations (which can change the transducer calibration factor) due to the variation in wall heat flux during the engine cycle are held to a minimum. Log p versus log V plots can be used to check the quality of cylinder pressure data. The first three of the above requirements can be validated using log p–log V diagrams for a motored engine. If the effects of thermal cycling of the pressure transducer are significant, the expansion stroke on the log p–log V plot for a firing engine shows excessive curvature. [10, 11]

Figure 9.12 shows pressure-volume data from a firing spark-ignition engine on both a linear p–V and a log p–log V diagram. On the log p–log V diagram the compression process is a straight line of slope about 1.3. The start of combustion can be identified by the departure of the curve from the straight line. The end of combustion can be located in similar fashion: the expansion stroke following combustion is essentially linear with slope 1.33. Since both the compression of the unburned mixture prior to combustion and the expansion of the burned gases following the end of combustion are close to adiabatic isentropic processes (for which pV_γ = constant; $\gamma = c_p/c_v$), the observed behavior is as expected. Extensive studies show that the compression and expansion processes are well fitted by a polytropic relation:

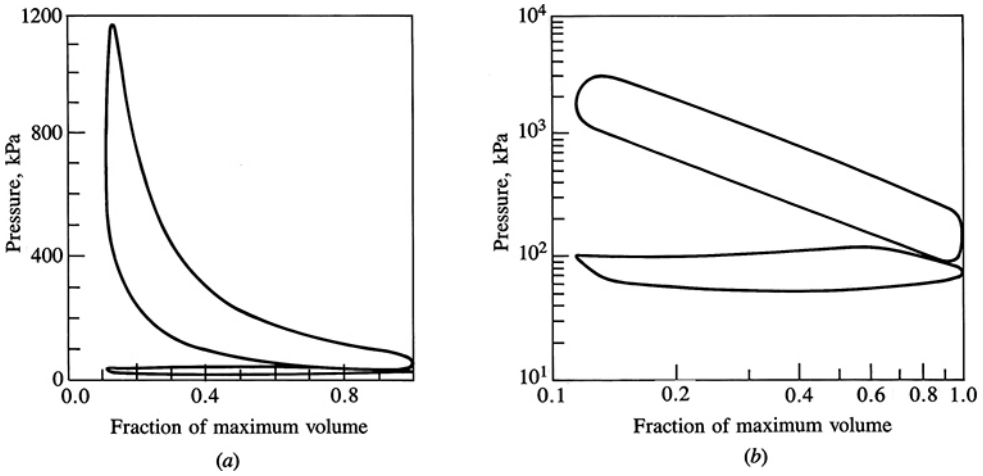


Figure 9.12 (*a*) Pressure-volume diagram; (*b*) log *p*-log (V/V_{max}) plot. 1500 rev/min, MBT spark timing, imep = 513 kPa, $\varphi = 0.8$, $r_c = 8.72$, propane fuel. [10]

$$pV^n = \text{constant} \tag{9.20}$$

The exponent *n* for the compression and expansion processes is 1.3 (± 0.05) for most fuels. It is comparable to the average value of γ_u for the unburned mixture over the compression process, but is larger than γb for the burned gas mixture during expansion due to heat loss to the combustion chamber walls (see Figs. 4.13 and 4.16).

Log *p*–log *V* plots such as Fig. 9.12 *b* approximately define the start and end of combustion, but do not provide a mass fraction burned profile. One well-established technique for estimating the mass fraction burned profile from the pressure and volume data is that developed by Rassweiler and Withrow. [2] They correlated cylinder pressure data with flame photographs, and showed how Eq. (9.20) could be used to account for the effect of cylinder volume change on the pressure during combustion. Assuming that the unburned gas filling the volume *Vu* ahead of the flame at any crank angle during combustion has been compressed polytropically by the advancing flame front, then the volume $V_{u,\,0}$ that mass occupied at time of spark is

$$V_{u,0} = V_u \left(\frac{p}{p_0} \right)^{1/n} \tag{9.21}$$

Similarly, the burned gas behind the flame filling the volume V_b would, at the end of combustion, fill a volume $V_{b,f}$ given by

$$V_{b,f} = V_b \left(\frac{p}{p_f} \right)^{1/n} \tag{9.22}$$

The mass fraction burned x_b is equal to 1 - ($V_{u,0}$/ V_0) and to $V_{b,f}$/ f, where V_0 and V_f are the total cylinder volume at time of spark (just before combustion) and at the end of combustion, respectively. Since $V = V_u + V_b$, Eqs. (9.21) and (9.22) then give:

$$x_b = \frac{p^{1/n} V - p_0^{1/n} V_0}{p_f^{1/n} V_f - p_0^{1/n} V_0} \tag{9.23}$$

This method is widely used, though it contains several approximations. Heat-transfer effects are included only to the extent that the polytropic exponent n in Eq. (9.22) differs appropriately from γ. The pressure rise due to combustion is proportional to the amount of fuel chemical energy released rather than the mass of mixture burned. Also, the polytropic exponent n is not the same before, and after combustion. Selecting an appropriate value for n (whether n is assumed to be constant or to vary through the combustion process) is one difficulty in applying this pressure data analysis procedure.

The effects of heat transfer, crevices, and leakage can be explicitly incorporated into cylinder pressure data analysis by using a “heat release” approach based on the first law of thermodynamics. A significant advantage of such an approach is that the pressure changes can be related directly to the amount of fuel chemical energy released by combustion, while retaining the simplicity of treating the combustion chamber contents as a single zone. Figure 9.13 shows the appropriate open-system boundary for the combustion chamber.[12] The first law for this open system is

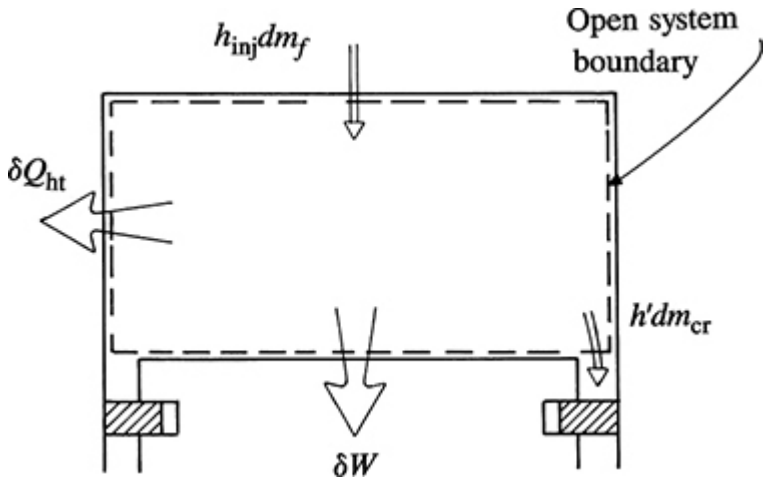


Figure 9.13 Open system boundary for engine combustion chamber for heat-release analysis.

$$\delta Q_{ch} = dU_s + \delta Q_{ht} + \delta W + \sum h_i dm_i \tag{9.24}$$

The change in sensible energy of the charge *dUs* is separated from that due to combustion-produced change in composition: the term δQ_{ch} represents the "chemical energy" released by combustion. The work δW is piston work and equal to *pdV*. δQ_{ht} is heat transfer to the chamber walls. The mass flux term represents flow across the system boundary. It includes direct fuel injection and the flow into and out of crevice regions (see Sec. 8.6).

The accuracy with which this energy balance can be made depends on how adequately each term in the above equation can be quantified. Assuming that *Us* is given by *mu*(*T*), where *T* is the mean charge temperature and *m* is the mass within the system boundary, then

$$dU_s = mc_v(T)dT + u(T)dm$$

Note that this mean temperature determined from the ideal gas law is close to the mass-averaged cylinder temperature during combustion since the molecular weights of the burned and unburned gases are essentially the same. Crevice effects can usually be modeled adequately by flow into and out of a single crevice volume at the cylinder pressure, with the gas in the crevice at a substantially lower temperature. Leakage to the crankcase (blowby) can

usually be neglected. Then Eq. (9.24) , on substituting for dU_s and dmi (= $dm_{cr} = -dm$), becomes

$$\delta Q_{ch} = mc_v dT + (h' - u)dm_{cr} + pdV + \delta Q_{ht} \tag{9.25}$$

where

$dm_{cr} > 0$ when flow is out of the cylinder into the crevice

$dm_{cr} < 0$ when flow is from the crevice to the cylinder, and

h' is evaluated at cylinder conditions when $dm_{cr} > 0$ and at crevice conditions when $dm_{cr} < 0$.

Use of the ideal gas law (neglecting the change in gas constant R) with Eq. (9.25) then gives

$$\delta Q_{ch} = \left(\frac{c_v}{R}\right)Vdp + \left(\frac{c_v}{R}\right)pdV + (h' - u + c_v T)dm_{cr} + \delta Q_{ht} \tag{9.26}$$

This equation can be used in several ways. When the heat or energy release term, δQ_{ch}, is combined with the heat-transfer and crevice terms, the combination is termed *net heat release*—the combustion energy release less heat lost to the walls. It is equal to the first two terms on the right-hand side of Eq. (9.26) which, together, represent the sensible energy change and work transfer to the piston. While heat losses during combustion are a small fraction of the fuel energy (10 to 15%), the distributions of heat release and heat transfer with crank angle are different; heat transfer becomes more important as the combustion process ends and average gas temperatures peak. The net heat-release profile obtained from integrating the first two terms on the right-hand side of Eq. (9.26) , normalized to give unity at its maximum value, is often interpreted as the burned mass fraction (or, more correctly, the energy-release fraction) versus crank-angle profile.

Use of Eq. (9.26) requires a value for c_v/R [= $1/(\gamma - 1)$]. The ratio of specific heats γ for both unburned and burned gases decreases with increasing temperature and varies with composition (see Figs. 4.13, 4.16, and 4.18). As the mean charge temperature increases during compression and combustion and then decreases during expansion, γ should vary. An approximate approach, modeling $\gamma(T)$ with a linear function of temperature

fitted to the appropriate curves in Figs. 4.13, 4.16, and 4.18 and with γ constant during combustion, has been shown to give adequate results. [13, 14]

The convective heat-transfer rate to the combustion chamber walls can be calculated from the relation

$$\frac{dQ_{ht}}{dt} = Ah_c(T - T_w)$$

where A is the chamber surface area, T is the mean gas temperature, T_w is the mean wall temperature, and h_c is the heat-transfer coefficient (averaged over the chamber surface area). h_c can be estimated from engine heat-transfer correlations (see Sec. 12.4.3). Since crevice effects are usually small, a sufficiently accurate model for their overall effect is to consider a single aggregate crevice volume where the gas is at the same pressure as the combustion chamber, but at a different temperature, T_w. Since these crevice regions are narrow, an appropriate assumption is that the crevice gas is at the wall temperature. Inserting this crevice model into Eq. (9.26) , with $\gamma(T) = a + bT$, gives the *chemical energy-* or *gross heat-release rate*:

$$\frac{dQ_{ch}}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} + V_{cr}\left[\frac{T'}{T_w} + \frac{T}{T_w(\gamma - 1)} + \frac{1}{bT_w} \ln\left(\frac{\gamma - 1}{\gamma' - 1}\right)\right]\frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta} \tag{9.27}$$

An example of the use of Eq. (9.27) to analyze an experimental pressure versus crank-angle curve for a spark-ignition engine is shown in Fig. 9.14. The integrated heat release is plotted against crank angle. The lowest curve shown is the net heat release. The addition of heat transfer and crevice models gives the chemical energy release. The curve at the top of the figure is the mass of fuel within the combustion chamber times its lower heating value. It decreases slightly as p_{max} is approached due to flow into the crevices. The difference between the final value of Q_{ch} and ($m_f Q_{LHV}$) should equal the combustion inefficiency (which is a few percent of mfQ_{LHV}). The combustion inefficiency can be determined from the exhaust gas composition (see Sec. 4.9.4). Inaccuracies in the cylinder pressure data and the heat-release calculation will also contribute to this difference. An important advantage of a heat-release analysis that relates the pressure

changes to the amount of fuel chemical energy within the cylinder is that this error can be determined. In the example in Fig. 9.14, the measured combustion inefficiency was close to the amount shown.

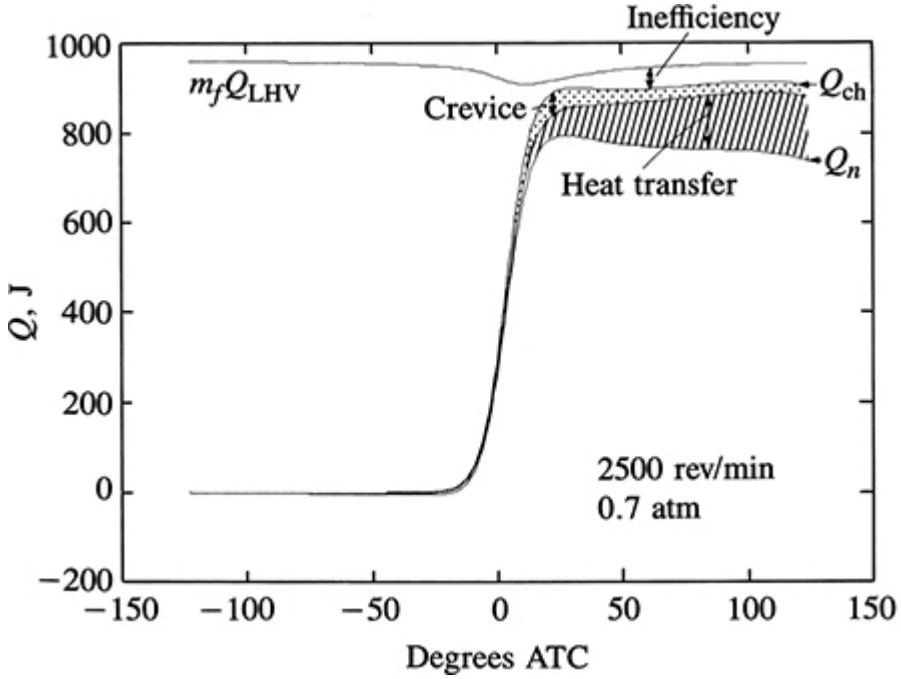


Figure 9.14 Results of heat-release analysis showing the effects of heat transfer, crevices, and combustion inefficiency.[12]

Two-zone models (one zone representing the unburned mixture ahead of the flame and one the burned mixture behind the flame) are also used to calculate the mass fraction burned profile from measured cylinder pressure data. Figure 9.10 shows results from such an analysis, using the methodology described in Sec. 9.2.1 .[9] The advantage of a two-zone analysis is that the thermodynamic properties of the cylinder contents can be quantified more accurately. The disadvantages are that the unburned and the burned zone heat-transfer areas must both now be estimated, and a model for the composition of the gas flowing into the crevice region must be developed. Due to this complexity, crevice models are usually omitted despite the fact that their impact can be significant.

9.2.3 Combustion Process Characterization

The mass fraction burned profiles as a function of crank angle in each individual cycle shown in Fig. 9.4 and the chemical energy- or gross heat-release curve in Fig. 9.14 have a characteristic S-shape. The rate at which fuel-air mixture burns increases from a low value immediately following the spark discharge, to a maximum about halfway through the burning process, and then decreases to close to zero as the combustion process ends. It proves convenient to use these mass fraction burned or energy-release fraction curves to characterize different stages of the spark-ignition engine combustion process by their duration in crank-angle degrees, thereby defining the fraction of the engine cycle that they occupy. The flame development process, from the spark discharge which initiates the combustion process to the point where a small but measurable fraction of the charge has burned, is one such stage. It is influenced primarily by the mixture state, composition, and gas motion in the vicinity of the spark plug (see Sec. 9.3). The stage during which the major portion of the charge burns as the flame propagates to the chamber walls comes next. This stage is obviously influenced by conditions throughout the combustion chamber. The final stage, where the remainder of the charge burns to completion, cannot as easily be quantified because energy-release rates are comparable to other energy-transfer processes that are occurring.

The following definitions are commonly used to characterize these energy-release aspects of combustion:

Flame-development angle $\Delta\theta_d$. The crank-angle interval between the spark discharge and the time when a small but significant fraction of the cylinder mass has burned or fuel chemical energy has been released. Usually this fraction is 10%, though other fractions such as 1% and 5% have been used. [b]

Rapid-burning angle $\Delta\theta_b$ The crank-angle interval required to burn the bulk of the charge. It is defined as the interval between the end of the flame-development stage (usually mass fraction burned or energy-release fraction of 10%) and the end of the flame-propagation process (usually mass fraction burned or energy-release fraction of 90%). [c]

Overall burning angle $\Delta\theta_o$. The duration of the overall burning process. It is the sum of and $\Delta\,\theta_d$ and $\Delta\,\theta_d$.

Figure 9.15 illustrates these definitions on a mass fraction burned, or fraction of fuel energy released, versus crank-angle plot. While the selection of the 10% and 90% points is arbitrary, such a choice avoids the difficulties involved in determining accurately the shape of the curve at the start and end of combustion. These angles can be converted to times (in seconds) by dividing by $6N$ (with N in revolutions per minute). (Examples of $\Delta\theta_{0\text{-}10\%}$ and $\Delta\theta_{10\text{-}90\%}$ can be found in Refs. 15 to 18.)

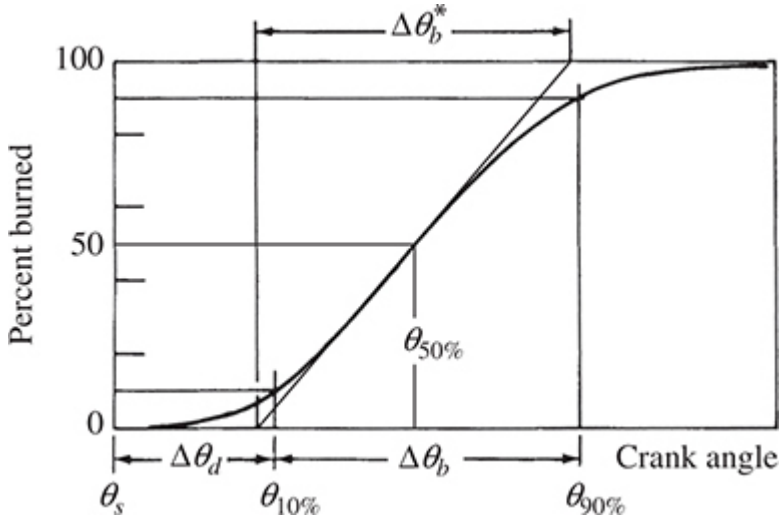


Figure 9.15 Definition of flame-development angle, $\Delta\theta_d$ and rapid-burning angle, $\Delta\theta_b$(or $\Delta\theta_b^*$) on mass fraction burned versus crank-angle curve.

The flame development angle $\Delta\theta_d$ can be estimated from the following scaling relationship: [16]

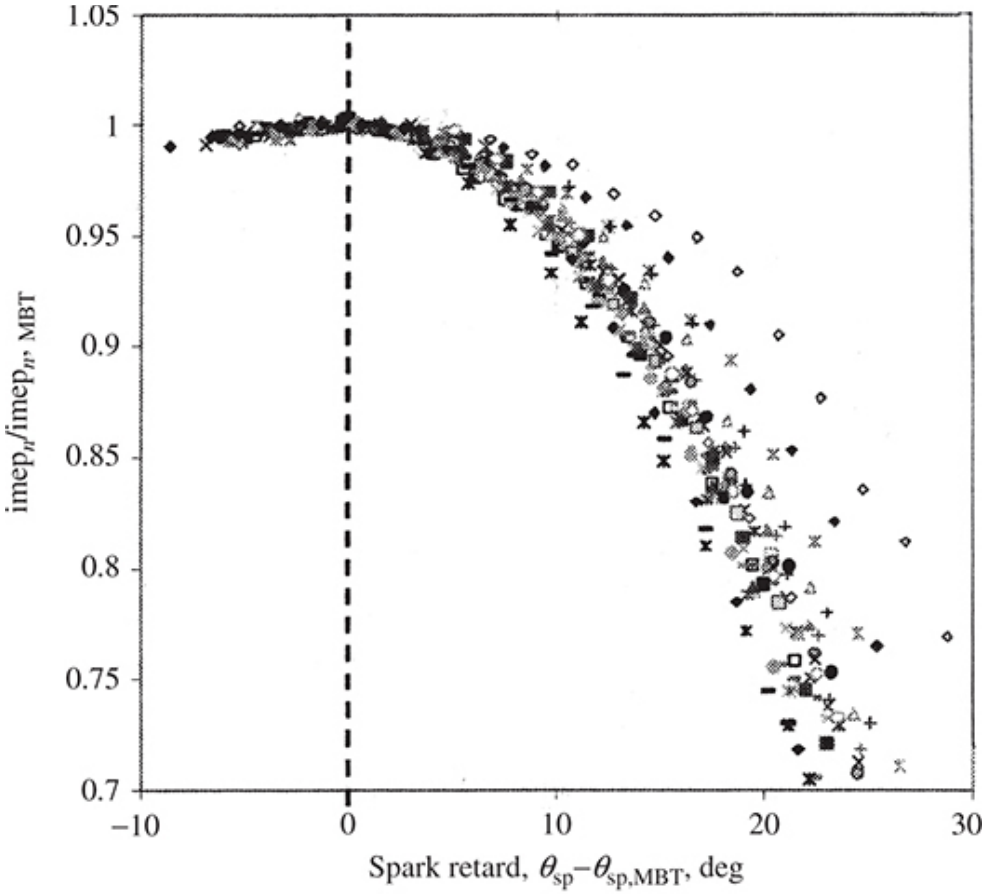
$$\frac{\Delta\theta_d}{\Delta\theta_{d,\text{ref}}} = \left(\frac{N}{N_{\text{ref}}}\right)^{1/3}\left(\frac{S_{L,\text{ref}}}{S_L}\right)^{2/3} \tag{9.28}$$

where N is engine speed, and S_L is the laminar flame speed (see Sec. 9.3.3) and includes the effects of relative air/fuel ratio, burned gas fraction, and mixture temperature and pressure.

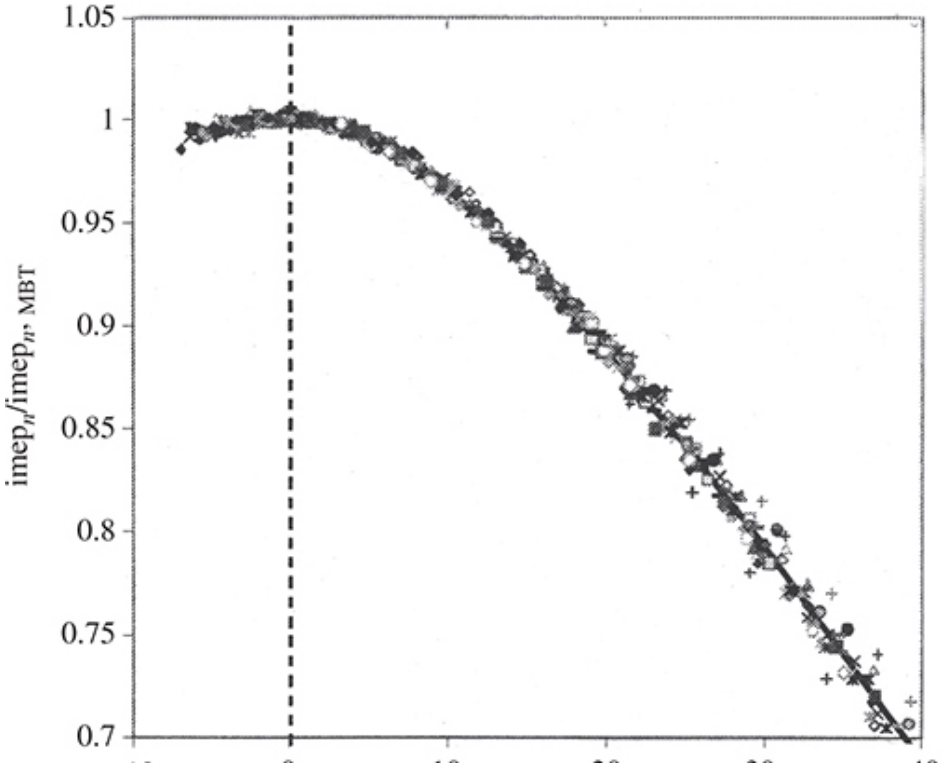
When mass fraction burned profiles are available (and as indicated above, they can readily be obtained from cylinder pressure versus crank-angle data), then the crank angle at which 50% of the in-cylinder mass has

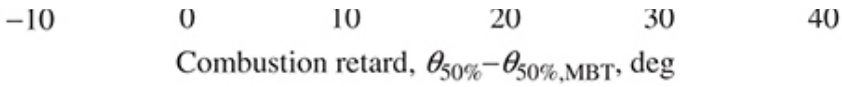
burned proves to be a useful combustion phasing parameter. Figure 9.15 identifies this 50% burned angle, $\theta_{50\%}$: it is used to characterize the *phasing* of the combustion process and determine a *combustion retard* parameter, ($\theta_{50\%}$–$\theta_{50\%\text{MBT}}$). It represents the extent (in crank-angle degrees) that the center of the combustion process has been shifted from its location at maximum torque: for a wide range of engines and their operating conditions, $\theta_{50\%}$ at MBT timing occurs at 5 to 7° ATC.

The ability of this combustion retard parameter to correlate the effects of retarding the combustion process (through spark retard from MBT timing) is illustrated in Fig. 9.16. An extensive set of mass fraction burned profiles for many different fuels, relative air/fuel ratios, compression ratios, intake pressures (different engine loads), and spark timings for a naturally-aspirated SI engine are shown. [15] Figure 9.16*a* shows net imep (normalized by the net imep at MBT timing) as a function of spark retard from MBT spark: Figure 9.16*b* shows the same normalized net imep data as a function of combustion retard, ($\theta_{50\%}$ - $\theta_{50\%\text{MBT}}$). The correlation of net imep with spark retard is good, but the few data sets on the upper right of the band of data, where the relative air/fuel ratio is very lean (λ = 1.6) and combustion is much slower, do not fit as well. In contrast, the same data with net imep plotted against combustion retard in Fig. 9.16 *b* show an extremely good correlation. The relationship between relative net imep and the extent to which the center of the combustion process is retarded from its optimum crank-angle location appears to be a universal one for naturally-aspirated SI engines. The correlation is well fitted by the equation


1.05
1
0.95
0.9
0.85
0.8
0.75
0.7
imep$_n$/imep$_{n,\,MBT}$
−10
0
10
20
30
Spark retard, $\theta_{sp}-\theta_{sp,MBT}$, deg

(*a*)


1.05
1
0.95
0.9
0.85
0.8
0.75
0.7
imep$_n$/imep$_{n,\,MBT}$



(*b*)

Figure 9.16 Net imep relative to net imep_{MBT} as a function of combustion retard from the optimum (MBT) phasing: (*a*) as a function of spark timing relative to MBT spark timing; (*b*) as a function of $\theta_{50\%}$ relative to $\theta_{50\%MBT}$.[15]

$$\frac{\text{imep}_n}{\text{imep}_{n,MBT}} = 1 - 0.168\left\{\left[1 + 4.44\times10^{-3}(\theta_{50\%} - \theta_{50\%MBT})^2\right]^{0.5} - 1\right\} \tag{9.29}$$

Here, $\theta_{50\%}$ is the location of the 50% mass fraction burned point in degrees ATC and $\theta_{50\%MBT}$ is the same parameter at MBT spark timing. Note that 5 crank-angle (CA) degrees of retard results in a 1% loss of imep *n*, and a 10-degrees CA retard results in a 3% imep *n* loss. Both of these losses are small. Equivalent spark retards show slightly larger net imep reductions. Net imep is the appropriate "load" parameter to correlate, since combustion phasing affects the pressure levels in the cylinder and, thus, the indicated work.

In turbocharged SI engines at higher loads where the compressor is supplying a significantly boosted airflow, the situation is more complex. As spark (and combustion) is retarded, the exhaust gas temperature and enthalpy increase. The turbocharger turbine is thus driven more strongly by the exhaust flow, and the turbocharger speed and boost pressure increase. This increases the net imep and torque thus partly offsetting the spark-retard-induced loss of net imep. For a turbocharged SI engine operating with significant boost, the corresponding curves to those of a naturally-aspirated SI engine in Fig. 9.16 are flatter. For example, at 1.4 bar (absolute) boost and 10 degrees spark retard, the drop in net imep is only 1%. At lighter loads, where the turbocharger has little effect on the engine's operation, spark retard effects are similar to naturally-aspirated engine behavior.

A functional form often used to represent the mass fraction burned versus crank-angle curve $x_b(\theta)$ is the Wiebe function:

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{m+1}\right] \tag{9.30}$$

where θ is the crank angle, θ_0 is the start of combustion, $\Delta\theta$ is the total combustion duration ($x_b = 0$ to $x_b = 1$), and a and m are adjustable parameters. Varying a and m changes the shape of the curve significantly. Actual mass fraction burned curves have been fitted with $a = 5$ and $m = 2$.

An example of how these burn angles can be used to characterize SI engine combustion is shown in Fig. 9.17. The figure shows the 0 to 90% burn time ($\theta_{0–10\%} + \theta_{10–90\%}$) for a 5.4-liter three-valve per cylinder V-8 SI engine with several features designed to improve the combustion characteristics. These features were: (1) a charge motion control valve (CMCV), a specially shaped "throttle" plate in each intake port which, when used at low engine speeds and light loads, provides a high level of in-cylinder charge motion to enhance the burn rate; (2) variable cam timing (VCT) which retards both the intake and exhaust valve events equally at part load to reduce the pumping losses, increase expansion stroke work, and delay the valve overlap (see Sec. 13.6); and (3) a centrally located spark plug to increase knock resistance so a high compression ratio (9.8) could be used.[18]

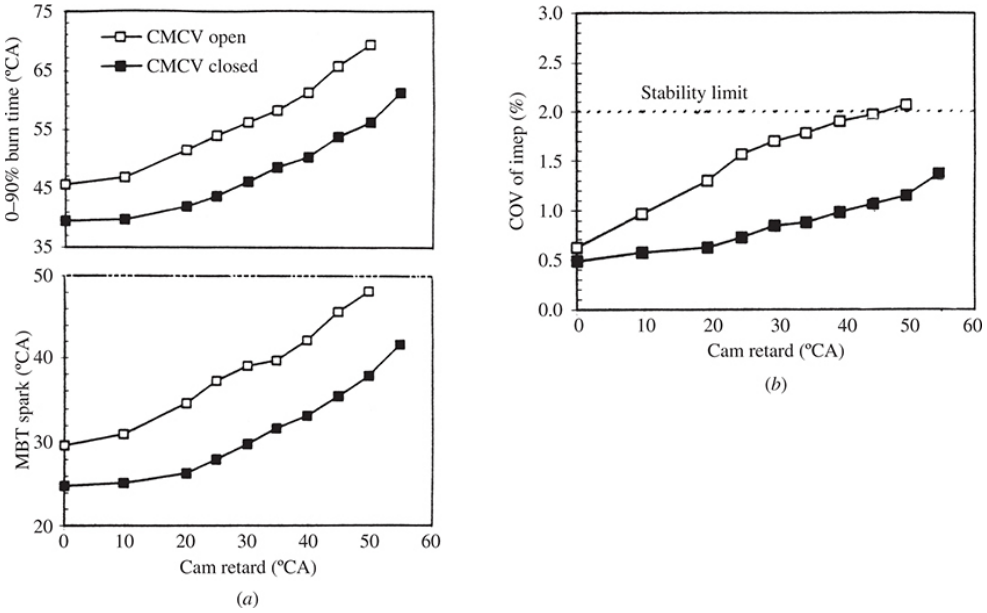


Figure 9.17 (*a*) Burn angle, spark to 90% burned, $\Delta\theta_{0-90}$, and MBT spark timing, and (*b*) coefficient of variation (%) of net imep, as a function of cam retard for a 5.4-liter three-valve per V-8 SI engine with several features designed to improve the combustion characteristics. Engine operating at 1500 rev/min and 2.62 bar bmep with a stoichiometric mixture. [18] Increasing cam retard increases the residual gas fraction in the unburned mixture. CMCV, charge motion control valve.

The data (Fig. 9.17*a*) show that, as the "dual equal" VCT increasingly retards both valve opening and closing angles, the spark timing must be advanced significantly as the combustion duration ($\Delta\theta_{0-90\%}$) steadily lengthens due to the increasing levels of mixture dilution with residual and exhaust gases, and due to the reduced compression temperature at spark due to late intake valve closing. The CMCV closing increases the burn rate significantly. Typically, $\Delta\theta_{0-10\%}$ and $\Delta\theta_{10-90\%}$ are comparable in magnitude so are some half of the $\Delta\theta_{0-90\%}$. Figure 9.17 *b* shows the impact of cam retard on cycle-by-cycle combustion variability. The coefficient of variation (COV) [d] of the cycle-by-cycle net imep (determined from analyzing individual cycle cylinder pressure data) needs to be less than about 2% for acceptably smooth engine operation at lighter loads (see Sec. 9.4.1). COV of net imep, combustion duration, and spark advance are related. Note that a spark advance of more than about 40° BTC at lightly loaded engine conditions usually results in substantial combustion variability. [15]

9.3 FLAME STRUCTURE AND SPEED

9.3.1 Overall Observations

The combustion process in the spark-ignition engine takes place in a turbulent flow field. This flow field is produced by the high shear flows set up during the intake process and modified during compression, as described in Chap. 8 . The importance of the turbulence to the engine combustion process was recognized long ago through experiments where the intake event,

and the turbulence it generates, was eliminated by closing the valves and delaying ignition to subsequent cycles: the rate of flame propagation decreased substantially. Understanding the structure of this engine flame as it develops from the spark discharge and the speed at which it propagates across the combustion chamber, and how that structure and speed depend on charge motion, turbulence levels, charge composition, and chamber geometry, are critical to engine optimization. This section reviews experimental evidence that describes the overall features of the flame development and propagation processes.

Direct flame photographs such as those in Fig. 9.2 indicate the location and shape of the actual flame reaction zone which radiates in the blue region of the visible spectrum. An irregular front is apparent. Further insight into the structure of the flame can be obtained from photographs taken with techniques that are sensitive to density changes in the flow field, such as schlieren and shadowgraph. With these techniques, a parallel light beam is passed through the combustion chamber. Portions of the beam, which pass through regions where density gradients normal to the beam exist, are deflected due to the refractive index gradients that result from the density gradients. In the schlieren technique, the beam is focused on a knife edge; the deflected parts of the beam are displaced relative to the knife edge and produce lighter or darker regions when subsequently refocused to produce an image. With the shadowgraph technique, the parallel beam emerging from the combustion chamber is observed directly; deflected parts of the beam produce lighter and darker regions in the image. With these techniques, details of flame structure can be discerned.

Figure 9.18 shows a set of photographs from one engine cycle, from a high-speed schlieren movie taken in a special visualization spark-ignition engine. Also shown are the cylinder pressure versus crank-angle data, and the mass fraction burned profile calculated from the pressure data using the method of Rassweiler and Withrow [2] (see Sec. 9.2.2). This engine had a square-cross-section cylinder with two quartz walls to permit easy optical access, but otherwise operated normally. [19] Visualization of the flame is especially important during the early stages of flame development when the pressure rise due to combustion is too small to be detected.

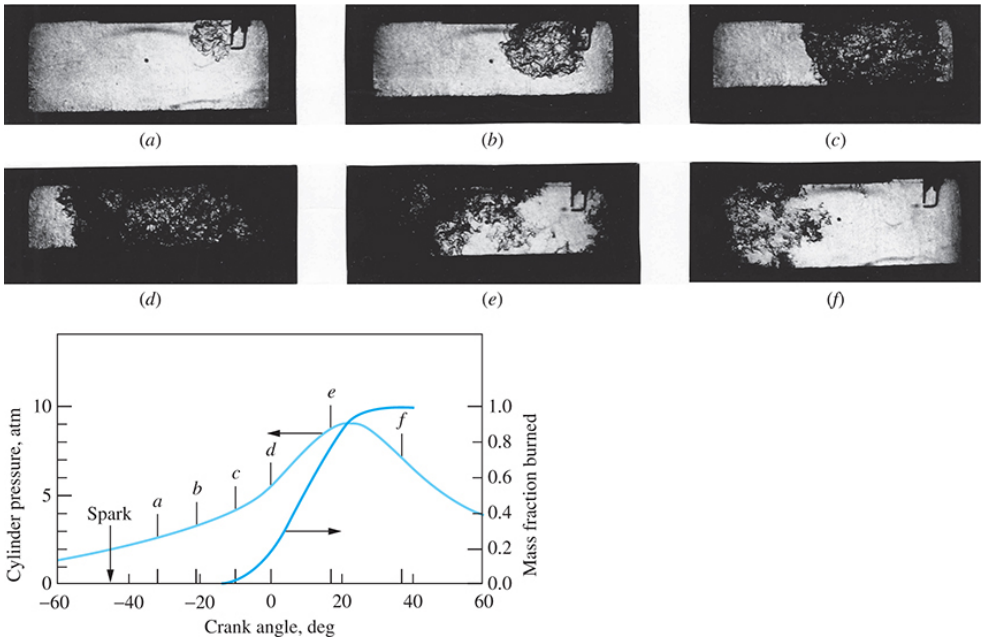


Figure 9.18 Sequence of movie frames from one engine cycle in a square-cross-section single-cylinder engine with two glass walls, and corresponding pressure and mass fraction burned curves. 1400 rev/min, 0.5 atm inlet pressure.

These photographs show how the flame, roughly spherical in shape, grows steadily from the time of spark discharge. The effect of turbulence is already visible in the convoluted flame surface in Fig. 9.18*a*. The volume enflamed behind the front continues to grow in a roughly spherical manner except where the flame reaches the chamber walls, as seen in Figs. 9.18*b* and *c*. The mass fraction burned and the associated pressure rise due to combustion become significant by the time the flame front has traversed two-thirds to three-quarters of the cylinder bore. (Note that the fraction of the cylinder filled with enflamed charge is less than is suggested by the photos because the front of the flame is approximately spherical and the cylinder has a square cross section.) Maximum cylinder pressure occurs close to the time the flame front makes contact with the far wall, as seen in Fig. 9.18 *e*. Finally, the unburned mixture (in the corners of the square cross-section chamber) ahead of, and within the front, burns out and the density gradients associated with the flame reaction zone disappear, clearing the field of view as shown in Figs. 9.18*e* and *f*.

A useful relationship between the mass fraction burned ($x_b = m_b/ m$), and the volume fraction occupied by the burned gas ($y_b = V_b/ V$), can be obtained from the identities

$$m = m_u + m_b \text{ and } V = V_u + V_b$$

along with the ideal gas law:

$$x_b = \left[1 + \frac{\rho_u}{\rho_b}\left(\frac{1}{y_b} - 1\right)\right]^{-1} \tag{9.31}$$

While the density ratio (ρ_u /ρ_b) does depend on the equivalence ratio, burned gas fraction in the unburned mixture, gas temperature, and pressure, its value is close to 4 for most spark-ignition engine operating conditions. Thus, the plot of x_b against y_b has a universal form, as shown in Fig. 9.19. This curve is an important aid in interpreting flame geometry information.

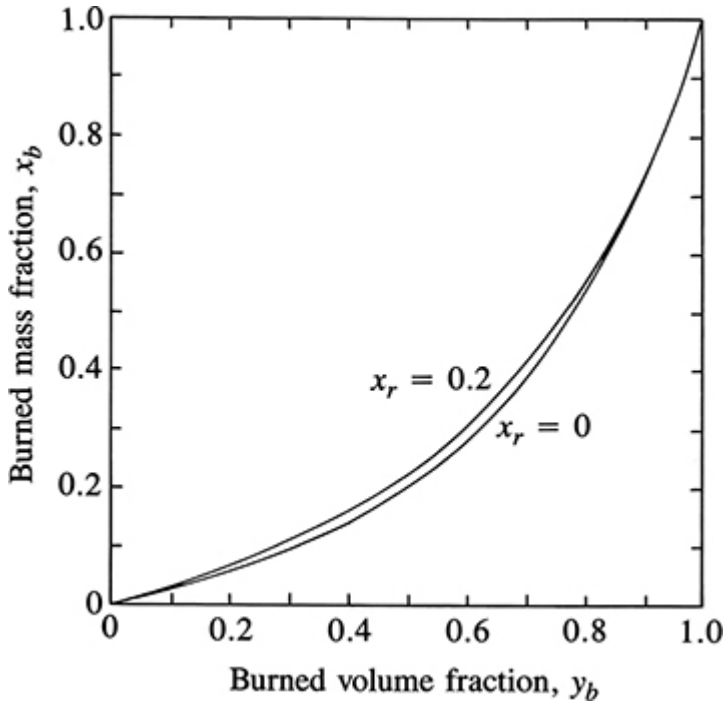


Figure 9.19 Relation between mass fraction burned x_b and volume fraction burned y_b: x_r is residual mass fraction.[4]

These overall features of the developing and propagating flame are common to almost all engine geometries and operating conditions. Figure 9.20 shows shadowgraph photographs of the flame at fixed crank-angle intervals after ignition, taken through a transparent cylinder head with different geometric and flow configurations.[20] The approximately spherical development of the flame from the vicinity of the spark plug, except where it intercepts the chamber walls, is evident for side and center ignition with one plug, and for ignition with two plugs in the absence of any intake generated swirl. With normal levels of swirl, the flame center is convected with the swirling flow, but the flame front as it grows is still approximately spherical in shape. Only with unusually high levels of swirl and aerodynamic stabilization of the flame at the spark plug location does the flame become stretched out and distorted by the flow in a major way.

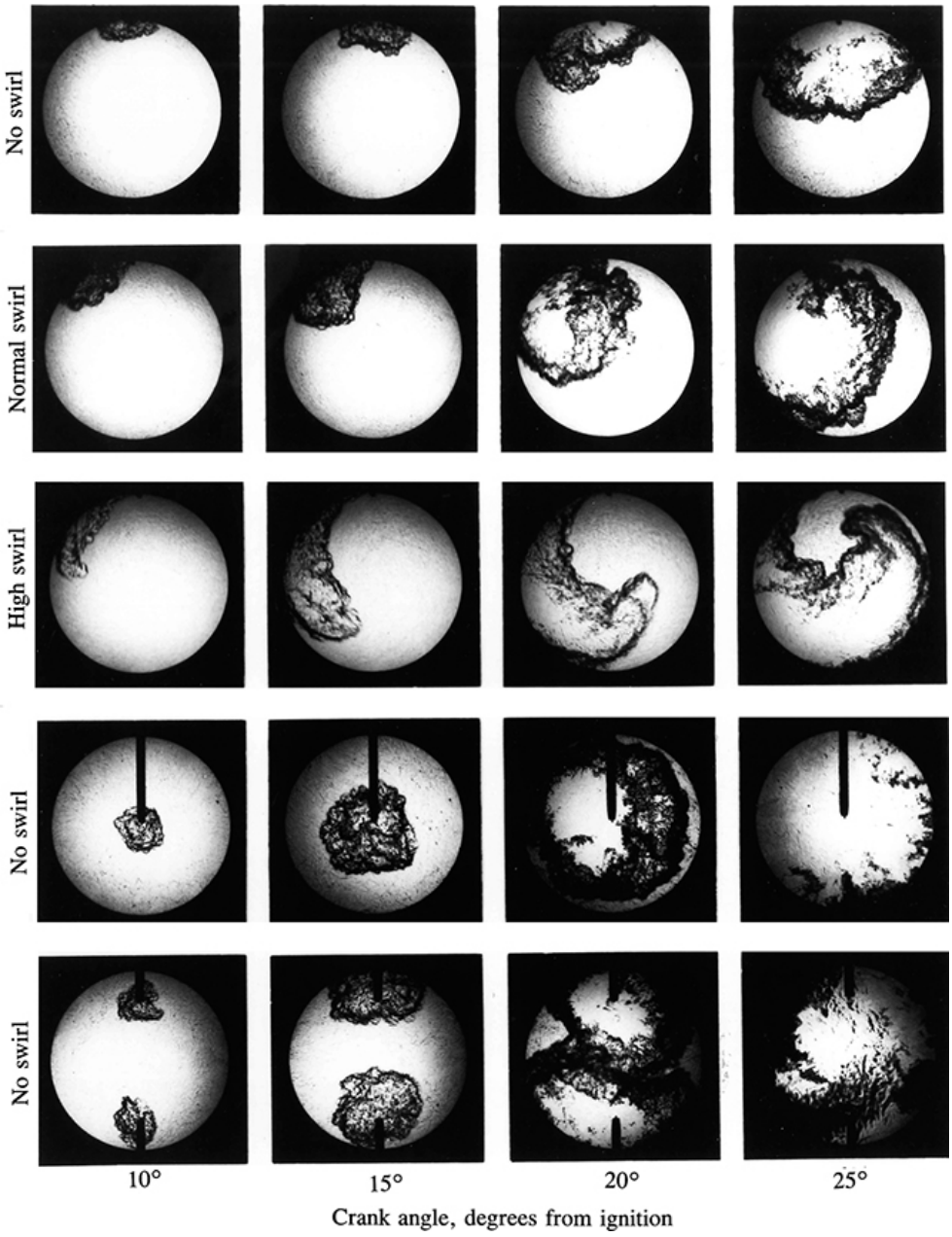


Figure 9.20 Laser shadowgraph photographs of engine combustion process taken in single-cylinder engine with transparent cylinder head. From top to bottom: side plug without swirl; side plug with normal swirl; side plug with high swirl; central plug without swirl; two plugs without swirl. [20]

At any given flame radius, the geometry of the combustion chamber surfaces and the spark plug location govern the flame front surface area—the area of that portion of the approximately spherical surface corresponding to

the leading edge of the flame, contained by the piston, cylinder head, and cylinder wall. The larger this surface area, the greater the mass of fresh charge that can cross this surface and enter the flame zone. The photos in Fig. 9.20 illustrate the importance of flame area. The center plug location gives approximately twice the flame area of the side plug geometry at a given flame radius, and thus burns about twice as fast (the fraction of the cylinder volume enflamed is about twice the size, at a fixed crank-angle interval after spark). The arrangement with two spark plugs on opposite sides of the chamber is not significantly different in enflamed volume and flame front area from the single center plug until the flame fronts are intersected by the cylinder wall.

Mixture burning rate is strongly influenced by engine speed. It is well established that the duration of combustion *in crank-angle degrees* only increases slowly with increasing engine speed. If this were not the case, engines would not be able to operate over the wide range of speeds (600 to 6000 rev/min) that they do. Figure 9.21 shows how the interval between the spark discharge and 10% mass fraction burned, the flame development angle $\Delta\theta_d$, and the interval between the spark and 90% mass fraction burned, the overall burning angle $\Delta\theta_d + \Delta\theta_b$ (see Sec. 9.2.3), vary with engine speed.[16] The two intervals both increase by a factor of 1.6 for a factor of 4 increase in engine speed; that is, the burning rate throughout the combustion process increases almost, though not quite, as rapidly as engine speed. Additionally, at a given engine speed, increasing in-cylinder gas velocities (e.g., with intake generated swirl or tumble) increases the burning rate: the flame size for the swirling flows in Fig. 9.20 is larger than for the quiescent case with the same plug location at the crank-angle intervals after spark as shown. Increasing engine speed and introducing swirl both increase the levels of turbulence in the engine cylinder at the time of combustion (see Sec. 8.6). Increased turbulence increases the rate of development and propagation of the turbulent premixed engine flame.

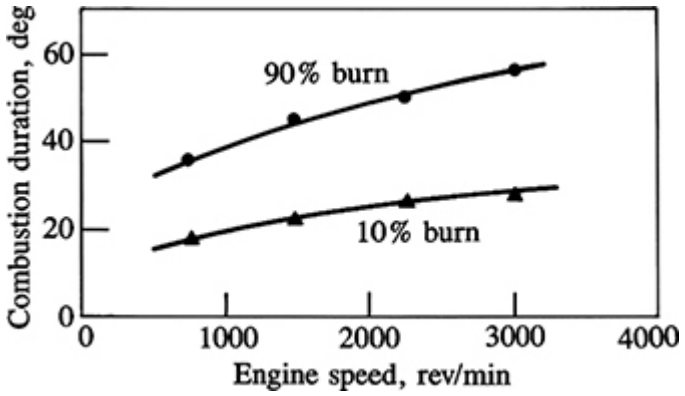


Figure 9.21 Effect of engine speed on flame-development angle (0 to 10% burned) and overall burning angle (0 to 90% burned). $\varphi = 1.0$, intake pressure = 0.54 atm, spark = 30° BTC.[16]

It is also well established that unburned mixture composition and state affect the burning rate. Reducing the inlet pressure (and maintaining the ratio of exhaust to inlet pressure fixed to hold the residual gas fraction constant) increases both the flame development and rapid burning angles. The relative air/fuel ratio or fuel/air equivalence ratio affects the burning rate. Both flame development and burning angles show a minimum for slightly rich mixtures ($\varphi \approx 1.1$) and increase significantly as the mixture becomes substantially leaner than stoichiometric. [16, 17] The burned gas fraction in the unburned mixture, due to the residual gas fraction *and* any recycled exhaust gases, affects the burning rate: increasing the burned gas fraction slows down both flame development and propagation. Fuel composition changes can be significant also. Mixtures of isooctane or conventional gasolines with air and burned gases (at identical conditions) have closely comparable burning rates; propane, methane, methanol, and ethanol mixtures exhibit a modest differences in burning rate; and hydrogen-air mixtures exhibit a substantial difference. The basic combustion chemistry of the fuel, air, burned gas mixture influences the combustion process. However, the relative importance of combustion chemistry effects depends on combustion chamber design and burn rate. Faster burning engines (which have higher turbulence) are less sensitive to changes in mixture composition, pressure, and temperature than are slower burning engines (which have lower turbulence). The effects of chamber geometry, gas motion, and gas composition and state are interrelated. [21]

9.3.2 Flame Structure

Figure 9.22 shows a schematic of the turbulent SI engine flame illustrating its structure and the relevant physical parameters. As indicated, the turbulent flame is a wrinkled thin reaction-sheet "laminar flame." The schematic shows the average leading and trailing boundaries of the flame (solid lines) with the wrinkled flame sheet in between, and the mean flame location (dashed line). The turbulent flame moves forward into the unburned mixture ahead of the flame—largely convected by the "random" turbulent motion (and any mean motion) of the gases within the combustion chamber. Relative to the unburned mixture ahead of the flame, this mean flame contour moves at the turbulent flame speed, S_T. Locally, each portion of this wrinkled reaction-sheet flame, as indicated in the schematic, moves orthogonally into the unburned mixture ahead of it at the local laminar flame speed, S_L. The wrinkled laminar flame sheet is thin: the turbulent flame brush—the average distance separating the leading and trailing locations of the wrinkled laminar flame—is thick. [22, 23]

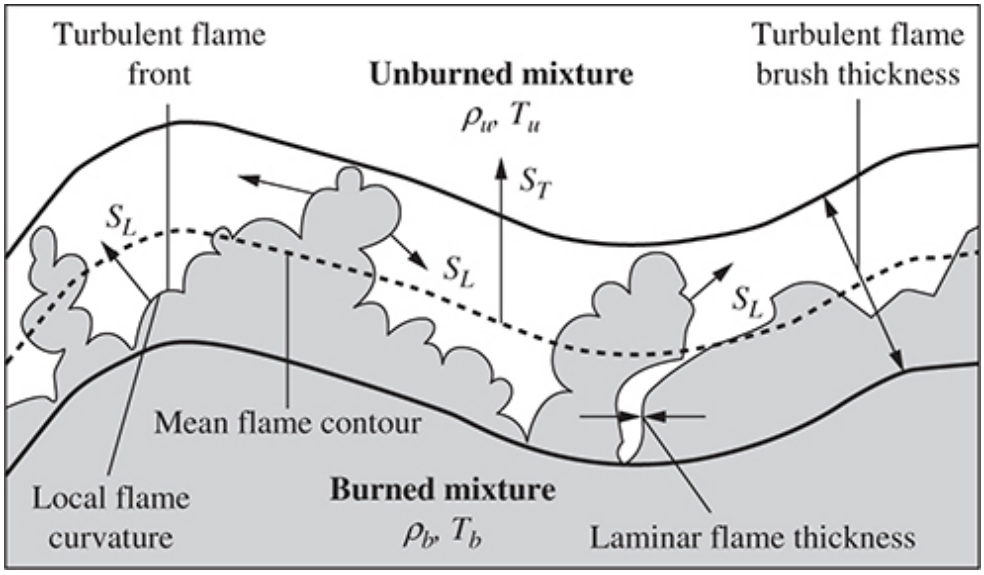


Figure 9.22 Schematic of the structure of the turbulent SI engine flame. Note the relevant flame-defining parameters. [22, 23]

Laminar premixed flames are characterized by their laminar flame speed S_L and a laminar flame thickness δ_L (see Sec. 9.1.1). The laminar flame

speed is the velocity at which the flame propagates into quiescent unburned mixture ahead of the flame. There are several ways to define the thickness δ_L of a laminar flame. [24] Dimensional arguments give the most commonly used definition. It is based on the thermal diffusivity $\alpha(= k/\rho c_p)$ as indicated in Fig. 9.1, or on molecular diffusivity D_L (see Sec. 4.8) of radical species through the flame:

$$\delta_L \approx \alpha/S_L \approx D_L/S_L$$

since the Lewis number (= Schmidt number/Prandtl number = a/ DL) is close to unity.

Turbulent flames are also characterized by the root mean square value of the velocity fluctuations, the turbulence intensity u' [Eq. (8.3)], and the various length scales of the turbulent unburned-mixture motion ahead of the flame. These various turbulence length scales are the integral length scale l_I [Eq. (8.8)] (a measure of the size of the large energy-containing structures of the flow) and the Kolmogorov scale l_K [Eq. (8.11)] which defines the smallest structures of the flow where kinetic energy is dissipated via molecular viscosity. Several other dimensionless parameters characterize turbulent premixed flames. The dimensionless parameter used to define the turbulence is the turbulent Reynolds number, $\mathrm{Re}_T = u' l_I/\nu$. For homogeneous and isotropic turbulence (i.e., no preferred direction), the integral and Kolmogorov scales are related by Eq. (8.14) : $l_K/l_I = \mathrm{Re}_T^{-3/4}$. A characteristic turbulent eddy turnover time τ_T can be defined as

$$\tau_T = \frac{l_I}{u'}$$

A characteristic chemical reaction time is the residence time in a laminar flame:

$$\tau_L = \frac{\delta_L}{S_L}$$

The ratio of the characteristic eddy turnover time to the laminar burning time is called the *Damköhler number*:

$$\mathrm{Da} = \frac{\tau_T}{\tau_L} = \left(\frac{l_I}{\delta_L}\right)\left(\frac{S_L}{u'}\right) \tag{9.32}$$

It is an inverse measure of the influence of the turbulent flow on the chemical processes occurring in the flame. Other ratios are of interest. The ratio δ_L / l_K is a measure of the stretch or local distortion to which a laminar flame is subjected by the turbulent flow. Unless $l_I / \delta_L >> 1$, the concept of a localized flame region has little significance. The ratio u' / S_L is a measure of the relative impact of the turbulence on the flame.

An additional consequence of the turbulence is the local stretching or straining of the reaction sheet that this flame wrinkling produces. Flame staining is quantified by the product of the Lewis number, Le, and the Karlovitz Stretch Factor, *K*. The Lewis number is the ratio of thermal diffusivity *a* to molecular diffusivity *D*: that is, Le = *a*/ *D*. The Karlovitz Stretch Factor is the product of the turbulent strain rate and the transit time *tL* for flow through the laminar flame:

$$K = \left(\frac{1}{A_L} \cdot \frac{dA_L}{dt}\right)\tau_L = \left(\frac{u'}{\ell_T}\right)\left(\frac{\delta_L}{S_L}\right) \tag{9.33}$$

where A_L is the laminar flame area, u' is the turbulence intensity, l_t is the Taylor microscale of the turbulence [see Eq. (8.12)], δ_L is the laminar flame thickness, and *SL* is the laminar flame speed. The higher the value of the product Le *K,* the more important is the impact of flame stretch in reducing the laminar burning velocity. Under normal engine operating conditions, the Lewis number (for a stoichiometric mixture) is close to 1.0, and *K* is of order 0.1 so the effect is small. For lean mixtures, Le increases above unity, and higher engine speeds increase the amount of stretch. At the lean operating limit of the engine (see Sec. 9.4.3), these combined effects are the likely cause of misfire—the extinguishing of the flame.

Table 9.1 gives values for the parameters, velocities, and length scales that characterize this engine flame under typical operating conditions. Note that u' and S_T increase with increasing engine speed (which is why fast-enough combustion can be achieved over the full range of engine speeds—600 to 6000 rev/min). Turbulence length scales decrease with increasing

speed, as expected from basic turbulence relationships. Adding air to a stoichiometric mixture (which makes it lean) or extra residual and/or EGR (which dilutes the unburned mixture) reduces the laminar flame speed by decreasing the temperature and concentration gradients in the laminar flame: see Fig. 9.1.

TABLE 9.1 Typical values of quantities which characterize SI engine flames* (Speed = 1500 rev/min; stoichiometric mixture)

Turbulence intensity, u'	2 m/s
Turbulent Reynolds no., Re_T	300
Damköhler no., Da	20
Karlovitz Stretch Factor, K	0.2
Integral scale, l_I	2 mm
Taylor microscale, l_T	0.7 mm
Kolmogorov scale, l_K	0.03 mm
Laminar flame thickness, δ_L	0.02 mm
Laminar flame speed, S_L	0.5 m/s
Ratio u'/S_L	4
Ratio S_T/S_L	4
Mean flame radius of curvature	2 mm

*Values taken from Refs. 22, 24–26.

Figure 9.23 is a plot of Damköhler number versus turbulent Reynolds number. It shows different regimes of turbulent flames. It has been assumed that $\alpha \approx D_L \approx \nu$ (the kinematic viscosity) and that the relationships for homogeneous isotropic turbulence are valid. Two regimes—distributed reactions and reaction sheets—are normally identified. In the distributed reaction regime, chemical reactions proceed in distributed reaction zones and thin-sheet flames do not occur. A sufficient condition for this regime is $l_I << \delta_L$. In the reaction sheet regime, propagating reaction fronts are wrinkled and convoluted by the turbulence. A sufficient condition for the existence of reaction sheets is $l_K >> \delta_L$. For $Re_T > 1$, there is a region in Fig. 9.23 where $l_I > \delta_L > l_K$: the characteristics of flames in this regime are unclear.

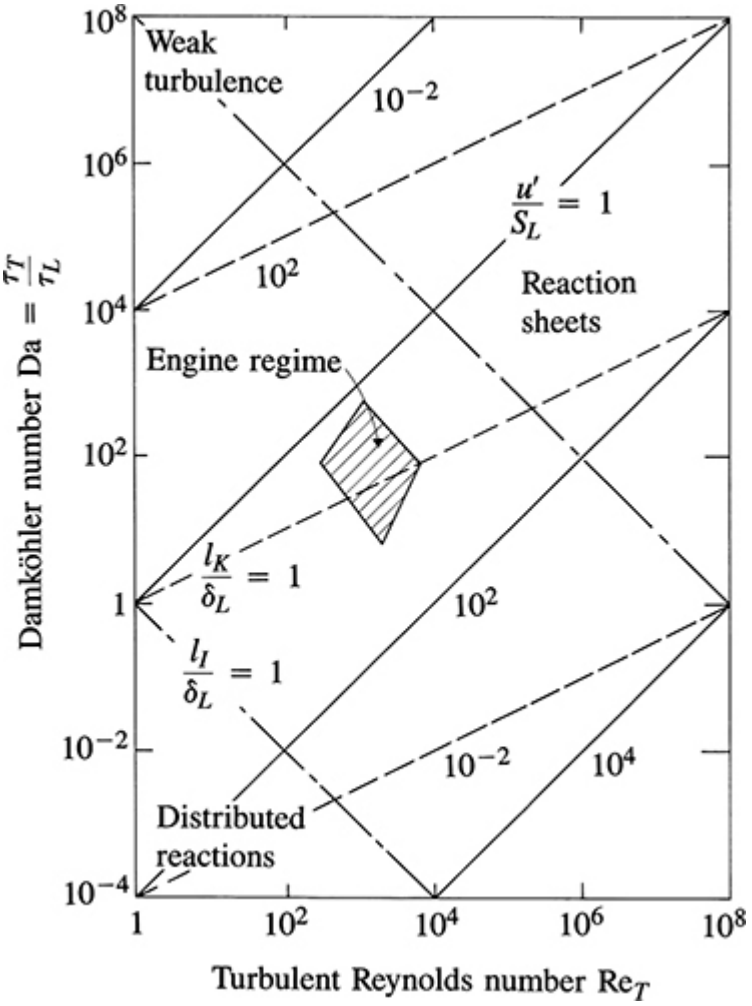


Figure 9.23 Different turbulent flame regimes on a plot of Damköhler number versus turbulent Reynolds number: u' is turbulence intensity; S_L is laminar flame speed; l_I, l_K, and δ_L are the integral scale, Kolmogorov scale, and laminar flame thickness, respectively.[27]

Values of Da and Re_T for a typical spark-ignition engine operation (the cross-hatched region in Fig. 9.23) lie predominantly in the reaction sheet flame regime. Engine operation at high speed (the lower right boundary) and low load (the lower left boundary) gives values of Da and Re_T which fall below the $l_K/\delta_L = 1$ line. This is largely due to the low values of laminar flame speed that result from the high amounts of residual gas and EGR under these conditions (see Sec. 9.3.4). Whether the flame structure under these conditions is significantly different is not known. Observations of engine

flames to date, described below, lie above the $l_K/\delta_L = 1$ line, within the reaction sheet regime. One would expect the structure of the flame in a spark-ignition engine, once developed, to be that of a thin reaction sheet wrinkled and convoluted by the turbulent flow.

Detailed observations have been made of flame structure from ignition to the flame extinguishing at the far cylinder wall. A flame develops from the spark discharge, which initiates combustion as follows. In the breakdown phase of ignition (first 100 nsec), a cylindrical discharge between the spark plug electrodes of some 1 mm diameter is established. As electrical energy is fed into the discharge, the plasma reaches temperatures of order 60,000 K, the discharge expands and exothermic chemical reactions rapidly occur, initiating a propagating flame. [23] Figure 9.24*a* shows how this development of a flame kernel occurs, with three schlieren photographs taken at 20, 100, and 310 μs after spark discharge onset, of the spark plug electrode gap in a special visualization engine. This sequence, of duration approximately 300 μs, corresponds to about 2 crank-angle degrees. The outer boundary of this developing flame kernel is roughly spherical, except where impeded by the plug electrodes, and is smooth with modest irregularities corresponding to a thin reaction zone with high-temperature gases inside. The extremely high discharge-generated gas temperatures result in rapid heat conduction to the surrounding gas and the electrodes. When the flame radius is small (~1 mm), flame curvature and stretch effects on the growth rate are significant. Then, as this developing sheet-like flame grows, it interacts with the turbulent flow field in the vicinity of the spark plug: the flame's outer surface becomes increasingly convoluted, and the flame center can be convected away from the plug in a direction and with a velocity that can vary substantially cycle-by-cycle, as seen in Fig. 9.24 *b*. [28] Both the bulk flow (through flame kernel convection), and the turbulence intensity (and corresponding turbulence length scales) in the vicinity of the spark discharge affect this stage of the flame development process.

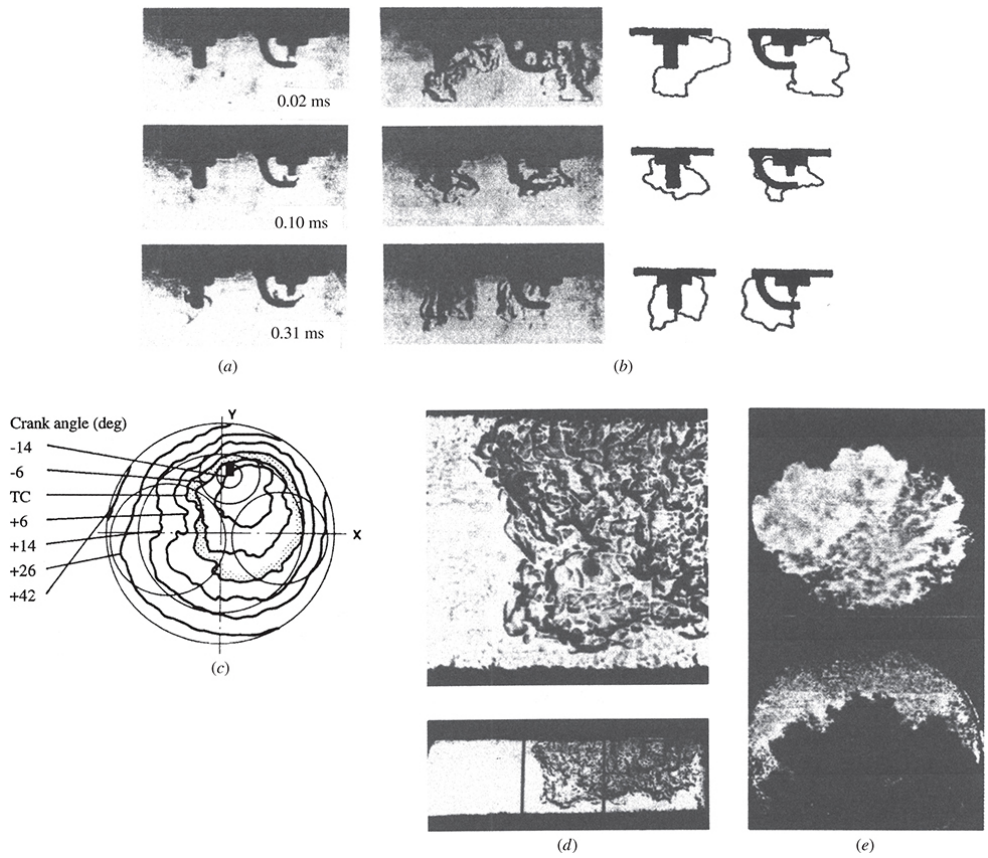


Figure 9.24 Images of SI engine flames using schlieren photography, flame radiation and laser-induced fluorescence techniques. (*a*) Photos of spark discharge and flame kernel development from a single cycle; stoichiometric operation, 1400 rev/min, mixture pressure and temperature at ignition 2 bar and 625 K. (*b*) Three consecutive cycles 0.84 ms (7 crank-angle degrees) after spark onset, showing the developing turbulent flame and its outer boundary; engine operating conditions as in (*a*). [28] (*c*) Flame contours illustrating flame propagation in one cycle: flame outer boundaries obtained from high-speed schlieren cinematography, and with multi-ionization probes; note the approximately spherical outer shape of the flame; 3000 rev/min, bmep = 5 bar, $A/F = 13$, spark 45° BTC, peak pressure 20 bar. [29] (*d*) Schlieren photo of a fully developed turbulent wrinkled-reaction-sheet SI engine flame: lower photo shows full square-cross-section combustion chamber; upper photo shows center portion enlarged. Operating conditions: stoichiometric, propane fuel, 1400 rev/min, 0.5 bar intake pressure. [30, 31] (*e*) Simultaneous images of flame radiation (top) and planar laser-induced-

fluorescence of unburned hydrocarbon fuel (bottom); image timing is 4° BTC; imaging plane is through the disc-shaped clearance volume. [32]

The increasingly wrinkled flame surface moves outward—due to both the flame propagation into the unburned mixture ahead of the flame *and* the expansion of each element of gas as it burns (Sec. 9.2.1)—in an approximately spherical manner from its center, as shown in Fig. 9.24*c*.[29] The flame contours shown define the fraction of the combustion chamber volume enflamed. Note that (see Sec. 9.2.1) with a centrally located spark plug (as in four-valve per cylinder SI engines) in a disc-shaped combustion chamber, only about 10% of the charge has burned when the flame radius is half the cylinder radius.

The wrinkled structure of the flame continues to develop as it propagates across the chamber. Evidence, largely from schlieren photographs and studies of flame structure with laser diagnostics, shows that early in the burning process the flame is a thin, moderately wrinkled but simply connected, reaction sheet between unburned and burned gas. The thickness of this flame front is about 0.1 mm, essentially the thickness of a laminar flame under the prevailing conditions. The scale of the wrinkles is typically 1 or 2 mm at engine speeds of 1000 to 2000 rev/min. As the flame propagates across the chamber, the thickness of the reaction sheet front remains roughly constant, the sheet-like flame becomes more convoluted, and the scale of the wrinkles tends to decrease with time. [30]

Additional insight into the structure of the developed engine flame can be obtained by examining the flame's leading edge. Figures 9.24 *d* and *e* show results from three different experimental techniques often used to visualize engine flames in their turbulent stage. Figure 9.24 *d* is a schlieren image of a turbulent engine flame in a square-cross-section visualization engine.[31] The flame shown was initiated at the spark plug electrode gap in the upper right-hand side of the lower photo and is propagating downward toward the piston crown and to the left toward the far cylinder wall. A portion of this photo is shown enlarged at the top. Schlieren photographs are integrated images along the optical path of the parallel light beam. Given that this flame is propagating outward in a roughly spherical manner, the schlieren image of the flame moving to the left can be interpreted as a tangential view of the spherical turbulent wrinkled reaction-sheet flame, where those sheet regions

that have moved ahead of the mean sheet or flame location appear as "hills," stacked beside and behind each other "on the horizon." That portion of the schlieren image that is viewed perpendicular to the plane of the mean flame location appears as an aerial view of "hilly terrain," where the "bottoms of the valleys" appear as lines. A feature of the wrinkled flame sheet model, where the turbulence distorts and convolutes the reaction sheet, is that the local laminar forward-diffusion of the sheet rounds the leading flame sheet regions and forms cusps at the trailing flame sheet regions (see Fig. 9.22).

Figure 9.24 *e* shows simultaneous images obtained from flame radiation, and from planar laser-induced fluorescence (LIF) from fuel vapor in the unburned mixture in an optically accessible spark-ignition engine with a disc-shaped combustion chamber.[32] The flame radiation technique produces an outer flame image similar to that produced by the schlieren technique. Two-dimensional planar imaging of flames has become a well-developed and valuable technique for examining the details of flame structure. The example shown, which identifies the unburned mixture region in a plane through the clearance height, shows finger-like regions of unburned gas that have penetrated into the turbulent flame, and similar regions of burned mixture that have moved ahead of the mean reaction-sheet location. These two-dimensional planar images usually show a more irregular boundary between the burned and unburned regions than is apparent with the schlieren visualization technique, which integrates density-gradient-produced refractive index changes along the optical path.

As explained earlier in this section, the turbulent spark-ignition flame is in the wrinkled reaction-sheet regime of turbulent combustion. The turbulent Reynolds number is in the range of 100 to 1000. The Damköhler number, which is the ratio of the turbulent eddy turnover time to the residence time in the laminar flame, is of order 100 (the fast chemistry regime), and the smallest turbulent eddy size is larger than the laminar flame thickness so the reaction sheet structure is little affected by the turbulence. The turbulence wrinkles and distorts the flame, increasing the area of thin reaction sheet contained within the turbulent flame zone (often called the flame brush) by a factor that is substantial and that increases with increasing turbulence. Most important, this increasing reaction-sheet area increases the mass-burning rate. An opposing effect that wrinkling also produces is the stretching of the flame sheet, which slows down the molecular diffusive processes within the

flame. Whether this flame-stretching effect is significant depends on strain rate and mixture composition.

The turbulence intensity— u' the root mean square value of the fluctuating in-cylinder flow velocity—at time of combustion, has a value about equal to the half the mean piston speed. Hence, the ratio of reaction sheet area (often called the laminar flame area) to the turbulent flame frontal area (for which *flame envelope area* is a useful term) increases with increasing engine speed. It is primarily for this reason that the mass burning rate, which is proportional to the reaction sheet area, increases almost linearly with engine speed, thereby maintaining the crank-angle interval required to burn from 10 to 90% of the in-cylinder charge almost constant over the full speed range of the engine. This ratio of laminar to turbulent flame area typically has a value of about 10 at 2000 rev/min.

The final stage of combustion, when the flame approaches the wall and extinguishes, has been much less extensively examined. There is a substantial literature on laminar flame quenching at a cool wall leading to the formation of quench layers—a thin layer of unburned fuel-air mixture adjacent to the wall (Sec. 11.4.3). It is often assumed that the local (laminar) regions of the wrinkled turbulent flame that come in contact with the wall quench as just described, while the remainder of the turbulent flame brush burns out as if the wall were not there. This inherently assumes that the unburned gas thermal boundary layer thickness is small relative to the turbulent flame brush thickness. However, as the flame approaches the wall, only successively smaller scales of turbulence are available to affect the flame so the final burn-up process will occur at a slower rate than would occur in the developed turbulent flame.

In terms of the engine *combustion system*—the chamber geometry and plug location, and the geometry of the cylinder head which determines the in-cylinder flow during combustion, the primary quantities of interest are:

- The mass burning rate of the unburned mixture, which through energy conservation, is directly linked to cylinder pressure and hence engine performance;
- The cycle-by-cycle variation in the mass burning rate because this, via torque fluctuations, determines the stability of engine operation;
- The burned gas heat transfer areas, which have a significant, though secondary effect on the energy balance.

In summary, at a more fundamental level, the factors that affect the mass-burning rate are the following:

- The mean flame frontal (or envelope) area contained within the boundaries of the combustion chamber;
- The relevant bulk and turbulence flow parameters during combustion —the local mean flow, the local turbulence intensity, and the flame-wrinkling length scales;
- The local unburned mixture composition and state, which determine the local laminar burning velocity.

9.3.3 Laminar Burning Speeds

An important intrinsic property of a combustible fuel, air, burned gas mixture is its laminar burning velocity. This burning velocity is defined as the velocity, relative to and normal to the flame front, with which unburned gas moves into the front and is transformed to products under laminar flow conditions. Details of the structure of this laminar flame, shown in Fig. 9.1, explain the significance of this quantity. A flame is the result of a self-sustaining chemical reaction occurring within the flame front where unburned mixture has been heated sufficiently to spontaneously combust. The flame consists of two major regions: a preheat and conduction zone, and a reaction zone. In the preheat zone, the temperature of the unburned mixture is raised mainly by heat conduction from the reaction zone: little reaction or energy release occurs and the temperature gradient is concave upward. Upon reaching the mixture's ignition temperature, exothermic chemical reactions begin. The conversion of chemical energy to thermal energy results in a zone where the temperature gradient is concave downward called the reaction zone. The thicknesses of the preheat and reaction zones can be calculated for one-dimensional flames from conservation equations of mass and energy. The thickness of the preheat zone $\delta_{L,\,\mathrm{ph}}$ is

$$\delta_{L,\mathrm{ph}} = \frac{4.6\bar{k}}{\bar{c}_p \rho_u S_L} \tag{9.34}$$

where $\bar{k}$ and $\bar{c}_p$ are the mean thermal conductivity and specific heat at constant pressure in the preheat zone, and S_L is the laminar burning velocity.

Thus, the factors which govern the laminar burning velocity of a specific unburned mixture—the velocity at which this flame structure propagates relative to the unburned gas ahead of it—are the temperature and species concentration gradients within the flame, and the mixture's transport and thermodynamic properties.

Laminar burning velocities at pressures and temperatures typical of unburned mixture in engines are usually measured in spherical closed vessels by propagating a laminar flame radially outward from the vessel center. The laminar burning velocity is then given by

$$S_L = \frac{dm_b/dt}{A_f \rho_u} \tag{9.35}$$

where the mass burning rate is determined from the rate of pressure rise in the vessel and A_f is the flame area. Because the laminar flame thickness [e.g., given by Eq. (9.33)] under engine conditions is of order 0.2 mm and is therefore much less than characteristic vessel dimensions, in applying Eq. (9.34) the flame can be treated as negligibly thin. Laminar burning velocities for methane, propane, isooctane, methanol, ethanol, gasoline, and hydrogen—premixed with air—at pressures, temperatures, and equivalence ratios which occur in engines have been measured using this technique. [33–37] Also, the effect of a burned gas diluent on laminar burning velocity with gasoline-air mixtures has been determined. [36] Correlations derived from these data are the most accurate means available for estimating laminar burning velocities for mixtures and conditions relevant to spark-ignition engines.

The effect of the mixture fuel/air equivalence ratio on laminar burning velocity for several hydrocarbon fuels and alcohols is shown in Fig. 9.25. The burning velocity peaks slightly rich of stoichiometric. The values for isooctane and gasoline are closely comparable. Data at higher pressures and temperatures have been fitted to a power law of the form:

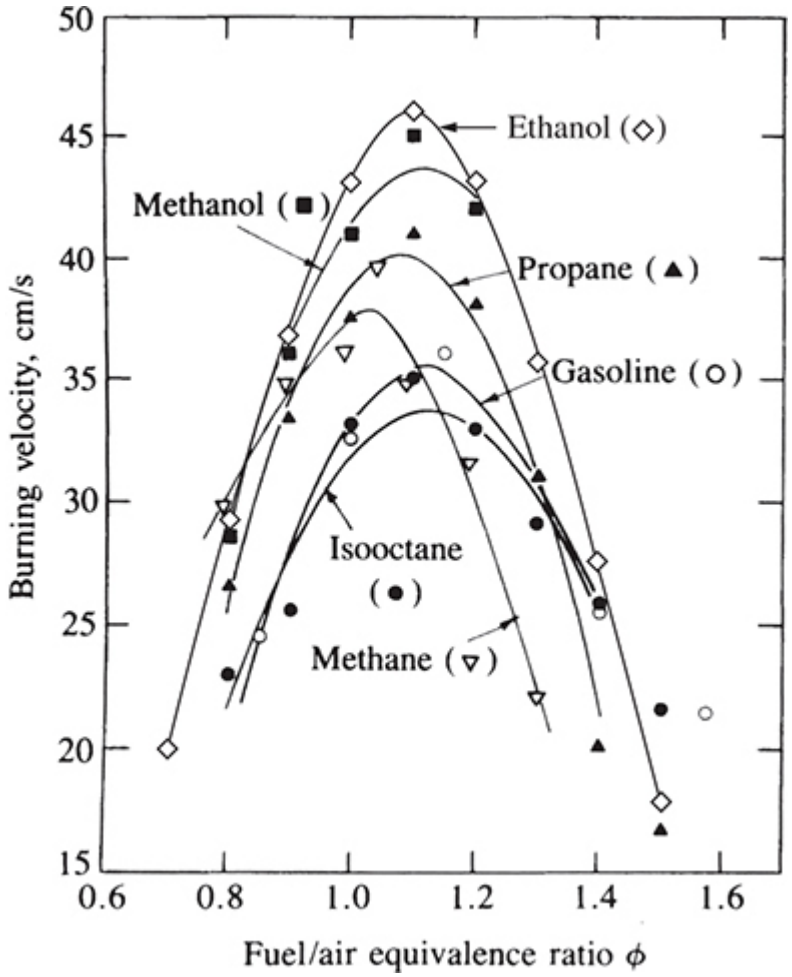


Figure 9.25 Laminar burning velocity for several fuels as a function of fuel/air equivalence ratio, at 1 atm and 300 K. Lines are least-squares polynomial fits to data.[33, 34, 35]

$$S_L = S_{L,0}\left(\frac{T_u}{T_0}\right)^{\alpha}\left(\frac{p}{p_0}\right)^{\beta} \tag{9.36}$$

where $T_0 = 298$ K and $p_0 = 1$ atm are the reference temperature and pressure, and $SL,_0$, a and b are constants for a given fuel, equivalence ratio, and burned gas diluent fraction. For propane, isooctane, ethanol, and methanol, these constants can be represented by

$$\alpha = 2.18 - 0.8(\phi - 1) \tag{9.37a}$$

$$\beta = -0.16 + 0.22(\phi - 1) \tag{9.37b}$$

and

$$S_{L,0} = B_m - B_\phi(\phi - \phi_m)^2 \tag{9.38}$$

where φ_m is the equivalence ratio at which $S_{L,\,0}$ is a maximum with value B_m. Values of φ_m, B_m, and B_{φ} are given in Table 9.2. [34]

TABLE 9.2 Parameters φ_m, B_m, and B_φ for Eq. (9.38)

Fuel	ϕ_m	B_m, cm/s	B_ϕ, cm/s	Ref.
Gasoline	1.21	30.5	−55	36
Isooctane	1.13	26.3	−85	34
Propane	1.08	34.2	−139	34
Ethanol	1.10	40.0	−200	37
Methanol	1.11	36.9	−140	34

For gasoline (a reference gasoline with average molecular weight of 107 and an H/C ratio of 1.69), additional data were available and were correlated by [36]

$$\alpha_g = 2.4 - 0.271\phi^{3.51} \tag{9.39a}$$

$$\beta_g = -0.357 + 0.14\phi^{2.77} \tag{9.39b}$$

For methane, simple equations such as (9.37 *a*, *b*) do not adequately correlate the data over the range of p and T_u relevant to engines. However, laminar burning velocity data from a spherical constant-volume bomb experiment have been obtained along an unburned gas isentropic path, as the pressure in the bomb rises during combustion. Variation in laminar burning velocity along such unburned gas isentropes does correlate with a power law:

$$S_{L,s} = S_{L,0}\left(\frac{\rho_u}{\rho_{u0}}\right)_s^{\varepsilon} \tag{9.40}$$

Values for $S_{L,\,0}$ and ε from the literature are summarized in Table 9.3.

TABLE 9.3 Parameters for methane-air laminar burning velocity correlation [Eq. (9.40)]

ϕ	p_i atm	$S_{L,0}$* cm/s	ε
1.0	0.5	49	0.51
1.0	1.0	35	0.2
0.8–1.2	1–8	†	0.17–0.19

*At 298 K initial temperature.
†See Fig. 9.25.

The presence of burned gas in the unburned cylinder charge due to residual gases and any recycled exhaust gases causes a substantial reduction in the laminar burning velocity. Any burned gas in the unburned mixture reduces the heating value per unit mass of mixture and thus, reduces the adiabatic flame temperature. It acts as a diluent. The effect of increasing burned gas or diluent fraction on laminar flame speed is shown in Fig. 9.26. The diluent used was a mixture of CO_2 and N_2, chosen to match the heat capacity of actual gasoline-air combustion products. [e] The proportional reduction in laminar burning velocity is essentially independent of the unburned mixture equivalence ratio, pressure, and temperature over the range of interest in engines. The data in Fig. 9.26 are correlated by the relation:

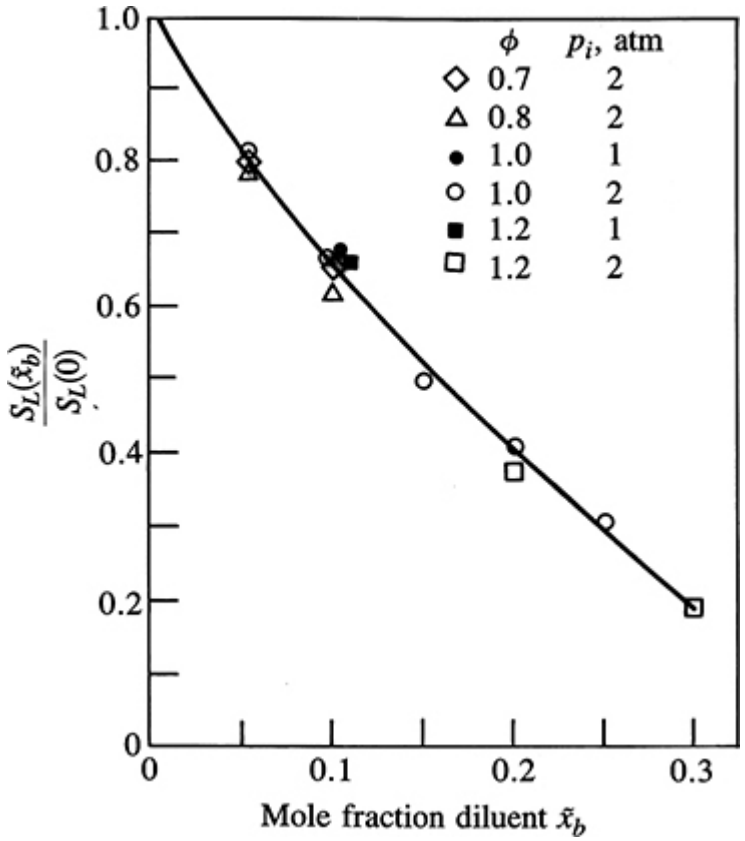


Figure 9.26 Effect of burned gas mole fraction $\tilde{x}_b$ in unburned mixture on laminar burning velocity. Fuel: gasoline. [36]

$$S_L(x_b) = S_L(\tilde{x}_b = 0)(1 - 2.06)\tilde{x}_b^{\,-0.77} \tag{9.41}$$

where $\tilde{x}_b$ is the mole fraction of burned gas diluent. Other studies corroborate the magnitude of this burned gas effect. [36]

Note that for equal heat capacity added to the unburned mixture, burned gases have a much larger effect on laminar burning velocity than does excess air. For example, the laminar burning velocity of a stoichiometric mixture as it is leaned to $\varphi = 0.8$ is reduced by 23%. Adding the same heat capacity of stoichiometric burned gases (which requires a burned gas mole fraction of 0.175) reduces the laminar burning velocity by 55%. [36] Allowance for the burned gas fraction in estimating laminar burning velocities for spark-ignition engines is most important.

The above correlations define the laminar burning velocity (of a given fuel) as a function of unburned mixture thermodynamic properties and

composition only. It has been assumed that flame thickness and curvature effects are negligible. Our interest in laminar burning velocity is twofold: first, it is used to define the characteristic chemical reaction time of the mixture in Eq. (9.32) ; second, a presumed consequence of the wrinkled thin-reaction-sheet turbulent-flame structure is that, locally, the sheet propagates at the laminar burning velocity. The above correlations adequately characterize a quiescent burning process. However, laminar flame propagation can be influenced by the local flow field in the unburned gas. If the flame thickness is less than the Kolmogorov scale, the primary effect is one of straining, which affects both the flame area (usually referred to as flame stretching for an area increase) and the local (laminar) burning velocity. Local straining can affect the laminar burning velocity and can cause flame extinction. The laminar burning velocity decreases with increasing strain rate, and the Lewis number of the unburned mixture has a significant influence on this rate of decrease. The Lewis number is the ratio of diffusivities of heat and mass. For stoichiometric mixtures it is close to one; it increases above unity as the unburned fuel-air mixture is leaned out. Thus, the local flow field may have a discernable effect on the local burning velocity of the thin laminar-like reaction-sheet flame, especially for lean or dilute mixtures.

9.3.4 Flame Propagation Relations

Combining the heat-release or mass burning rate analysis of Sec. 9.2.2 with an analysis of flame geometry and structure data allows us to quantify the behavior of spark-ignition engine flames. Flame photographs (such as those in Figs. 9.2, 9.18, and 9.20 and 9.24) effectively define the position of the front or leading edge of the turbulent engine flame. The leading "surface" of the enflamed zone, under normal engine conditions, is close to spherical: only in the presence of high swirl (Fig. 9.20) does substantial distortion of the flame shape occur. The schematic in Fig. 9.22 lays out the two primary approaches to analyzing flame propagation: the wrinkled laminar flame sheet approach, and the turbulent brush model. To a good approximation, the surface that defines the leading edge of the flame (ahead of which only unburned mixture exists) is a portion of the surface of a sphere. Figure 9.27 indicates the geometrical parameters which define this flame surface: r_c , a_c , z_c, the coordinates of the flame center, r_f the radius of the best-fit sphere to

the flame front outer boundary, and the geometry of the combustion chamber walls. The flame is initiated at the spark plug; however, the center may move away from the plug during the early stages of its development as shown. Here, we define the *flame front area Af* as the spherical surface of radius r_f coinciding with the leading edge of the flame contained within the combustion chamber, and the *enflamed volume* V_f as the volume within the chamber behind this flame front.

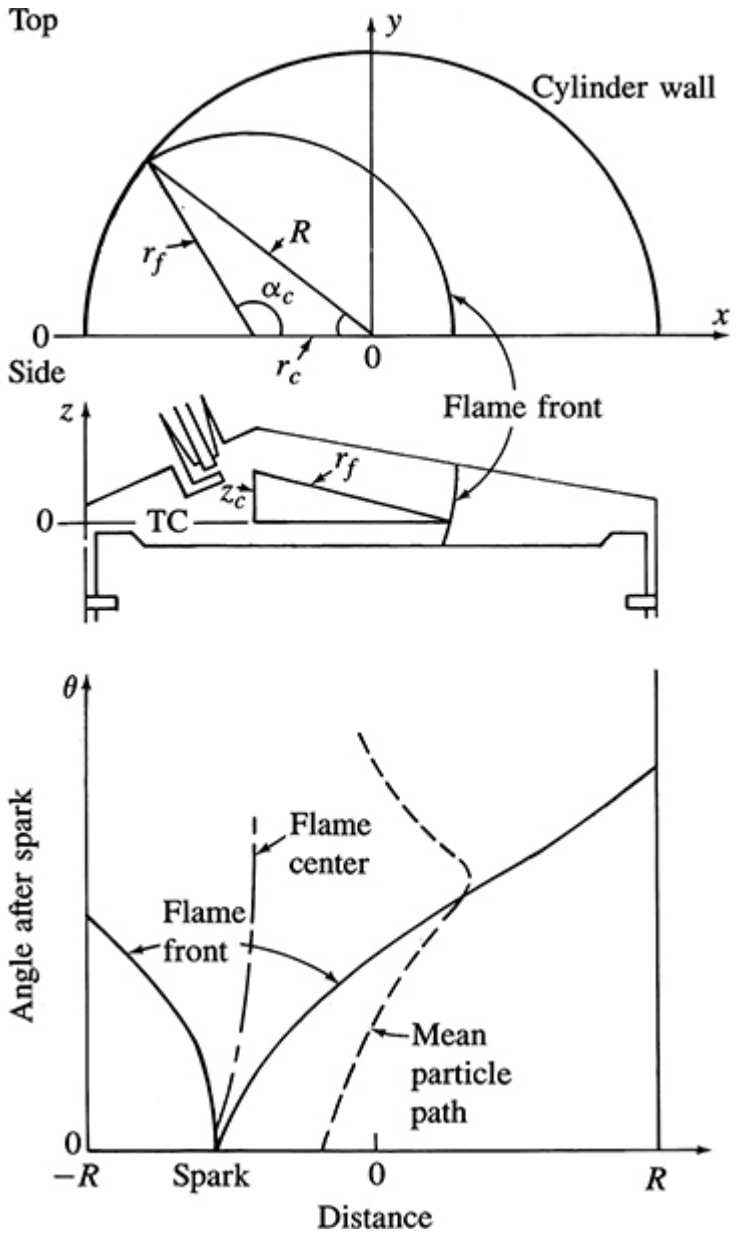


Figure 9.27 Schematic of spherical flame front in engine combustion chamber identifying parameters which define flame geometry. (*From Beretta*

et al. [4])

Wrinkled Flame Sheet Approach

The thermodynamic analysis of cylinder pressure data allows us to define additional geometrical parameters. The *burned gas radius* r_b is the radius of the spherical surface within the combustion chamber which would contain all the burned gas behind it, that is,

$$V_b(r_b, r_c, \alpha_c, z_c) = V_b(p, \theta) \tag{9.42}$$

The *spherical burning area* A_b is the area of this spherical surface, that is,

$$A_b = \frac{\partial V_b(r_b, r_c, \alpha_c, z_c)}{\partial r_b} \tag{9.43}$$

The *laminar burning area* A_L is the surface area the flame would have if it burned locally at the laminar flame speed, that is,

$$A_L = \frac{dm_b/dt}{\rho_u S_L} \tag{9.44}$$

where S_L is the laminar flame speed in the unburned mixture ahead of the flame (see Sec. 9.3.3).

Several velocities can be defined. The *mean expansion speed* of the *front* u_f is given by

$$u_f = \frac{dV_f/dt}{A_f} \tag{9.45}$$

where A_f is the surface area of the "best-fit" sphere through the leading edge of the flame enclosed within the combustion chamber, and V_f is the volume behind that leading flame surface into which unburned mixture is entrained. The *mean expansion speed* of the *burned gas* u_b is

$$u_b = \frac{dV_b/dt}{A_b} \tag{9.46}$$

The *burning speed* S_b is defined by

$$S_b = \frac{dm_b/dt}{\rho_u A_b} \tag{9.47}$$

S_b is the (turbulent) flame speed S_T with which the flame frontal or envelope surface (of area A_f) propagates into the unburned gas. The *mean gas speed* just ahead of the flame front *ug* is

$$u_g = u_b - S_b \tag{9.48}$$

Note that combining Eqs. (9.44) and (9.47) gives the relation

$$S_b A_b (= S_T A_b) = S_L A_L$$

Also, it follows from Eqs. (9.31) , (9.46), and (9.47) that

$$\frac{u_b}{S_b} = \frac{\rho_u}{\rho_b}(1 - y_b) + y_b = \frac{\rho_u/\rho_b}{[(\rho_u/\rho_b) - 1]x_b + 1} \tag{9.49}$$

where S_b and S_T are synonymous. As x_b and $y_b \to 0$, u_b/S_b approaches the expansion ratio, ρ_u/ρ_b. As x_b and $y_b \to 1$, u_b/Sb approaches unity.

The variation of the above quantities during the engine combustion process, coupled with the photographs and discussion in Sec. 9.3.2 , provides substantial insight into the flame development and propagation process. Figure 9.28 shows results from an analysis of cylinder pressure data and the corresponding flame front location information (determined from high-speed movies through a window in the piston) of several individual engine combustion cycles.[4] The flame radius initially grows at a rate that increases with time and exhibits substantial cycle-by-cycle variation in its early development (Fig. 9.28*a*). Later ($r_f \gtrsim 30$ mm) the growth rate, which approximates the expansion speed u_b, reaches an essentially constant value.

The flame radius r_f is initially equal to the burned gas radius r_b: it increases above r_b as the flame grows and becomes increasingly distorted by the turbulent flow field (Fig. 9.28*b*). Eventually r_f - r_b goes to an essentially constant value of about 6 mm for $r_b \gtrsim 30$ mm. This difference, r_f - r_b, is approximately half the thickness of the turbulent flame brush: see Fig. 9.22.

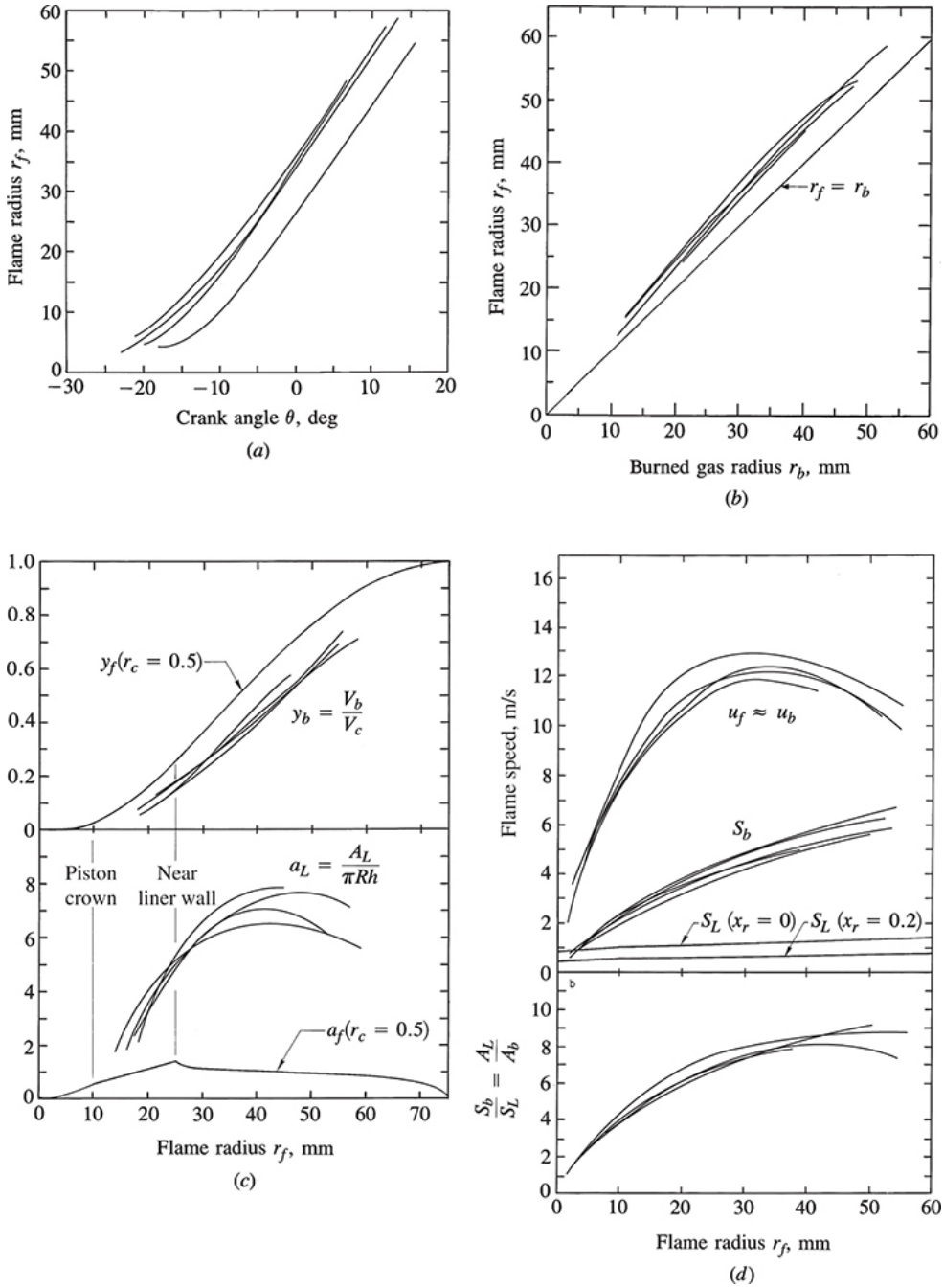


Figure 9.28 Variation of flame geometry and velocity parameters during four individual combustion cycles in an SI engine at 1044 rev/min, $\varphi = 0.98$, 1 atm inlet pressure: (*a*) flame radius r_f versus crank angle; (*b*) flame radius r_f versus burned gas radius r_b; (*c*) normalized enflamed volume y_f, burned volume y_b, normalized flame front area A_f, and laminar area A_L versus flame radius; (*d*) front expansion speed u_b, burning speed S_b, and laminar flame speed S_L versus flame radius. (*From Beretta et al.* [4])

Normalized enflamed and burned volumes, and normalized flame front area and wrinkled laminar-flame burning area, are shown in Fig. 9.28 *c*. Volumes are normalized by the clearance volume and areas by πRh, where h is the average clearance height and R the cylinder radius. Discontinuities occur in the flame area A_f at the points where the flame front contacts first the piston face and then the near cylinder wall. The laminar flame area A_L is initially close to the flame area A_f, and then increases rapidly as the flame grows beyond 10 mm in radius (Fig. 9.28*d*). During the rapid burning combustion phase the value of y_f is significantly greater than y_b. During this phase, the laminar area grows to exceed the flame surface or envelope area by almost an order of magnitude. These observations indicate the existence of substantial pockets of unburned mixture behind the leading edge of the flame.

The ratio of the volume of the unburned mixture within the turbulent flame zone (V_f - Vb) to the reaction-sheet area within the flame zone (A_L - A_f) defines a characteristic length

$$\ell_T = \frac{V_f - V_L}{A_L - A_f} \tag{9.50}$$

which can be thought of as the scale of the pockets of unburned mixture within the flame. For the data set of Fig. 9.28, l_T is approximately constant and about 1 mm. [30]

These flame geometry results would be expected from the previous photographic observations of how the flame grows from a small approximately spherical smooth-surfaced kernel shortly after ignition to a

highly wrinkled reaction-sheet turbulent flame of substantial *overall* thickness. Initially, the amount of unburned gas within the enflamed volume is small. During the rapid burning phase of the combustion process, however, a significant fraction (some 25%) of the gas within the turbulent flame brush is unburned.

The front expansion speed u_f, burning speed S_b, and laminar flame speed S_L are shown in Fig. 9.28 *d*. The expansion speed increases as the flame develops to a maximum value that is several times the mean piston speed of 3 m/s. The burning speed S_b (the turbulent flame speed S_T) increases steadily from a value close to the laminar flame speed at early times to almost an order of magnitude greater than S_L during the rapid burning phase. During this rapid burning phase, since (r_f - r_b) is approximately constant, the flame front expansion speed and the mean burned gas expansion speed are essentially equal. The difference between $u_b \approx u_f$ and S_b is the unburned gas speed u_g just ahead of the flame front. Note that the ratio u_b / S_b ($\approx u_f / S_b$ decreases monotonically from a value close to the expansion ratio (ρ_u / ρ_b) at spark, to unity as the flame approaches the far wall, as required by Eq. (9.49) .

The effect of flame propagation on the flow field in the unburned mixture ahead of the flame is important because it is the turbulence just ahead of the flame that determines the local burning velocity. Note that the mean flow varies cycle-by-cycle, the turbulence is not homogeneous, and the flame motion and shape show significant cyclic variations. The mean velocity normal to the front increases steadily from shortly after ignition, as the combustion-produced gas expansion displaces unburned mixture toward the walls. Whether the turbulence intensity increases due to the compression of the unburned mixture, which occurs during combustion, is unclear. Rapidly imposed distortions of a turbulent flow field produced by combustion-generated gas expansion would lead to an increase in turbulence intensity.

The variation of burning speed with engine speed and thus with the turbulence intensity, has also been carefully examined. A linear correlation between S_b and u' results for the rapid burning combustion phase. [38]

Once the flame front reaches the far cylinder wall, the front can no longer propagate, however, combustion continues behind the front until all the unburned mixture entrained into the enflamed region is consumed. This final burning or termination phase of the combustion process can be approximated

by an exponential decay in the mass-burning rate with a characteristic time constant τ_b (which varies with engine speed, and is of order 1 ms). Since these "peninsulas or islands" of unburned mixture behind the leading edge of the flame have a characteristic scale l_T, based on the laminar flame area [Eqs. (9.44) and (9.50)], it follows that

$$l_T = \tau_b S_L \tag{9.51}$$

In summary, the above flame data analysis shows that the relationships between r_f and r_b, V_f and Vb, A_f and A_L, u_f, S_b or S_T and S_L, are distinctly different in the three phases of combustion: (1) the development phase, where a growing, increasingly-wrinkled, thin reaction-sheet evolves into a turbulent flame from the essentially spherical small flame kernel established by the spark discharge; (2) the rapid-burning phase, where this thick "developed" turbulent flame propagates across the combustion chamber to the far wall during which most of the charge mass is burned; and (3) the termination phase after the flame front has reached the far wall and propagation of the front is no longer possible, when the remaining unburned mixture within the flame burns up. The burning velocity, in the rapid-burning phase of the combustion process, scales with turbulence intensity, which in turn scales with engine speed.

Turbulent Flame Brush Approach

The second approach to describing flame propagation is based on *entrainment* of unburned mixture into the "thick" turbulent flame brush, illustrated in Fig. 9.29. As unburned mixture crosses the flame surface a portion of it burns immediately; then a substantial fraction burns *within* the turbulent flame brush. This part of the entrained unburned mixture forms peninsulas, even islands, each surrounded by a thin reaction-sheet flame. These inclusions then burn up at a rate determined by their characteristic size and local laminar burning velocity S_L, as shown. Combustion in the turbulent flame can be thought of as an unburned mixture entrainment process, followed by a local diffusion-controlled burning process.

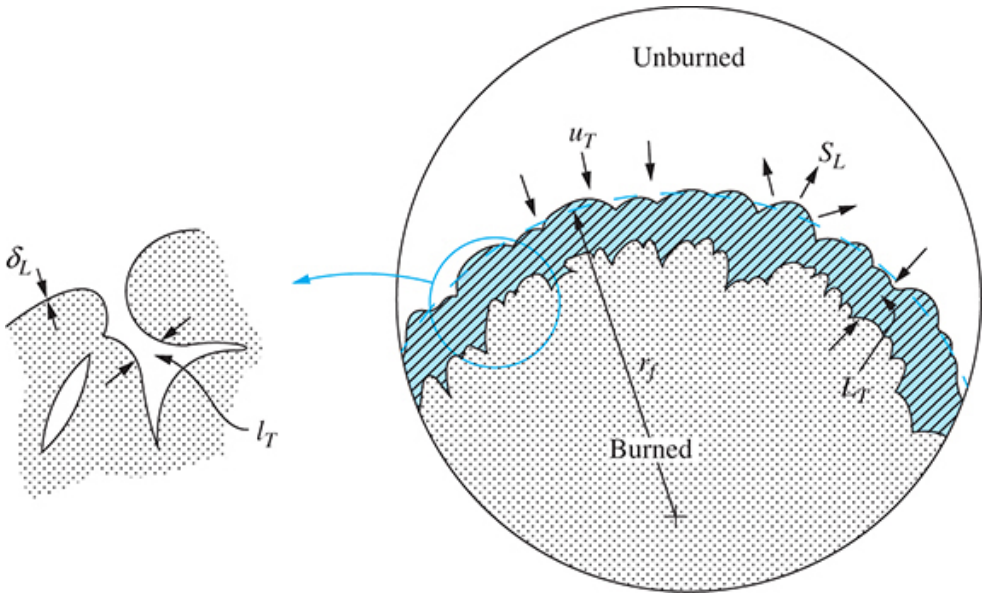


Figure 9.29 Schematic of turbulent premixed spark-ignition engine flame, illustrating the entrainment-based burning law of Eqs. (9.52) to (9.54). The approximately spherical front of the "thick" flame (dashed line) diffuses outward at the laminar flame speed S_L . Fresh mixture also crosses this front at a characteristic velocity u_T due to turbulent convection. Schematic on left show flame structure; δ_L is the reaction-sheet thickness, l_T is the characteristic scale of wrinkles in the sheet.

This conceptual model, illustrated in Fig. 9.29, results in two rate equations: one for the mass of unburned-mixture entrained into the flame front (or equivalently, entrained as that front moves into the unburned mixture) and a second for the mass burning rate (Keck et al. [4, 30, 39]). Turbulent convection brings fuel-air mixture into the flame at velocity, u_T. Molecular diffusion of the laminar thin reaction-sheet flame propagates it forward at the laminar flame speed, S_L. The two phenomena effectively add. Thus, the rate of entrainment of unburned mixture into the flame is

$$\frac{dm_e}{dt} = \rho_u A_f S_L + \rho_u A_f u_T (1 - e^{-t/\tau_b}) \tag{9.52}$$

where u_T is the characteristic speed of the turbulent convective transport process. The first term represents molecular (laminar) diffusion through the

flame front. The second term represents turbulent convection characterized by u_T. The bracketed term allows for the early flame development stage as the initially laminar flame kernel transitions to a turbulent flame. τ_b is a characteristic burning time for unburned mixture within the turbulent flame brush given by

$$\tau_b = \ell_T / S_L \tag{9.53}$$

The rate at which mixture burns is then,

$$\frac{dm_b}{dt} = \rho_u A_f S_L + \frac{m_e - m_b}{\tau_b} \tag{9.54}$$

where (m_e - m_b) represents the mass of unburned mixture already entrained into the thick turbulent flame (the cross-hatched region in Fig. 9.29) that has yet to burn:

$$(m_e - m_b) = \rho_u (V_f - V_b) = \rho_u (A_L - A_f)\, \ell_T \tag{9.55}$$

where l_T, V_f, V_b, A_L, and A_f are defined earlier in this section.

To summarize, as illustrated in Fig. 9.29, this approach combines the turbulent convective and laminar (molecular diffusive) propagation of the approximately spherical front of the "thick" turbulent flame into the unburned mixture ahead of the flame, with the local, distributed, laminar-like burn-up of the mixture entrained into the flame. Thus the name: "entrainment" or "eddy-burning" combustion model.

Note that for a quiescent mixture, $u_T \to 0$ or $l_T \to \infty$, and $S_b \to S_L$. Initially, at $t \approx 0$, $S_b \approx S_L$. In the developed (quasi-steady-state) turbulent flame combustion phase,

$$S_b \approx u_T + S_L \tag{9.56}$$

During the final burning stage after the flame front reaches the wall, A_f decreases exponentially to zero as does (m_e - m_b), the remaining unburned mass within the flame.

To apply Eqs. (9.52) and (9.54), the quantities u_T and τ_b or ($l_T = t_{b\,SL}$) must be evaluated. Two approaches have been taken: (1) use of empirical correlations for these variables, derived from engine flame data (such as those described in Sec. 9.3.4) and (2) use of more fundamental models to predict these quantities.

Keck derived the following correlations for u_T and l_T based on the application of Eqs. (9.52) and (9.54) to several sets of engine combustion data:

$$u_T = 0.08\bar{u}_i\left(\frac{\rho_u}{\rho_i}\right)^{1/2} \text{ and } \ell_T = 0.8L_{\text{iv}}\left(\frac{\rho_i}{\rho_u}\right)^{3/4} \tag{9.57}$$

u_T was found to be proportional to $\sqrt{\rho_u}$ (at time of spark) and to correlate well with mean inlet gas speed $\bar{u}_i = \eta_v(A_p/A_{\text{iv}})\bar{S}_p$, where η_v is volumetric efficiency, A_p is piston area, A_{iv} is the maximum open area of the inlet valve, $\bar{S}_p$ is mean piston speed, and ρ_i is intake air density. l_T appears to scale with intake valve lift, L_{iv} ; it decreases with increasing density at a rate proportional to $\rho_u^{-3/4}$. While u_T and l_T are not constant during the combustion process, their variation is modest. [30]

A quantitative comparison of predicted and measured flame radius as a function of time is shown in Fig. 9.30 for hydrogen and propane fuel-air mixtures which exhibit widely different behavior: the figure indicates both the behavior and validity of this thick flame entrainment model. Predicted burned gas expansion speeds u_b [see Eq. (9.46)] are shown in Fig. 9.30*a* as a function of burned gas radius; the parameters u_T and l_T were chosen to fit the propane data. Figure 9.30 *b* shows that the measured flame front radii, r_f, are in good agreement with the predicted flame and burned gas radii, r_f and r_b, for these two fuels. The initial expansion speed of hydrogen is about 10 times that of propane. Since $r_f \approx r_b$ for early times and $s_b \approx s_L$, this ratio is expected. As r_b becomes large, $r_f' r_b \rightarrow u_T \tau_b$. Note that τ_b ($= l_t / S_L$) is several times smaller for hydrogen mixtures than for propane mixtures since hydrogen's laminar flame speed is much higher.

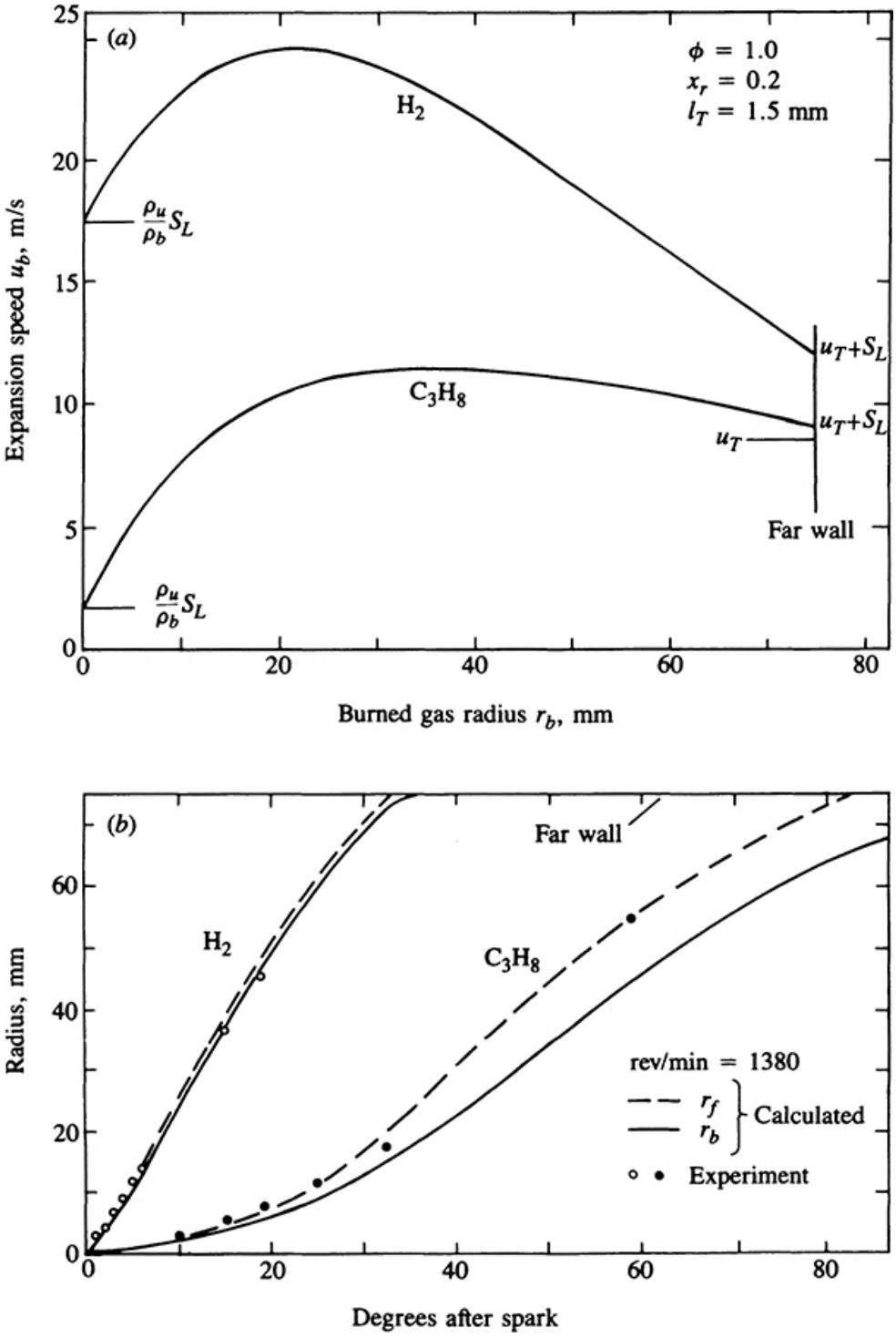


Figure 9.30 (*a*) Calculated burned gas expansion speed u_b during combustion for stoichiometric hydrogen-air and propane-air mixtures as a function of burned gas radius r_b. *(b)* Comparison of experimentally measured (points) and calculated (dashed curve) flame radii r_f for these mixtures as a

function of crank angle. Also shown (solid curve) is the burned gas radius r_b. [30]

An adaption of this approach developed by Tabaczynski et al. [39] used basic turbulent flame propagation theory to argue that l_T in Fig. 9.29 is essentially the Taylor microscale l_M, which is a function of the integral length scale l_I and the turbulent Reynolds number as indicated by Eq. (8.15) . From these turbulence assumptions they argued that the flame propagates with a velocity $u' + S_L$, where u' is the local turbulence intensity. Thus, in Eqs. (9.52) and (9.53), u_T and τ_b are given by

$$u_T \approx u' \quad \text{and} \quad \tau_b = \frac{l_T}{S_L} \approx \frac{l_M}{S_L} \tag{9.58}$$

where l_M, the microscale, is determined from the integral scale and the turbulent Reynolds number via Eq. (8.15) assuming that the turbulence is homogenous and insentropic. The task therefore becomes one of evaluating u' and l_I.

9.3.5 Combustion with Direct Fuel Injection

This chapter has described combustion in spark-ignition engines operating with essentially "premixed" fuel-vapor/air mixtures. The tacit assumption is that the unburned mixture within the cylinder is sufficiently close to homogeneous so that mixture nonuniformities are not important. This is a valid assumption with port fuel injection where the fuel for each cylinder is injected into the intake port largely onto the intake valve head and port walls in a warmed-up engine at normal operating conditions. At more extreme operating conditions, such as approaching maximum power and when the engine is cold, this close-to-homogeneous mixture assumption is less valid but the engine's combustion process is still only modestly affected. With direct fuel injection into each of the engine's cylinders, the mixture preparation process is significantly different from the port fuel injection process. (See Chap. 7 , which discusses SI engine mixture preparation.)

One direct-fuel-injection approach—injection of fuel into each cylinder largely during the intake stroke—does produce a close to homogeneous fuel-vapor/air mixture within the cylinder (as is intended) and combustion is then an essentially premixed turbulent flame propagation process as has been discussed already in this chapter. While this early-direct-injection in-cylinder mixture is "less homogeneous" than mixture resulting from port fuel injection, the flame propagation differences are small. The air pollutant emission differences may be more significant.

However, direct fuel injection into each engine's cylinder provides the opportunity with *late* fuel injection (during the latter part of compression stroke) to operate the engine *fuel lean by* stratifying the fuel-vapor/air mixture within the combustion chamber so that mixture adjacent to the spark plug is close to stoichiometric (and thus burns rapidly) and the excess air is located around this contained ready-to-burn fuel-vapor/air "cloud". In this lean stratified operating mode, the spark-ignited combustion process is different.

Thus, direct-fuel-injection approaches are categorized as *homogeneous* or *stratified*. In this section, we explore the engine combustion differences under stratified operating conditions. The operating regime where stratified operation would improve efficiency is at engine loads and speeds that are less than about half the maximum values. From mid- to high-loads (and speeds) these stratified DI SI engine concepts transition to homogeneous operation, with the overall relative air/fuel ratio shifting to stoichiometric to realize high specific power and more effective emissions control. The aspects of part-load stratified operation that contribute to higher efficiency are:

- With lean operation overall, compared with stoichiometric, the intake manifold pressures are higher so pumping losses are reduced.
- With excess air, the average burned gas specific heat ratio gb is higher (due to lower concentrations of CO_2 and H_2O; see Fig. 4.18 and Table 4.7), and average burned-gas temperatures are lower (further increasing gb); this increases the expansion stroke work extracted.
- The lower (average) burned gas temperature also results in reduced heat losses to the combustion chamber walls.[40]

Figure 9.31 shows pressure versus crank-angle measurements and corresponding burn rate profiles for several different gasoline engine

concepts at light and mid-load. The data come from a single-cylinder engine that could embody the different concepts compared, as well as engine simulation studies. [41] The concepts compared in Fig. 9.31 are: spark-ignited port fuel injection (SI-PFI)—stoichiometric mixture turbulent flame propagation in a port-fuel-injected engine; spark-ignited direct fuel injection (SI-DI)—the same combustion process in a direct-injection engine operated in stoichiometric homogeneous mode; spark-ignited variable valve actuation (SI-VVA)—homogeneous DI operation with late intake valve closing so the engine operates essentially unthrottled at these loads; spark-ignited direct-injection stratified charge (SI-STRAT)—direct fuel injection with a fuel-lean mixture overall, in stratified combustion mode (the primary topic, here); and controlled autoignition (CAI)—a different combustion mode with a lean and dilute largely premixed mixture that ignites spontaneously through control of unburned mixture temperature near TC. [f] For each of these five engine setups, fuel injection timing and spark timing were adjusted to be close to optimum fuel consumption without compromising combustion stability and the achievement of low engine-out emissions (e.g., through use of substantial EGR).

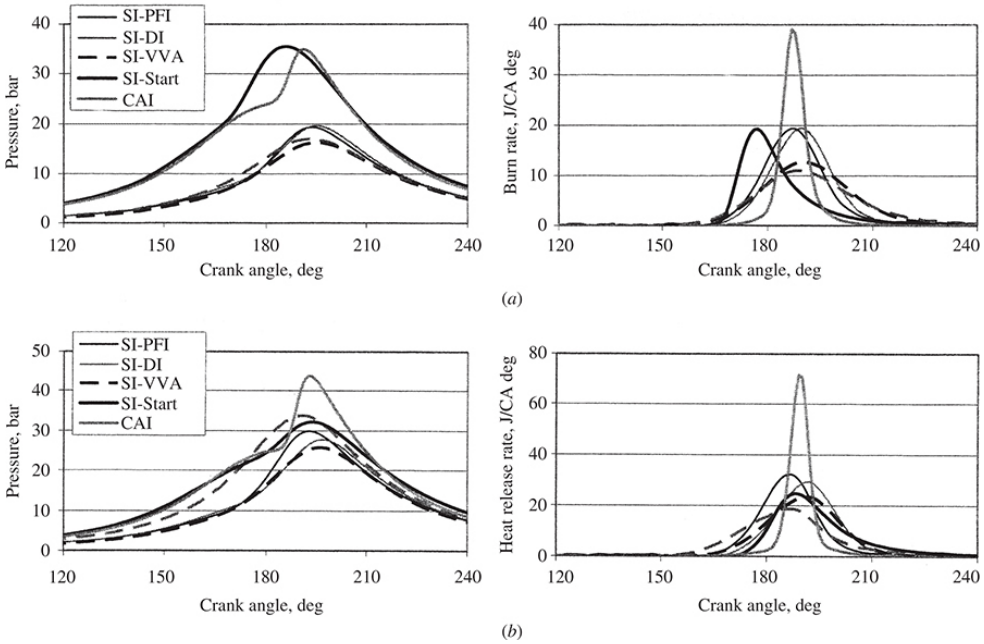


Figure 9.31 Cylinder pressure and burn rate profiles for several different gasoline engine combustion systems, all tested in the same single-cylinder engine: spark-ignited port fuel injection (SI-PFI), spark-ignited direct fuel

injection (SI-DI), spark-ignited variable valve actuation system with high internal EGR (SI-VVA), spark-ignited direct-injection stratified charge (SI-STRAT), and a controlled autoignition concept (CAI) (see Sec. 10.7). For each setup, fuel injection timing and spark timing were adjusted to give close to optimum fuel consumption without compromising combustion stability and emissions. At 2000 rev/min: (*a*) 3 bar nimep; (*b*) 5 bar nimep [41]. See text.

The SI-PFI and SI-DI, both operating with a homogeneous stoichiometric mixture, show closely comparable pressure and burning rate profiles (at the higher load, Fig. 9.31 *b*, the DI combustion profile is slightly retarded relative to MBT timing and thus has a lower maximum pressure). The SI-VVA has a slower, more spread out, burn rate profile due to lower in-cylinder turbulence levels because it operates unthrottled and at lower peak pressure levels (there is less overall unburned mixture compression). [41]

The stratified DISI engine, however, shows a significantly different burning profile that is not symmetric about the maximum burn rate point. The lighter-load condition shows the greater asymmetry. Fuel-vapor/air mixtures burn fastest when close to stoichiometric. Very lean mixtures will not burn at normal engine temperatures and pressures; the laminar flame speeds are much too low (see Fig. 9.25). Thus stratified engine operation requires the creation of a close-enough to stoichiometric fuel-air-residual gas mixture "cloud" in which the spark plug is embedded at the crank angle when the spark discharge initiates combustion. This significantly richer-than-average fuel-air cloud is achieved through use of the appropriate fuel injection technology, injection timing, fuel spray development—atomization, vaporization, and mixing with air, between start of injection and ignition, aided by both intake- and compression-generated in-cylinder air motion and piston-crown and cylinder-head wall shaping (see Sec. 7.7.2). This close-to-stoichiometric "cloud" then burns rapidly as a "premixed" turbulent flame propagates through it. The flame slows down as it approaches the leaner outer regions of the cloud, and then extinguishes. The air (containing some fuel vapor) then mixes with the burned gases produced by combustion of the cloud, completing the burn up of rich burned regions within the cloud and of the fuel that "overmixed" with the excess air during the clouds development, to complete combustion and "homogenize" the burned gases during expansion. At lighter loads, with less fuel relative to the air, this rapid cloud

premixed-flame-burning ends sooner, and the burn rate profile is more asymmetric as the figure indicates.

In summary, stratified-charge DI engine combustion occurs in two stages. Following spark ignition, a partially premixed flame propagates through the predominant mixture region where the fuel-air-burned gas (residual plus EGR) distribution has a high laminar burning velocity. This is followed by the still-rapid burnout of locally rich regions behind the flame front as oxygen from leaner regions and some of the surrounding air mix into the hot post-flame gases. Then a slower mixing-controlled combustion occurs of large fuel-rich pockets, such as that resulting from the last injected fuel, which enters the combustion chamber at lower velocity with poorer atomization, and thus slower fuel-air mixing. At lighter loads, the flame may never fill the combustion chamber. As a consequence, at lighter loads, the engine-out hydrocarbon emissions rise significantly. [40, 42]

The CAI-controlled auto-ignition gasoline engine combustion concept—also shown in Fig. 9.31—is different from the standard spark-ignited premixed flame propagation process, and the diesel engine's diffusion flame (Chap. 10). A premixed lean or dilute fuel-vapor/air/residual mixture is created in-cylinder through early fuel injection with enough high-temperature trapped residual for the mixture temperature when combustion commences to be significantly higher than in a standard spark-ignited engine. At the appropriate time in the engine cycle, spontaneous ignition and combustion occur in an essentially volume burning (not flame propagation) manner. The figure shows a rapid symmetric burn rate profile (which limits this CAI combustion process to lighter engine loads). This controlled autoignited combustion mode is discussed in Sec. 10.7 .

9.4 CYCLIC VARIATIONS IN COMBUSTION, PARTIAL BURNING, AND MISFIRE

9.4.1 Observations and Definitions

Observation of cylinder pressure versus time measurements in a spark-ignition engine, for successive operating cycles, shows that substantial

variations on a cycle-by-cycle basis exist. Since the pressure development is uniquely related to the combustion process, substantial variations in the combustion process on a cycle-by-cycle basis are occurring. In addition to these variations in each individual cylinder, there can be significant differences in the combustion process and pressure development between the cylinders in a multi-cylinder engine. Cyclic variations in the combustion process are caused by variations in mixture motion within the cylinder at the time of spark, cycle-by-cycle, variations in the amounts of air and fuel fed to the cylinder each cycle, and variations in the mixing of fresh mixture and residual gases within the cylinder each cycle which are especially important in the vicinity of the spark plug. Variations between cylinders are caused by differences in these same phenomena, cylinder-to-cylinder.

Cycle-by-cycle variations in the combustion process are important for two reasons. First, since the optimum spark timing is set for the "average" cycle, faster-than-average cycles effectively have over-advanced spark-timing and slower-than-average cycles effectively have retarded timing, so reductions in torque and efficiency result. Second, it is the extremes of the cyclic variations that limit engine operation. At high load, the fastest burning cycles with their over-advanced spark timing are most likely to knock. Thus, the fastest burning cycles determine the engine's fuel octane requirement and limit its compression ratio (see Sec. 9.6.3). The slowest burning cycles, which are effectively retarded relative to optimum timing, are most likely to burn incompletely. Thus these cycles set the practical lean operating limit of the engine or limit the amount of exhaust gas recycle (used for NO emissions control), which the engine will tolerate. Due to cycle-by-cycle variations, the spark timing and average air/fuel ratio must always be compromises, which are not necessarily the optimum for the average cylinder combustion process. Also, variations in cylinder pressure have been shown to correlate with variations in brake torque, which directly relate to vehicle driveability.

An example of the cycle-by-cycle variations in cylinder pressure, and the variations in mixture burning rate that cause them, are shown in Fig. 9.32. Pressure, and gross heat-release rate calculated from the cylinder pressure using Eq. (9.27) , for several successive cycles at a mid-load mid-speed point are shown as a function of crank angle. The maximum heat-release rate and the duration of the heat release or burning process vary by a factor of two from the slowest to the fastest burning cycle shown. The peak cylinder pressure varies accordingly. The faster burning cycles have substantially

higher values of maximum pressure than do the slower burning cycles; with the faster burning cycles, peak pressure occurs earlier, closer to TC.

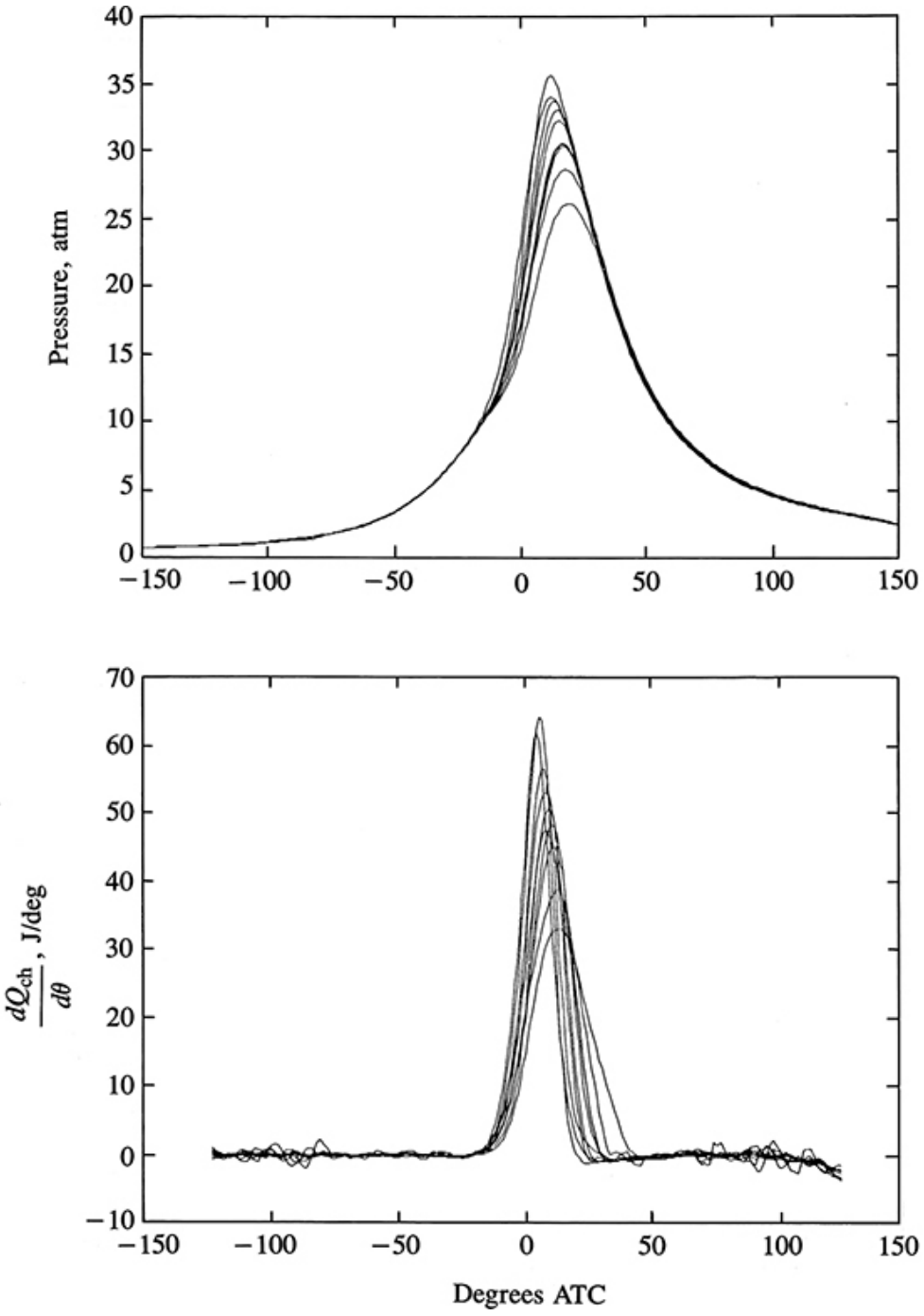


Figure 9.32 Measured cylinder pressure and calculated gross heat-release rate for 10 cycles in single-cylinder spark-ignition engine operating at 1500 rev/min, $\varphi = 1.0$, $p_{inlet} = 0.7$ atm, MBT timing = 25° BTC.[12]

The heat-release rate data in Fig. 9.32 show that there are cycle-by-cycle variations in the early stages of flame development (from zero to a few percent of the total heat release) and in the major portion of the combustion process—the rapid-burning phase—as indicated by the variations in the maximum burning rate.

As the mixture becomes leaner with excess air, or more dilute with a higher burned gas fraction from residual gases or exhaust gas recycle, the magnitude of cycle-by-cycle combustion variations increases. Eventually, some cycles become sufficiently slow burning that combustion is not completed by the time the exhaust opens: a regime where *partial burning* occurs in a fraction of the cycles is encountered. For even leaner or more dilute mixtures, the *misfire* limit is reached. At this point, the mixture in a fraction of the cycles fails to ignite. While spark-ignition engines will continue to operate with a small percentage of the cycles in the partial-burn or misfire regimes, such operation is obviously undesirable from the perspective of efficiency, hydrocarbon emissions, torque variations, and roughness. The partial-burn and misfire regimes are discussed in Sec. 9.4.3 .

Various measures of cycle-by-cycle combustion variability are used. It can be defined in terms of variations in the cylinder pressure between different cycles, or in terms of variations in the details of the burning process, which cause the differences in pressure. The following quantities have been used:

- *Pressure-related parameters.* The maximum cylinder pressure $p_{\max}$; the crank angle at which this maximum pressure occurs $\theta_{p\ max}$; the maximum rate of pressure rise $(dp/d\theta)_{\max}$; the crank angle at which $(dp/d\theta)_{\max}$ occurs; the indicated mean effective pressure [which equals $\int p\, dV/V_d$; see Eqs. (2.14) , (2.15), and (2.19)].
- *Burn-rate-related parameters.* The maximum heat-release rate (net or gross; see Sec. 9.2.2); the maximum mass burning rate; the flame development angle $\Delta\theta_d$, and the rapid burning angle $\Delta\theta_b$; the crank angle at which 50% of the mixture is burned $q_{50\%}$ (see Sec. 9.2.3).
- *Flame front position parameters.* Flame radius, flame front area, enflamed or burned volume, all at given times; flame arrival time at given locations.

Pressure-related quantities are easiest to determine; however, the relation between variations in combustion rate and variations in cylinder pressure is complex. [43] Equation (9.26) defines the factors that govern this relationship. Because the rate of change of pressure is substantially affected by the rate of change of cylinder volume as well as rate of burning, changes in the phasing of the combustion process relative to TC (e.g., which result from changes in flame development angle), as well as changes in the shape and magnitude of the heat-release rate profile, affect the pressure. Figure 9.33 illustrates how the magnitude of the maximum cylinder pressure $p_{\max}$, and the crank angle at which it occurs $\theta_{p\ max}$ vary as the crank angle at which combustion effectively starts (e.g., θ at which 1% of the cylinder mass has burned) and the burning rate is varied. Curve CABE shows how $p_{\max}$ and $\theta_{p\ max}$ vary for a fixed fast-burning heat-release profile (the duration of the heat-release process and its maximum value are held constant), as the phasing of this combustion process relative to TC is varied. A corresponds to MBT timing where the start of combustion is phased to give maximum brake torque, B corresponds to retarded timing, and C to over-advanced timing. C′A′B′D′E′ is a similar curve for a slow-burning heat-release profile. A′corresponds to MBT timing, and B′ and D′ to increasingly retarded timing. Note that with a sufficiently slow-burning heat-release profile—beyond D′— $\theta_{p\ max}$ decreases as the burn process is increasingly retarded. This occurs when the rate of increase of pressure due to combustion becomes so low that it is more than offset after TC by the pressure decrease due to volume increase: eventually for extremely slow and late burning, the maximum pressure approaches the motored pressure at TC. The dashed lines show the constant start-of-combustion timing *relative* to MBT for each burn rate curve. Note that $\theta_{p\ max}$ for constant relative timing varies little as the heat-release profile or burn rate varies. [43]

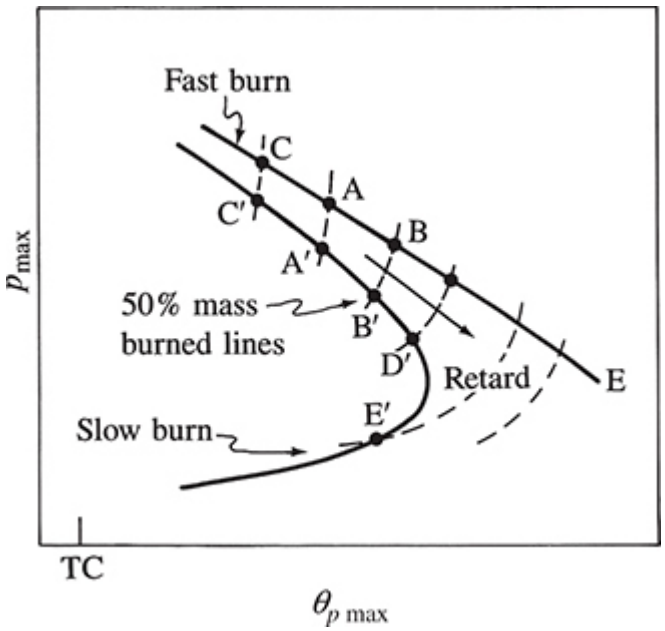


Figure 9.33 Schematic of variation in maximum cylinder pressure and crank angle at which it occurs, in individual cycles. CABE typical of fast heat-release process; C ,A ,B ,D ,E , typical of slow heat-release process. (*From Matekunas.* [43])

We can now explain the effects of variations in the heat-release profile (both in the development stage of the burning process which effectively changes the location of the start of combustion, and in the rate of burning throughout the process) on $p_{\ \max}$ and $\theta_{p\ max}$, when the spark timing occurs at a fixed crank angle. For a fixed burning rate profile (duration of burn and maximum burning rate), as the start of combustion is delayed to be closer to TC, $p_{\ \max}$ decreases and $\theta_{p\ max}$ initially increases (A to B or A′ to B′) . This is the effect of a change in relative timing or phase of the burning process due to a slower initial rate of flame development with fixed spark timing. If, in addition to flame development being slower the heat-release rate throughout the burning process is lower, then that combustion process is even more retarded from the optimum and $p_{\ \max}$ decreases further, and $\theta_{p\ max}$ increases further, to their values at D′ . The effect of a faster initial flame development and faster burning rate, with fixed spark timing, is the opposite. The magnitudes of the changes in $p_{\ \max}$ and $\theta_{p\ max}$ depend, obviously, on the extent of the cyclic variations; they also depend on whether the average burn

process is fast or slow. For fast-burning engines, a larger fraction of the heat release occurs near TC when the chamber volume is changing relatively slowly. Thus pressure variations are mainly due to combustion variations. With slow-burning engines, where a significant fraction of the energy release occurs well after TC, the effect of volume change also becomes significant and augments the effect of combustion variations. For large variations and a slow average burning process, $p_{\max}$ can fall below E′, and $\theta_{p\ max}$ then decreases. A fast-burning combustion process significantly reduces the magnitude of cycle-by-cycle combustion variations and thus their impact on engine performance. [15, 44, 45]

We can now evaluate the various measures of combustion variability. The maximum pressure variation has been shown to depend on both changes in phasing and burning rate. The magnitude of this variation depends on whether the combustion chamber is faster or slower burning, on average. It also depends on whether the burning process is substantially retarded relative to MBT. It depends, too, on cyclic cylinder fuel and air charging variations. Thus the interpretation of variations in $p_{\max}$ [or in the maximum rate of pressure rise $(dp/d\theta)_{\max}$] in terms of variations in the rate and phasing of the burning process must be done with care. The location of maximum pressure $\theta_{p\ max}$ also depends on relative phasing of combustion and on the burn rate profile. In addition, for slow-burning chambers and retarded timing (around E′) variations produce little change in $\theta_{p\ max}$. However, for fast-burning chambers, with MBT or only slightly retarded timing, the location of peak pressure depends essentially on the phasing of each combustion process relative to its MBT phasing, and is independent of charging variations. For these reasons, $\theta_{p\ max}$ is a useful measure of variability in combustion event phasing. [25]

One important measure of cyclic variability, derived from pressure data, is the *coefficient of variation* in *indicated mean effective pressure.* It is the standard deviation in imep divided by the mean imep, and is usually expressed in percent:

$$\mathrm{COV}_{\mathrm{imep}}(\%) = \frac{\sigma_{\mathrm{imep}}}{\mathrm{imep}} \times 100 \tag{9.59}$$

It defines the cyclic variability in indicated work per cycle (on a gross or net basis), and it has been found that vehicle driveability problems usually result when COV $_{imep}$ exceeds about 2 to 5%, depending on engine operating conditions.

Figure 9.34 illustrates the relationships between p_{max}, $\theta_{p\ max}$, and imep for 120 cycles of an engine cylinder at fixed operating conditions and three different spark timings.[43] The MBT timing data show a spread in imep at a fixed value of $\theta_{p\ max}$. This imep data band is relatively flat and is centered around $\theta_{p\ max} \approx 15°$; only at later values of $\theta_{p\ max}$ does imep fall off significantly. The vertical spread in imep around $\theta_{p\ max} = 15°$ is due to variations in the amount of fuel entering the cylinder each cycle; normal variations in the burn profile under these conditions, which effectively change the phasing of the combustion process, produce only modest reductions in imep. For early $\theta_{p\ max}$ (the extreme upper left of Fig. 9.34 *a*), the variations in p_{max} are also due mainly to these fuel-charging variations, cycle-by-cycle: these are the fastest burning cycles with the most advanced phasing. As $\theta_{p\ max}$ increases, the dispersion increases as cyclic variations in phasing and burning rate have increasing impact.

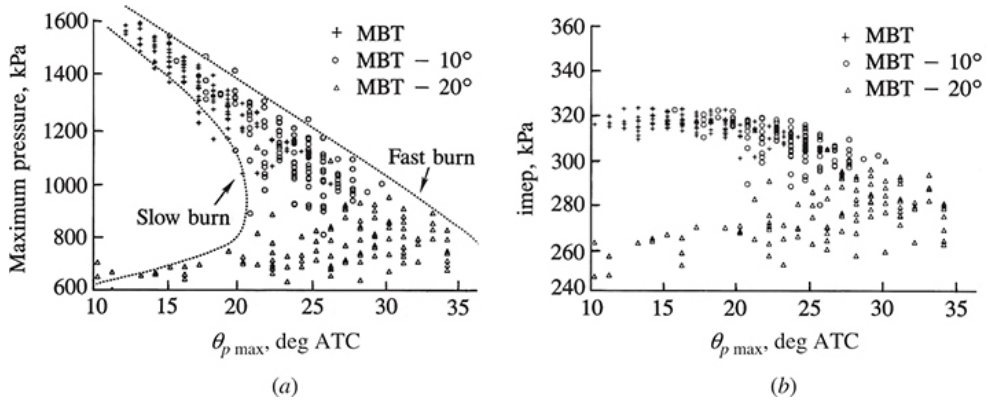


Figure 9.34 (*a*) Individual-cycle maximum pressure versus crank angle at which p_{max}, occurs. (*b*) Individual-cycle indicated mean effective pressure versus $\theta_{p\ max}$.[43]

The shapes of the frequency distributions in individual-cycle pressure data [e.g., in p_{max}, (dQ/dq) $_{max}$, $\theta_{p\ max}$] and in burn rate data such as $\Delta\theta_d$,

$\Delta\theta_b$, $(dQ/dq)_{max}$ depend on whether the combustion process is fast and "robust" (e.g., with close-to-stoichiometric mixtures at higher loads at optimum timing—well away from the lean operating limit of the engine) or slower and less repeatable, closer to the lean or dilute-mixture operating limit. Under robust combustion conditions these distributions are close to normal distributions. [15] When the combustion process is much slower, the cyclic variability becomes large and the distribution becomes skewed toward the slower burning cycles which have low imep (due to the substantial retard of these slower cycles). Examples of the net imep and 10 to 90% burn angles, which illustrate these frequency distribution trends, are shown in Fig. 9.35. A standard gasoline engine was operated at a typical light load, 3.5 bar net imep at 1500 rev/min. With a stoichiometric mixture and fast burn, the COV_{imep} is small (0.6%) and the distributions in imep n and $\Delta\theta_{10-90}$ are narrow and symmetric. As the relative air/fuel ratio increases from 1.0 through 1.4 to 1.7, the burn rate in these increasingly leaner mixture decreases, and the spread in both imep n and $\Delta\theta_{10-90}$ becomes larger and, then asymmetric. COV_{imep} increases from its small value at stoichiometric (0.6%), to about 1.0%, to close to 4%. With the leanest mixture shown, some cycles become very slow burning and the net imep decreases significantly as a consequence. Slow-burning cycles are effectively retarded, and much of the mixture burning takes place in an ever expanding combustion chamber volume, which slows the burning process even more since unburned mixture pressures and temperatures are decreasing. [15, 46]

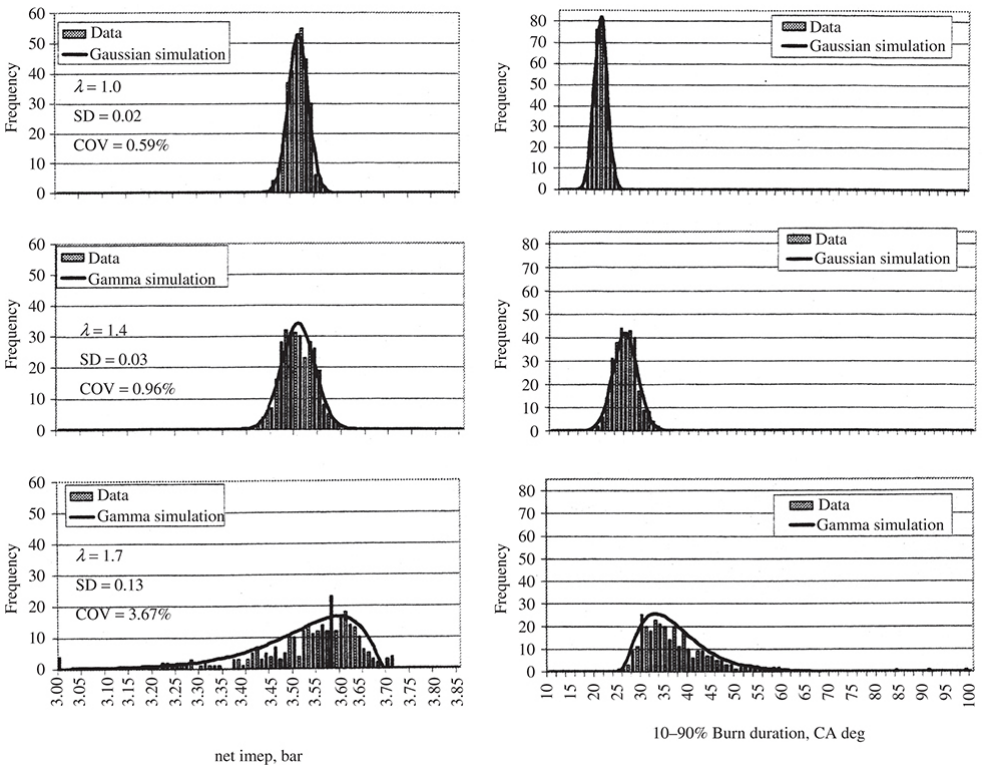


Figure 9.35 Frequency distributions of net imep and 10 to 90% burn durations for different relative air/fuel ratios (λ = 1.0, 1.4, 1.7): MBT spark timing, engine operating at 1500 rev/min, net imep = 3.5 bar with indolene, at r_c = 9.8. SD, standard deviation; COV, coefficient of variation. [17]

Cylinder pressure data are often averaged over many cycles to obtain the mean cylinder pressure at each crank angle. The primary use of this average pressure versus crank-angle data is in calculating the average indicated mean effective pressure (which is a linear function of p). Since combustion parameters are not linearly related to the cylinder pressure [see Eq. (9.27)], analysis of the *average* pressure data will not necessarily yield accurate values of average combustion parameters. The error will be most significant when the combustion variability is largest. It is best to determine mean combustion parameters by averaging their values obtained from a substantial number of individual cycle analysis results. The number of cycles, which must be averaged to obtain the desired accuracy depends on the extent of the combustion variability. For example, while 40 to 100 cycles may define imep to within a few percent when combustion is highly repeatable, several

hundred cycles of data may be required when cyclic combustion variations are large. [10]

9.4.2 Causes of Cycle-by-Cycle and Cylinder-to-Cylinder Variations

Cycle-by-cycle combustion variations are evident from the beginning of the combustion process. Analysis of flame photographs from many engine cycles taken in special research engines with windows in the combustion chamber has shown that dispersion in the fraction of the combustion chamber volume inflamed is present from the start of combustion (e.g., see Refs. 3, 29, and 37). Dispersion in burning rate is also evident throughout the combustion process (see Figs. 9.4 and 9.32). Three factors have been found to influence this dispersion:

1. The variation in gas motion in the cylinder during combustion, cycle-by-cycle;
2. The variation in the amounts of fuel, air, residual, and recycled exhaust gas supplied to a given cylinder each cycle;
3. Variations in mixture composition within the cylinder each cycle—especially near the spark plug—due to variations in mixing between air, fuel, residual gas and recycled exhaust gas.

The relative importance of these factors is not yet fully defined, and depends on engine design and operating variables. Table 9.4 provides a framework for discussing how the detailed components of these three overall factors lead to variations in the three stages of the SI engine combustion process: early flame development, flame propagation, and the final (late) stage of burning.

TABLE 9.4 Factors affecting SI engine combustion

Phases of combustion:

1. Early flame development
2. Flame propagation
3. Late stage of burning

Factors affecting SI engine cycle-to-cycle variations:

a. Spark energy deposition into gas (1)
b. Flame kernel motion (1)
c. Heat losses from kernel to spark plug (1)
d. Local turbulence characteristics near plug (1)
e. Local mixture composition near plug (1)
f. Overall charge components – air, fuel, residual, EGR (2, 3)
g. Average turbulence in the combustion chamber (2, 3)
h. Large scale features of the in-cylinder flow (3)
i. Flame geometry interaction with the combustion chamber walls (3)

The variation in the velocity field within the engine cylinder throughout the cycle, and from one cycle to the next, has been reviewed in Sec. 8.2.2 . Toward the end of the compression stroke, the ensemble-averaged rms velocity fluctuation is of comparable magnitude to the mean piston speed, and may be larger than the mean flow velocity if there is no strongly directed local mean flow pattern (see Figs. 8.8 and 8.9). This ensemble-averaged velocity fluctuation combines both cycle-by-cycle variation in the mean flow and the turbulent velocity fluctuations. During compression, these two components are of comparable magnitude (see Fig. 8.9). The available database indicates that substantial variations in the mean flow exist, cycle-by-cycle, both in the vicinity of the spark plug *and* throughout the combustion chamber. Velocity variations contribute to variations in the initial motion of the flame center as it grows from the kernel established by the spark, and in the initial growth rate of the flame: they can also affect the burning rate once the flame has developed to fill a substantial fraction of the combustion chamber. Variations in gas motion near the spark plug convect the flame in its early stages in different directions and at different velocities, cycle-by-cycle. This affects the flame's interaction with the cylinder walls, changing the flame area development with time. Variations in the turbulent velocity

fluctuations near the spark plug will result in variations in the rate at which the small initially laminar-like flame kernel develops into a turbulent flame. Variations in the mean flow throughout the chamber will produce differences in flame front shape: also, they may produce differences in turbulence, which affect the propagation velocity of the front.

It is well known that the fuel, air, and recycled exhaust gas flows into each cylinder of a multi-cylinder engine are not identical. These flow rate differences are typically a few percent (see Secs. 7.6.2 and 7.6.3). It is also known that the flow patterns within the different cylinders are not necessarily identical due to differences between the individual intake manifold runner and port geometries in many production engines. All these factors contribute to cylinder-to-cylinder variations in the combustion process: there can be significant differences in the mean burn rate parameters as well as in the cyclic variations in these parameters. Also, the limited data available on the variation in mixture composition within each cylinder for each cycle indicate that cyclic charging variations in individual cylinders are comparable in magnitude to cylinder-to-cylinder differences (i.e., a few percent). Also, the amount of residual gas left in the cylinder may vary somewhat, cycle-by-cycle. At higher loads, where the combustion process is more repeatable (and always completed relatively early in the expansion stroke), and the residual gas fraction is smaller (see Sec. 6.4), variations in the total amount of residual are not expected to be significant. At light loads (particularly at idle), where combustion variability is much higher, variations in the residual gas mass and its composition may become important.

In addition, the in-cylinder mixing of fuel, air, recycled exhaust, and residual is not complete: nonuniformities in composition exist within the cylinder at the start of combustion. Composition variations, cycle-by-cycle, in the vicinity of the spark plug electrode gap will affect the early stages of flame development, especially as the flame grows through the laminar-like burning phase following the creation of a small flame kernel by the spark discharge (see Sec. 9.5.1). The standard deviation of the cycle-by-cycle air/fuel ratio fluctuations can be a few percent of the mean (A/F). Significant fluctuations, cycle-by-cycle, in the mixing of fresh mixture with the in-cylinder residual gas have been noted. Nonuniformities in EGR distribution between cylinders and in EGR mixing within the cylinder can also occur and increase the variations in burned gas fraction locally at the spark gap, cycle-by-cycle. [48]

Studies in a multi-cylinder production SI engine, where the fuel/air ratio nonuniformities and the nonuniform mixing of fresh mixture with residual gas were removed in turn as contributors to cycle-by-cycle variations (by comparing premixed propane operation with carbureted operation with gasoline, and by removing residual gas by purging with nonfiring cycles), indicate that the three contributing factors to cyclic combustion variations—velocity variations, fuel/air ratio variations, and residual gas mixing variations—are of comparable importance under light-load operating conditions.

This explanation of the underlying causes for cycle-by-cycle variations is consistent with the description of the turbulent flame propagation process in Sec. 9.3.4 . Conditions in the *vicinity* of the spark plug will influence the initial stages of the flame propagation process—establishing a stable kernel and its development into a turbulent flame. During the developed flame propagation phase, the *average* conditions in the bulk gas within the combustion chamber will be the determining factors since the flame front spans the chamber, effectively averaging out local nonuniformities. By conditions are meant the turbulent velocity fluctuations and length scales in the flow, proportions of fuel, air, and burned gas in the mixture, and the mixture state.

The flame development angle $\Delta\theta_d$ (the time to burn a few large eddies and establish a developed turbulent flame) can be expressed as [16]

$$\Delta\theta_d = C\left(\frac{l_i}{u'}\right)^{1/3}\left(\frac{l_M}{S_L}\right)^{2/3} \tag{9.60}$$

The turbulent flow field influences $\Delta\theta_d$ through l_I , u', and l_M, the integral scale, the turbulence intensity, and the microscale, respectively. The mixture composition influences $\Delta\theta_d$ through the laminar flame speed S_L. There is, therefore, a variability in the flame development period, since all of these quantities can vary in the vicinity of the spark plug on a cycle-by-cycle basis.

The pressure development during the rapid-burning developed turbulent flame propagation phase, when the flame spans the combustion chamber, depends on the average rate of burning in the flame. Thus, variations in the turbulent flow field and mixture composition across the gas entering the flame front are averaged out and are not important. However, variation in

chamber-average quantities are significant. The following expression for the maximum burning rate:

$$\left(\frac{dm_b}{dt}\right)_{\max} = \frac{Cm_f(h^*/B)(\rho_u^*/\rho_i)^{10/9}[(\overline{u}'^* S_L^*)/h_i]}{\nu^{*1/3}} \tag{9.61}$$

Here, m_f is the mass of fuel in the chamber, h is the instantaneous (mean) clearance height, B is the bore, r the density, $\overline{u}'$ the average turbulence intensity across the flame front, and n the kinematic viscosity: the * denotes the value at the time of the maximum burning rate and the subscript i the value at spark; C is a constant depending on engine design. This equation indicates that cycle-by-cycle variations in the maximum burning rate can result from variations in the overall flow pattern within the combustion chamber (which vary $\overline{u}^*$) and from variations in the amount of fuel (m_f) that enters the cylinder each cycle, [g] Also, it can be seen that variations in the flame development process will result in variations in the maximum burning rate because the crank angle at which the maximum burning rate occurs is shifted, and all the starred parameters in Eq. (9.59) will have different values.

From the discussion of flame development and structure in Sec. 9.3 , and Eqs. (9.58) and (9.59) above, we would expect that mixture conditions and motion leading to slower flame development rates (longer flame development angles, $\Delta\theta_d$)—lower turbulence intensities and more dilute mixtures—would also give lower burning rates (longer rapid burning angles, $\Delta\theta_b$). Data from many different engines and a wide range of operating cases show that this is the case. The standard deviation of $\Delta\ \theta_b$ for a given chamber and operating condition usually increases as the average burning process becomes slower.

One final factor of importance is how variations in flame development and burning rate affect engine torque. With fixed spark timing, such variations in the combustion process cycle-by-cycle result in slower flame developing and burning cycles being retarded and faster developing and burning cycles being over-advanced. The curve of torque versus combustion timing (relative to optimum timing, Fig. 9.16) is almost independent of the burning rate; that is, a given magnitude retard (of say 10° CA) relative to optimum timing gives almost the same reduction in torque for a fast burn as it does for a slow burn. This is because the burning process, for optimum timing, is centered at about

8 to 10° ATC independent of the burn rate, and retard or advance shifts this "center" by equal amounts for all burn rates. One of the major advantages of fast-burning SI engines is now apparent. The magnitude of the variations in the flame development process and subsequent flame propagation rate are decreased as the burning rate is increased. Thus, these smaller combustion variations in fast-burn engines, which correspond to modest retard and over advance in non-average burn rate cycles, have little effect on torque. In contrast, the larger combustion variations of slow-burning engines result in more significant cyclic torque variations.

9.4.3 Partial Burning, Misfire, and Engine Stability

As the unburned mixture in a spark-ignition engine is leaned out with excess air or is diluted with increasing amounts of burned residual gas and exhaust gas recycle, the flame development period, the duration of the rapid burning phase, and the cycle-by-cycle fluctuations in the combustion process all increase. Eventually a point is reached where engine operation becomes rough and unstable, and hydrocarbon emissions increase rapidly. The point at which these phenomena occur effectively defines the engine's *stable operating limit*. [h] These phenomena result from the lengthening of all stages of the combustion process as the unburned mixture is diluted. With increasing dilution, first a fraction of the cycles burns so slowly that combustion is only just completed prior to exhaust valve opening. Then as burning times lengthen further, in some cycles there is insufficient time to complete combustion within the cylinder: also, flame extinguishment before the exhaust valve opens and before the flame has propagated across the chamber may start to occur in some cycles. Finally, misfiring cycles where the mixture never ignites may start to occur. The proportion of partial burning or non-burning cycles increases rapidly if the mixture is made even more lean or dilute: the point is eventually reached where the engine will not run at all.

The impacts on engine stability of increasing combustion variability, due to increasingly lean (excess air) operation and/or higher amounts of exhaust gas recycle, at part-load, are shown in Figs. 9.35 and 9.36. Figure 9.36 *a* shows the distributions of individual-cycle indicated mean effective pressure values for 0, 20, and 28% EGR. Without EGR at these conditions, the spread

in imep is narrow. Increasing EGR widens the distribution significantly and cycles with low imep, and eventually zero imep, occur. Figure 9.36 *b* shows how the COV of imep and hydrocarbon emissions increase as EGR is increased. Slow burn, then partial burn, and then misfire cycles occur with increasing frequency. In the slow-burn cycles, combustion is complete but ends after 80° ATC and the indicated mean effective pressure is low (between 85 and 46% of the normal-burn value). Imep in partial-burn cycles was less than 46% of this value. In misfiring cycles, imep < 0. [48]

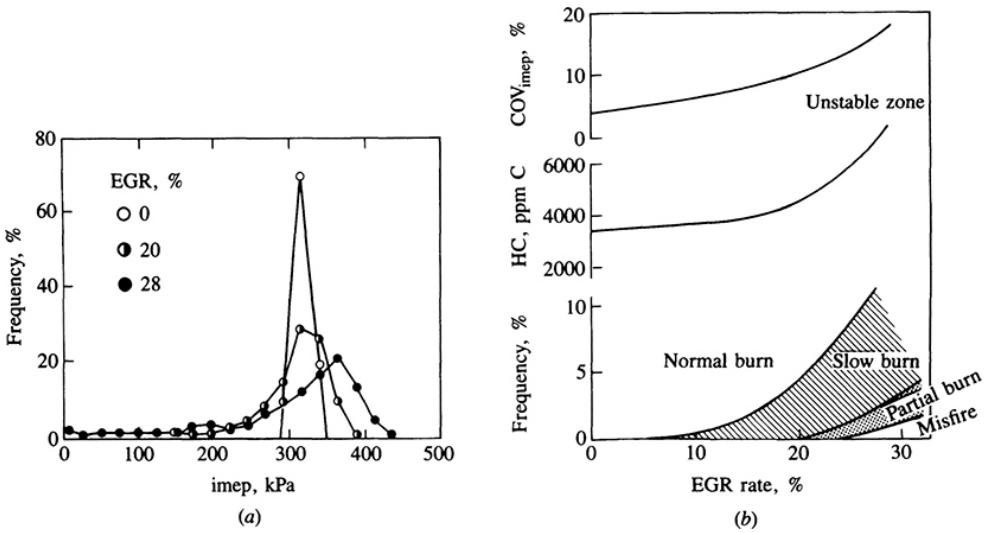


Figure 9.36 (*a*) Frequency distributions in indicated mean effective pressure at different EGR rates for the engine studied; 0% EGR gave good engine stability, 10% acceptable stability, and above 20% poor stability. (*b*) Coefficient of variation in imep, HC emissions, and percentage of normal, slow, partial-burn, and misfire cycles. Engine conditions: 1400 rev/min, φ = 1.0, MBT timing, imep = 324 kPa. [48]

An explanation of the combustion phenomenon that occurs at an engine's stable operating limit has been developed by Quader. [50] It involves the following terms:

Ignition-limited spark timing or the **ignition limit**. The spark timing (advanced from MBT timing) at which *misfire* (i.e., failure of flame initiation) first occurs at a given mixture composition in a given small but arbitrary fraction of cycles (e.g., 0.5 to 1%).

Partial-burn-limited spark timing or the **partial-burn limit**. The spark timing (retarded from MBT) at which incomplete flame propagation occurs at a given mixture composition in a given small percent of the cycles (again, this frequency is selected arbitrarily for experimental convenience).

Lean or dilute misfire limit at MBT spark . The leanest mixture stoichiometry at which the engine could operate stably at MBT spark timing with a misfire frequency below a specified value (again, this frequency, usually a percent or less of the cycles, is selected arbitrarily for convenience). A *dilute misfire limit*, the maximum amount of exhaust gas recycle that can be absorbed at a given stoichiometry for stable engine operation, can be similarly defined.

Engine experiments have defined the locations of the ignition limit line and the partial-burn limit line; they are shown schematically in Fig. 9.37. At a given spark timing, on this spark timing versus fuel/air equivalence ratio plot, progressive leaning of the mixture fed to the engine will lead to the onset of misfire or to the onset of partial burning, depending on the location of the lines and the spark timing selected. The individual figures show the possible interactions of the MBT timing line—leaner mixtures require greater advance—with the ignition limit and partial-burn limit lines. At MBT timing, the partial-burn limit may or may not be reached prior to misfire or the ignition limit. It will depend on the engine and ignition system design and operating conditions. For retarded spark timings relative to MBT, partial burning and not failure of flame initiation is the primary cause of unstable engine operation.

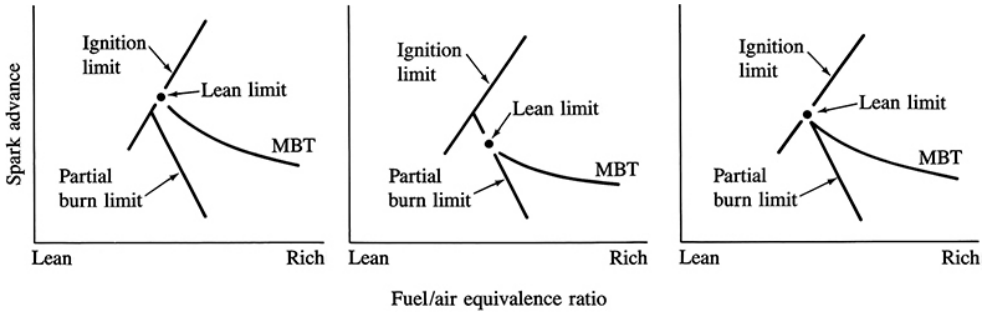


Figure 9.37 Schematics of three possible combinations of ignition limit, partial-burn limit, and MBT timing curves as a function of fuel/air

equivalence ratio. (*From Quader.* [49, 50])

From the above it is clear that flame initiation is a necessary but not sufficient condition for complete combustion. Too slow flame development and propagation following successful ignition is usually the factor which limits engine operation with dilute mixtures. Experiments have shown that a limited crank-angle interval, of some 75 degrees (depending on engine geometry and spark plug location) is available during the engine cycle when conditions are favorable for the complete flame propagation process to be completed. Outside of this crank-angle interval, the mixture pressure and temperature, and the turbulence intensity, are too low to sustain a sufficiently rapid rate of combustion.

9.5 SPARK IGNITION

In spark-ignition engines, the electrical discharge produced between the spark plug electrodes by the ignition system starts the combustion process close to the end of the compression stroke. The high-temperature plasma kernel created by the spark develops into a self-sustaining and propagating flame front—a thin reaction sheet where the exothermic combustion chemical reactions occur. The function of the ignition system is to initiate this flame propagation process, in a repeatable manner, cycle-by-cycle, over the full load and speed range of the engine at the appropriate point in the engine cycle. Shadowgraph and schlieren photographs of the kernel created by the discharge between the plug electrodes, the growth of that kernel, and its transition to a propagating flame have already been presented in Figs. 9.24*a* and *b*, and described in the accompanying text. A spark will arc from one plug electrode to the other when a sufficiently high voltage is applied between them. Ignition systems commonly used to provide this spark are: battery ignition systems where the high voltage is obtained with an ignition coil (coil ignition systems); battery systems where the spark energy is stored in a capacitor and transferred as a high-voltage pulse to the spark plug by means of a special transformer, capacitive-discharge ignition (CDI) systems; and magneto ignition systems where the magneto—a rotating magnet or armature—generates the current used to produce a high-voltage pulse. The

dominant system in automotive applications is the Transistorized Coil Ignition (TCI) system. The electrical energy is stored in the inductance of a coil and is released over about 2 ms when the spark discharge is required. Figure 9.38 shows the basic coil ignition system layout for a system with individual coil and plug arrangement. Before ignition is desired, the ignition driver stage switches a current from the electrical system through to the primary winding of the ignition coil. While the primary current circuit is closed (the dwell period) this current builds up a magnetic field in the coil. At the time of ignition, the primary winding current is interrupted producing a high voltage across the coil's secondary winding as the magnetic-field energy is discharged. This voltage (if high enough) causes the gas between the spark plug electrodes to break down and thus produces the spark discharge discharge. [51]

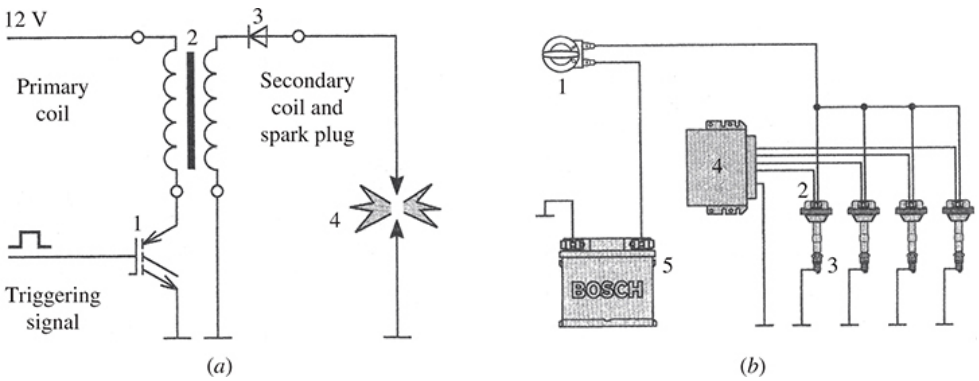


Figure 9.38 Example of ignition system with single spark coils. (*a*) Circuit layout: (1) ignition driver stage, (2) ignition coil, (3) activation arc diode (suppresses the activation spark), (4) spark plug. (*b*) System layout: (1) ignition lock, (2) ignition coils, (3) spark plugs, (4) control unit, (5) battery. (*Courtesy Robert Bosch GmbH and SAE.* [51])

The next section reviews our basic understanding of electrical discharges in inflammable gas mixtures relevant to engine ignition (Sec. 9.5.1), the operating characteristics of conventional engine ignition systems (Sec. 9.5.2), and briefly, some alternative approaches to generating a propagating flame (Sec. 9.5.3).

9.5.1 Ignition Fundamentals

A discharge can arc from one spark-plug electrode to the other only if a sufficiently high voltage is applied. In a typical spark discharge, the electrical potential across the electrode gap is increased until breakdown of the intervening mixture occurs. Electrons, accelerated across the gap by the high-applied electric field, rapidly excite and ionize gas molecules. These ionizing "streamers" then propagate from one electrode to the other. The impedance of the gap decreases drastically when a streamer reaches the opposite electrode and the current through the gap increases rapidly. This stage of the discharge is called the *breakdown phase.* It is followed by the *arc phase*, where the thin cylindrical plasma arc expands largely due to heat conduction and diffusion and, with inflammable mixtures, initiates the exothermic reactions which lead to a propagating flame kernel (Fig. 9.24*a*). This is usually followed by a *glow discharge phase* where, depending on the details of the ignition system, the energy storage device (e.g., the ignition coil) will dump its energy into the discharge circuit. [52]

Figure 9.39 shows the behavior of the spark-discharge voltage and current as a function of time for a TCI system. Typical values of discharge voltage V, current I, are shown; actual values depend on the details of the electrical components. The breakdown phase is characterized by a high-voltage (~10 kV), high-peak current (~ 200 A), and an extremely short duration (≤10 ns). A narrow (~ 40 μm diameter) cylindrical ionized gas channel is established very early. The electrical energy supplied is transferred almost without loss to this plasma column. The temperature and pressure in the column rise very rapidly to values up to about 60,000 K and a few hundred atmospheres, respectively. A strong shock or blast wave propagates outward, the channel expands and, as a result, the plasma temperature and pressure fall. Some 30% of the plasma energy is carried away by the shock wave; however, most of this is regained since spherical blast waves transfer most of their energy to the gas within a small (~ 2 mm diameter) sphere into which the breakdown plasma soon expands. [52]

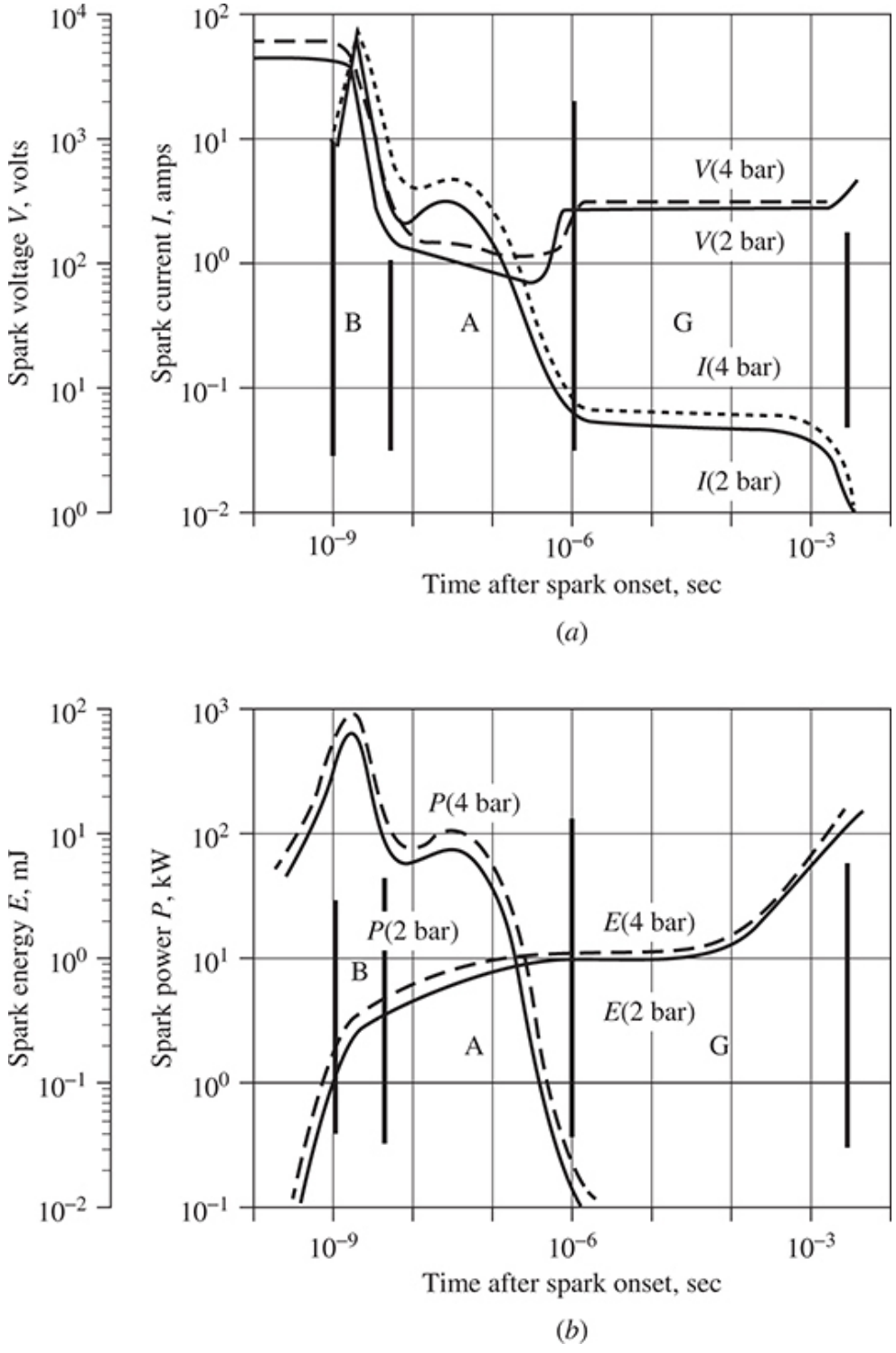


Figure 9.39 (*a*) Voltage and current variation with time and (*b*) power and energy variation with time for a commercial transistorized coil spark-ignition system, in air at 300 K supplying 30 mJ to a 1-mm spark plug gap. The pressure of 4 bar roughly corresponds to engine conditions at spark onset. [23]

Note the three phases of the discharge—breakdown (B), arc (A), and glow discharge (G).

A breakdown phase always precedes arc and glow discharges: it creates the electrically conductive path between the electrodes. The arc phase voltage is low (<100 V), though the current can be as high as the external circuit permits. In contrast to the breakdown phase where the gas in the channel is fully dissociated and ionized, in the arc phase the degree of dissociation may still be high at the center of the discharge but the degree of ionization is much lower (about 1%). Voltage drops at the cathode and anode electrodes are a significant fraction of the arc voltage, and the energy deposited in these electrode sheath regions, which is conducted away by the metal electrodes, is a substantial fraction of the total arc energy (see Table 9.4). The arc requires a hot cathode spot, so heating and evaporation of the cathode material occur. The arc increases in size due primarily to heat conduction and mass diffusion. Due to these energy transfers the gas temperature in the arc is limited to about 5000 K: the temperature and degree of dissociation decrease rapidly with increasing distance from the arc axis. Currents less than 200 mA, a large electrode voltage drop at the cathode (300 to 500 V), a cold cathode, and less than 0.01% ionization are typical for the glow discharge. Energy losses are higher than in the arc phase, and peak equilibrium gas temperatures are about 3000 K. [52]

About 0.2 mJ of energy is required to ignite a quiescent stoichiometric fuel-air mixture at normal engine conditions by means of a spark. For substantially leaner and richer mixtures, and where the mixture flows past the electrodes, an order of magnitude greater energy may be required. TCI systems deliver from 30 to 100 mJ of electrical energy to the spark, with most of this in the glow discharge phase. Due to the physical characteristics of the discharge modes discussed above, only a fraction of the energy supplied to the spark gap is transmitted to the gas mixture. The energy balance for the breakdown, arc, and glow phases of the discharge is given in Table 9.5. Radiation losses are small throughout. The end of the breakdown phase occurs as a hot cathode spot develops, turning the discharge into an arc; heat losses to the electrodes then become substantial. The breakdown phase reaches the highest power level (~ 1 MW), but the energy supplied is small (<1 mJ). The glow discharge has the lowest power level (~ 10 W) but

the highest energy (30 to 100 mJ), due to its long discharge time. The arc phase lies between (see Fig. 9.39). [23] The different energy transfer capabilities for breakdown, arc, and glow discharges arise primarily from the differences in heat losses to the electrodes, as explained above. These losses increase with increasing supplied energy. In arc and glow discharges, increases in either discharge time or discharge current (or in both) always lead to substantial decreases in energy-transfer efficiency. If the glow discharge current is increased above about 100 mA, the discharge changes to the arc mode and heavy electrode erosion will result. Thus there are practical limits to the arc and glow discharge currents; also, the time available for ignition in the engine limits increases in discharge time.

TABLE 9.5 Energy distribution for breakdown, arc, and glow discharges †

	Breakdown, %	Arc, %	Glow, %
Radiation loss	<1	5	<1
Heat loss to electrodes	5	45	70
Total losses	6	50	70
Plasma energy	94	50	30

† Typical values, under idealized conditions with small electrodes.
Source: Maly and Vogel.[52]

Figure 9.40 shows shadowgraph photos of the spark-generated flame kernel, which illustrate the context for this discussion of the engine spark discharge. The first photo (on the left) shows that by the time the glow discharge commences, the breakdown and arc dischargers have already initiated a 1- to 2-mm diameter laminar-like flame kernel. This then steadily grows in size as the flame propagates outward into the unburned mixture in the vicinity of the electrodes. The photos indicate that heat losses from hot flame kernel to the plug electrodes will be significant.

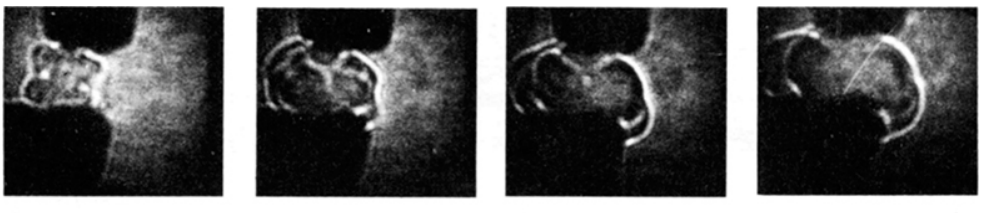


Figure 9.40 Shadowgraph photographs of spark-generated kernel. First photograph on left shows the kernel between the two spark-plug electrodes (dark in photo), about 35 μs after breakdown; 40 μs between photos. Stoichiometric mixture, 1100 rev/min. (*Courtesy A. Douad, Institut Francais du Petrole.*)

The initial expansion velocities in the discharge are much higher than those in self-propagating flames. Figure 9.41 shows the expansion velocities and diameters of the volumes activated by a 3-mJ 100-μs discharge from a CDI system as a function of time after spark onset. The curves shown are (1) for the shock wave following a discharge in air at 1 atm; (2) the growth in plasma region in air at 1 atm; (3) the electrical and chemical plasma region following a discharge in a stoichiometric methane-air mixture at 1 atm. The initially strong shock wave attenuates rapidly to the local sound speed. Up to times of order microseconds, the effects of fuel combustion chemistry are small. The change in slope of the velocity curves for the plasma at about 100 μs indicates the transition from an expansion caused by the initial high pressure in the breakdown discharge, to expansion resulting from heat conduction and diffusion.

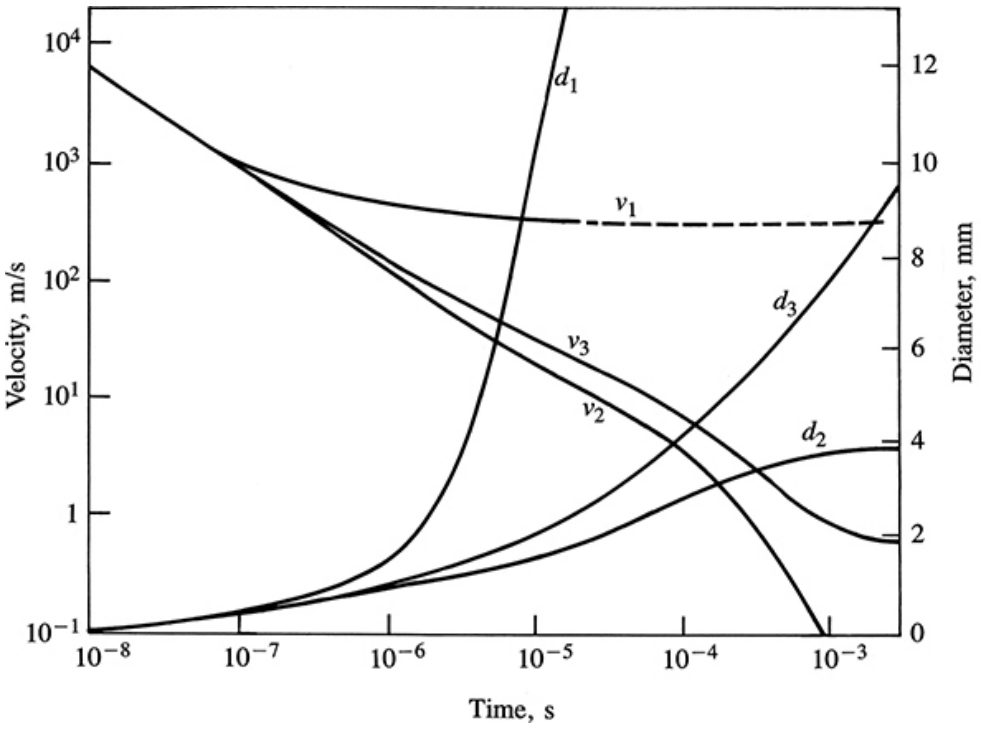


Figure 9.41 Diameters (d) and expansion velocities (v) of volumes activated by a capacitive-discharge ignition system: 3 mJ electrical energy, 100 μs duration. Subscripts denote: 1, shock wave in air at 1 atm; 2, plasma in air at 1 atm; 3, electrical and chemical plasma in stoichiometric methane-air mixture at 1 atm. [52]

The temperature distributions within the three different types of discharge provide additional insight: see Fig. 9.42. During the breakdown discharge, on a time scale of 10 nanoseconds, the temperature rises to 60,000 K. Increasing the breakdown energy does not produce higher kernel temperatures; instead the channel diameter increases producing a larger plasma volume. The kernel temperatures then decrease to the order of 10,000 K on a microsecond time scale as the plasma region expands behind the shock wave. Arc and glow discharges, because their power inputs are much lower, do not increase the kernel temperatures; rather, they extend the cooling period on a microsecond and millisecond time scale, respectively. [23]

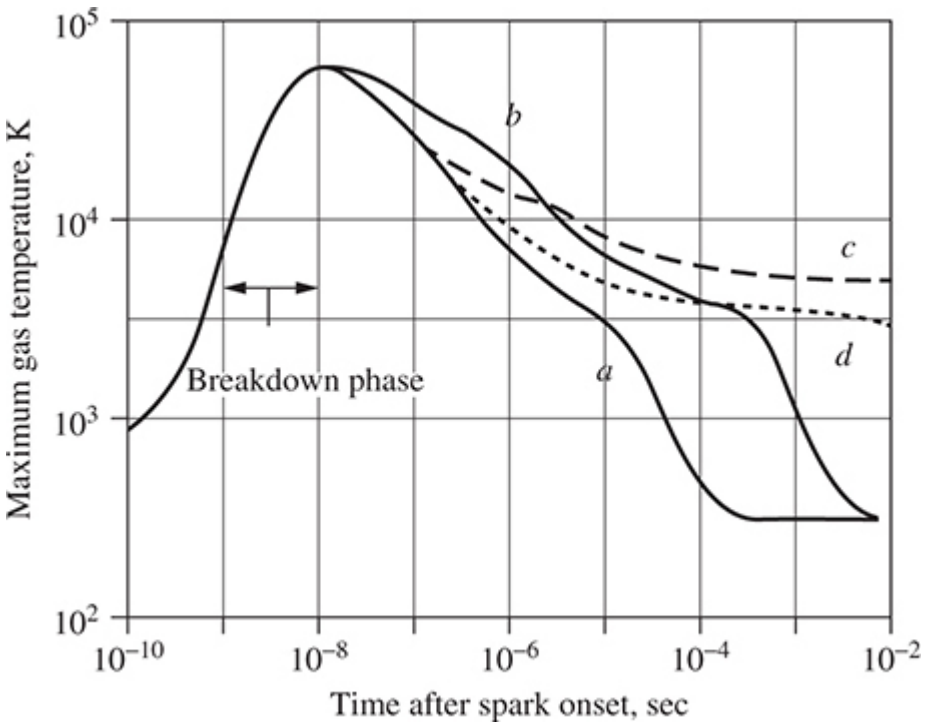


Figure 9.42 Maximum measured gas temperatures as a function of time in different discharge modes and ignition systems: (a) CDI, 3 mJ, 100 μs; (b) breakdown, 30 mJ, 60 ns; (c) CDI plus superposed current (2 A), arc, 30 mJ,

230 μs; (*d*) CDI plus superposed current (60 mA), glow discharge, 30 mJ, 770 μs.[23]

The characters of the temperature profiles that each of these three types of discharge create in air, with essentially the same total electrical energy input (30 mJ), are different (see Fig. 9.43). The radial profiles in the mid-plane of the discharge are shown. The expansion-wave-induced expansion of the plasma behind the shock wave generated by the breakdown discharge produces a larger plasma, earlier, with a steep temperature front. Thus it creates favorable conditions for transferring heat and radicals to the surrounding unburned mixture. In addition cold gas flows into the central region of the plasma, due to the boundary layers which the rapidly expanding flow sets up on the electrodes, effectively insulating the hottest plasma region from the cold electrode surfaces. [52] The arc and glow discharges, each preceded by a much lower energy initial breakdown process, show a significantly slower expansion rate and the more gradual temperature profile produced by heat conduction and diffusion.

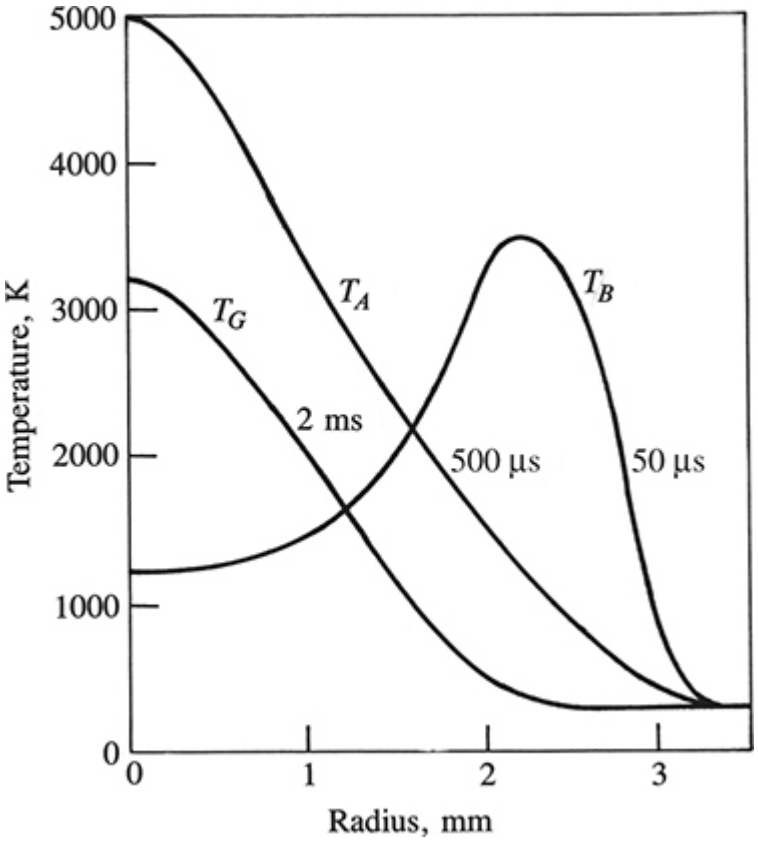


Figure 9.43 Radial temperature profiles at selected times after spark onset for ignition systems with different electrical energies and discharge times in air at 1 atm, τ_b, breakdown discharge, 30 mJ energy, 60 ns duration; T_A, capacitive discharge, 3 mJ energy, 100 µs duration with superimposed current arc of 2 A, 30 mJ energy, for 230 µs; T_G , capacitive-discharge system, 3 mJ energy, 100 µs duration with superimposed constant-current glow discharge of 60 mA, 30 mJ energy, 770 µs duration.[52]

Chemical reactions can be observed spectroscopically a few nanoseconds after spark onset. They are initiated by the very high radical density in the breakdown plasma where all the heavy particles are present as highly excited atoms and ions. Since the kernel temperatures are much too high to allow the species present in normal combustion products to exist, combustion reactions take place at the outer plasma surface where the conditions are ideal for rapid chemical activity (temperatures of a few thousand kelvins). The chemical energy released is added to the plasma energy and becomes

evident when the plasma velocity falls below about 100 m/s (see Fig. 9.41). At this point, the interior of the plasma still consists of a fully dissociated reacted gas mixture with much of its energy stored in radicals. An indication of the gas composition across the steep temperature profile at the plasma interface can be obtained from Table 9.6, which shows the equilibrium composition of C_8H_{18}-air combustion products over the relevant temperature range. Note, however, that the gas will not be in equilibrium. The different radicals have different diffusivities with the hydrogen atom some five times that of other species. Thus the H radical will diffuse furthest into the as-yet unreacted mixture. On the high-temperature side of the inflammation zone, the large number of radical particles transfers their energy to the mixture molecules within a few collisions. On the low-temperature side of the zone, above-equilibrium concentrations of combustion-initiating radicals (O, H, and OH) build up. In addition, a high heat flux into the region occurs, from the plasma core, by conduction down the steep temperature gradient. As a consequence of these conditions, reactions will occur and energy will be released more rapidly than in a normal flame. [52]

TABLE 9.6 Equilibrium composition of stoichiometric isooctane-air combustion products

	Temperature, K†			
Species	2000	3000	4000	5000
CO	2.4(−3)	6.1(−2)	9.4(−2)	9.0(−2)
CO_2	1.2(−1)	5.7(−2)	5.0(−3)	3.5(−4)
H	2(−5)	9.7(−3)	1.2(−1)	1.9(−1)
H_2	6.1(−4)	1.5(−2)	2.4(−2)	3.5(−3)
H_2O	1.4(−1)	1.0(−1)	l.l(−2)	1.2(−4)
N	*n*	l(−5)	6.6(−4)	1.2(−2)
NH	*n*	*n*	2(−5)	6(−5)
NO	5.7(−4)	1.5(−2)	2.9(−2)	1.7(−2)
N_2	7.3(−l)	6.9(−l)	5.7(−l)	5.1(−1)
O	l(−5)	8.6(−3)	9.8(−2)	1.6(−1)
OH	4.5(−4)	2.2(−2)	3.3(−2)	5.6(−3)
O_2	l.l(−3)	2.3(−2)	1.7(−2)	2.2(−3)

†At 4 atm pressure: mole fractions, $9.0(-2) = 9.0 \times 10^{-2}$. $n = <5 \times 10^{-6}$.

The characteristics of the breakdown phase of the discharge have the greatest impact on inflammation. The size of the activated volume at a given time interval after spark initiation, the temperature difference across the kernel outer boundary, and the velocity of the boundary are all substantially increased by increasing the breakdown phase energy. Additional energy input during the arc and glow discharge phases has a more modest effect on these critical kernel properties.

These energy balances for the spark-generated flame kernel growth process, over the 10 μs to fraction of ms time frame are illustrated in Fig. 9.44. These are experimentally based results, and the various energy fluxes are defined in the flame kernel, spark plug, and gap schematic. The flame outline is a schlieren photo taken some 0.8 ms after spark onset, well into the glow-discharge phase. The electrical energy released (and dissipated) in the flame kernel during this period is about 20 W. The heat loss rate to the plug electrodes, which depends on the portion of the flame surface "wetted" by the plug surfaces (which varies cycle-by-cycle), is of comparable magnitude (and opposite in sign). The combustion energy release rate—as the flame propagates outward—is initially comparable to the electrical and heat loss energy transfer rates. As the flame grows, the rate of combustion energy release within the kernel then outpaces the electrical and heat loss components. We would expect (over the first 0.2 ms) cyclic variations in the early flame kernel growth to be significant.

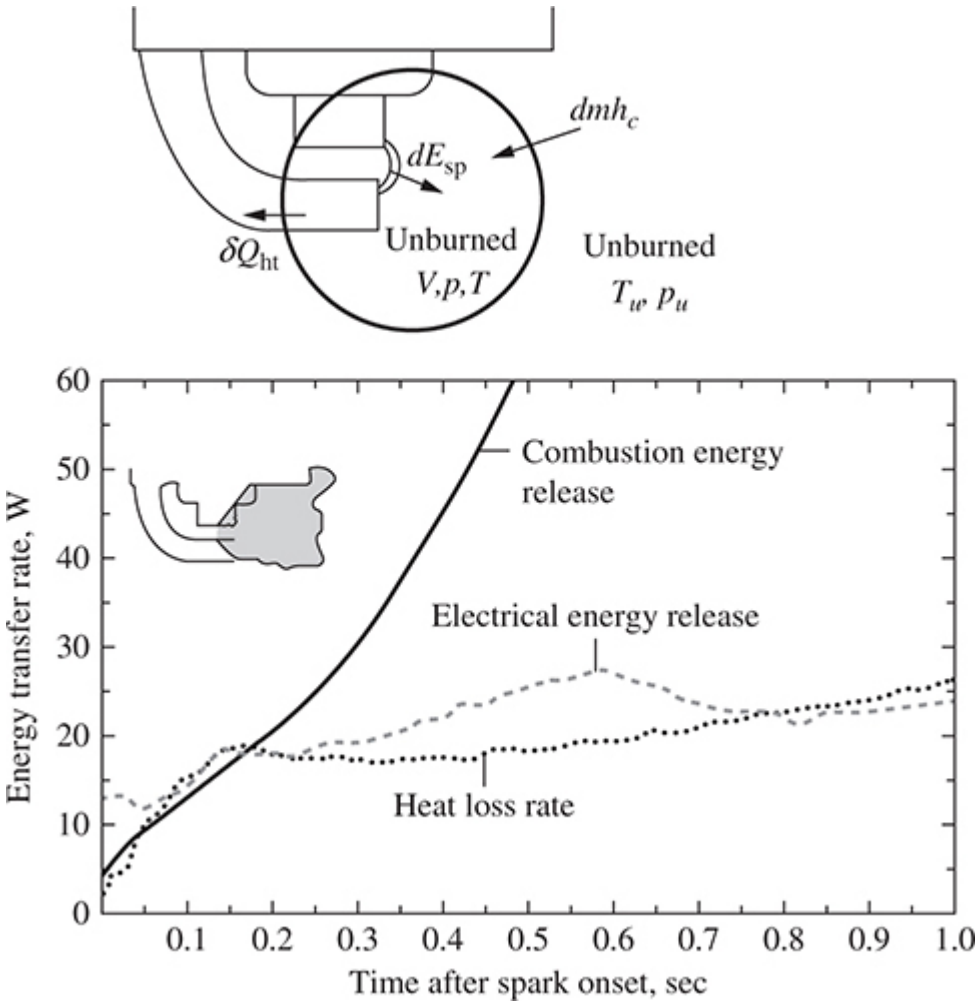


Figure 9.44 Energy flows (in W) for spark-ignited flame kernel: electrical energy dissipation rate, heat loss rate to spark plug electrodes and insulator, fuel chemical energy release rate, as a function of time after spark onset. Spark timing 65° BTC, relative *A/F* 1.25. [28]

Models of the plasma unburned-mixture interface have been developed in attempts to quantify the complex flame initiation and propagation phenomena described above (e.g., Refs. 23, 53, 54, and 55). They are based on the requirement that the energy release due to chemical reaction in the plasma front (and production of radical species) exceed the losses due to conduction and diffusion to the unburned gas ahead of the front. Such models provide a theoretical basis for the well-known fact that the minimum energy required to ignite a premixed fuel-air mixture depends strongly on mixture composition and state. The ignition energy has a minimum for slightly rich-of-

stoichiometric mixtures; the minimum energy required for successful ignition increases rapidly as the mixture is leaned out. While the initial plasma kernel growth (up to 10 to 100 μs) is not greatly affected by the mixture equivalence ratio, the inflammation process and the thickness and rate of propagation of the resulting flame are strongly affected. The lean side of the minimum (and the effect of EGR dilution) is of more practical interest. Because the chemical energy density of the mixture and flame temperature decrease as the mixture is leaned out or diluted, the flame speed decreases and the flame becomes thicker. Thus more time is available for heat losses from the inflammation zone, less energy is available to offset these losses, and the rate of energy transfer into the zone decreases. The consequence is that, as the mixture is leaned out, the approximately spherical discharge-created plasma must grow to a larger size before inflammation will occur. Substantially more energy must therefore be supplied to the discharge. [52]

In engines, the mixture is not quiescent: mean and fluctuating velocities in the range 1 to 10 m/s exist in the clearance volume at TC (see Sec. 8.2.2). On the time scale of the breakdown discharge phase (10 ns), this fluid motion is not important. In the arc and glow discharge phases, however, the arc may be convected by the flow out of the plug electrode gap, and lengthened accordingly. For flow velocities below about 15 m/s, a steady increase in discharge channel length occurs. For higher velocities, an increasing number of re-ignitions occur, so the discharge energy is distributed into many separate channels. As the channel lengthens, the ratio of total discharge voltage to anode plus cathode voltage drop increases substantially, and the relative importance of heat losses to the electrodes decreases. Thus more energy is transferred to the gas—the energy-transfer efficiency increases. However, as the channel is lengthened, the energy transferred is spread over a larger volume. Depending on flow velocity, and mixture and discharge conditions, increasing velocity may increase or decrease the minimum ignition energy or the lean ignition limit for a specific ignition system. Both the mean flow velocity and turbulence levels are important. With a TCI system, increasing mean flow velocity up to the point where re-ignitions start to occur extends the lean limit. With breakdown discharge systems, the lean ignition limit decreases as flow velocity increases. With capacitive-discharge systems at low flow velocities, the flow has little impact: at high flow velocities the minimum ignition energy increases.

Maly and Vogel [52] have summarized the fundamental aspects of spark-discharge ignited flames as follows:

- Of the total electrical energy supplied to the spark, only that fraction contained within the outer surface layer of the plasma (of thickness of the order of the inflammation zone) is available for initiating the flame propagation process. The energy density and the temperature gradient in this layer depend on the discharge mode. Highest energy densities and temperature gradients are achieved if the ignition energy is supplied in the shortest time interval.
- A minimum radius of the spark-generated plasma is required for inflammation of the fuel-air mixture to occur. This radius increases rapidly as the mixture is leaned out (or diluted); it decreases with increasing pressure and increasing plasma expansion velocity.
- After inflammation, burning rates are proportional to flame surface area. Thus discharges and plasma geometries that produce the largest inflammation zone surface area, most rapidly, are advantageous.
- The time over which the ignition energy can be used effectively for inflammation decreases as the initial flame velocity increases. Ignition energy supplied after inflammation has occurred will have only a modest impact on flame propagation.

9.5.2 Standard Ignition Systems

The ignition system must provide sufficient voltage across the spark plug electrodes to initiate the discharge and supply enough energy to the discharge to ignite the combustible mixture adjacent to the plug electrodes under all operating conditions. It must create this spark at the appropriate time in each cylinder's operating cycle (normally toward the end of the compression stroke). Usually spark timing is set to give maximum brake torque for that specific operating condition, though the largest torque available may be constrained by maximum pressure limits, or knock control requirements. For a given engine design, this optimum spark timing varies as engine speed, inlet manifold pressure, and mixture composition vary. Thus, in most applications, and especially automotive applications, the system must have means for automatically changing the spark timing as engine speed and load vary.

With an equivalence ratio best suited for ignition (close-to-stoichiometric) and with homogeneous mixture distribution, spark energies of order 1 mJ and durations of a few microseconds would suffice to initiate the combustion process. In practice, circumstances are less ideal. The air, fuel, and recycled exhaust are not uniformly distributed between cylinders; the mixture of air, fuel, recycled exhaust gas, and residual gas within each cylinder is not homogeneous. Also, the pressure, temperature, and density of the mixture between the spark plug electrodes at the time the spark is needed affect the voltage required to produce a spark. These vary significantly over the load and speed range of an engine. The spark energy and duration, therefore, have to be sufficient to initiate combustion under the most unfavorable conditions expected in the vicinity of the spark plug over the complete engine operating range. Usually, if the spark energy exceeds about 50 mJ and the duration is longer than 0.5 ms, reliable ignition is obtained.

In addition to the spark requirements determined by mixture quality, pressure, temperature, and density, there are others determined by the state of the spark plugs. The erosion of the plug electrodes over extended mileage increases the gap and requires a higher breakdown voltage. Also, spark plug fouling due to deposit buildup on the spark plug insulator can result in side-tracking of the spark. When compounds formed by the burning of fuel, lubricating oil, and their additives are deposited on the spark plug insulator, these deposits provide an alternative path for the spark current. If the resistance of the spark plug deposits is sufficiently low, the loss of electrical energy through the deposits may prevent the voltage from rising to that required to break down the gas. The influence of side-tracking on spark generation decreases with lower source impedance of the high-voltage supply, and therefore with a higher available energy.

The fundamental requirements of the high-voltage ignition source can be summarized as: (1) a high ignition voltage to break down the gap between the plug electrodes; (2) a low source impedance to achieve a rapid voltage rise; (3) a high-energy storage capacity to create a spark kernel of sufficient size; and (4) sufficient duration of the discharge to ensure ignition. There are several commonly used concepts that partly or fully satisfy these requirements. [51]

Coil Ignition Systems

Inductive coil ignition systems have been used in spark-ignition engines for many years. TCI systems with their advantageous performance characteristics and low cost dominate. Figure 9.38 shows the general circuit arrangement of a coil ignition system. The system includes an ignition driver stage, an ignition coil with a primary and secondary winding, a spark plug connected to the high-voltage connection point of the wiring. The circuit functions as follows. Before the spark is needed, the ignition driver stage switches a current from the electrical system through the primary winding of the coil: this builds up a magnetic field in the ignition coil's iron core. At time of ignition, the current through the primary winding is interrupted. The decay of magnetic flux in the coil induces a voltage in both the primary and secondary windings. The voltage induced in the secondary winding is routed to the spark plug to produce the ignition discharge.

Typical current and voltage waveforms are shown in Fig. 9.39. The primary current for any given dwell time t is given by

$$I_p = \frac{V_0}{R}(1 - e^{-Rt/L_p}) \tag{9.62}$$

where I_p is the primary current, V_0 is the supply voltage, R is the total primary circuit resistance, and Lp is the primary circuit inductance. The primary current requires time to build up. At low speeds the time of contact closure is sufficient for the primary current to reach the maximum permitted by the circuit resistance; at high speeds the primary current may not reach its maximum. Thus, only at higher engine speeds does the term e^{-Rt/L_p} becomes significant. When the primary current falls to zero, a voltage of order 15 kV is induced in the secondary winding. The peak value of this voltage is the maximum voltage that can be produced by the system and is called the *available voltage Va*. The maximum energy transferred to the secondary system is given by

$$E_{s,\max} = \frac{1}{2} C_s V_a^2$$

where Cs is the total capacitance of the secondary circuit. Hence, the available voltage of the system is given by

$$V_a = \left(\frac{2E_{s,\max}}{C_s} \right) \tag{9.63}$$

If all the energy stored in the primary circuit of the coil, $\frac{1}{2} L_p I_p^2$, is transferred to the secondary,

$$V_a = I_p \left(\frac{L_p}{C_s} \right)^{1/2} \tag{9.64}$$

When the coil is connected to a spark plug, the secondary voltage will rise to the breakdown potential of the spark plug, and a discharge between the plug electrodes occurs. During this first breakdown phase, as explained earlier, on the 10 nanosecond timescale a current of some 10 A flows as the voltage reduces to a lower value (some 100 V) over about 1 μs as the energy is dissipated in the arc phase. The value of the voltage which caused breakdown to occur is called the *required voltage* of the spark plug. The interval during which the discharge occurs is called the *spark duration.* The available voltage of the ignition system must always exceed the required voltage of the spark plug to ensure breakdown. The spark must then possess sufficient energy and duration to initiate combustion under all conditions of operation.

In automotive applications, the need for reduced ignition system maintenance, extended spark plug life, improved ignition of lean and dilute mixtures, and increased reliability has led to the widespread use of transistorized coil ignition systems which provide a higher output voltage and which use electronic triggering to realize the required timing without wear or adjustment. The higher output voltage is required because spark plugs are now set to wider gaps (about 1 mm) to extend the ability to ignite the fuel/air burned gas mixture over a wider range of engine operation, and because during the extended mileage between spark plug replacement, electrode erosion further increases the gap. In automotive applications an available ignition voltage of some 30 kV is now usually provided. In addition to higher voltage, longer spark duration (about 2 ms) has been found to extend the engine operating conditions over which satisfactory ignition is achieved.

Capacitive-Discharge Ignition (CDI) Systems

With this type of system, a capacitor rather than an induction coil is used to store the ignition energy. The capacitance and charging voltage of the capacitor determine the amount of stored energy. The ignition transformer steps up the primary voltage, generated at the time of spark by the discharge of the capacitor through a thyristor, to the high voltage required at the spark plug. The CDI trigger box contains the capacitor, thyristor power switch, charging device (to convert battery voltage to the charging voltage of 300 to 500 V by means of pulses via a voltage transformer), pulse shaping unit, and control unit. The principal advantage of CDI is its insensitivity to electrical shunts in the high-voltage ignition circuit that result from spark plug fouling. Because of the fast capacitive discharge, the spark is strong but short (0.1 to 0.3 ms). This can lead to ignition failure at operating conditions where the mixture is very lean or dilute.

Magneto Ignition System

With this type of ignition system, a magneto supplies the ignition voltage for the spark discharge independent of a battery or generator. Magneto ignition is used in small-displacement four-stroke and two-stroke engines. The system operates as follows. A time-varying magnetic flux is set up in the ignition armature as the rotating permanent magnets on the pole wheel generate a current in the closed primary winding. To generate the ignition voltage, the primary current flow is interrupted and the flux collapses rapidly producing a high-voltage pulse in the winding which is connected to the spark plug electrode. The current can be interrupted with contact breakers or with a transistor. Since the flux generated by the rotating pole wheel depends on engine speed, the magnitude of the ignition voltage varies with speed.

Spark Plug Design

The function of the spark plug is to provide an electrode gap across which the high-voltage discharge occurs that ignites the compressed mixture of fuel vapor, air, and burned (residual) gas in the combustion chamber. In addition, it must provide a gas-tight conducting path from the high-voltage source to the electrode gap. Figure 9.45 shows a typical spark plug design. There are three principal components: electrodes, an insulator, and a shell. The insulating material must have adequate thermal shock resistance, tensile and compressive strength, and impact strength. It must also have low porosity to

limit absorption of combustion gases and high resistivity to prevent leakage of high-voltage charge at both ambient and normal operating temperatures. Alumina is usually used as the insulator material. The electrodes are normally made of high-nickel alloys to withstand the high ignition voltage, high temperatures, and corrosive gases with minimum erosion. In the standard J-plug shown, the center-electrode surface temperatures can reach about 800°C under normal operating conditions. The ground electrode temperature can reach about 1000°C. [57]

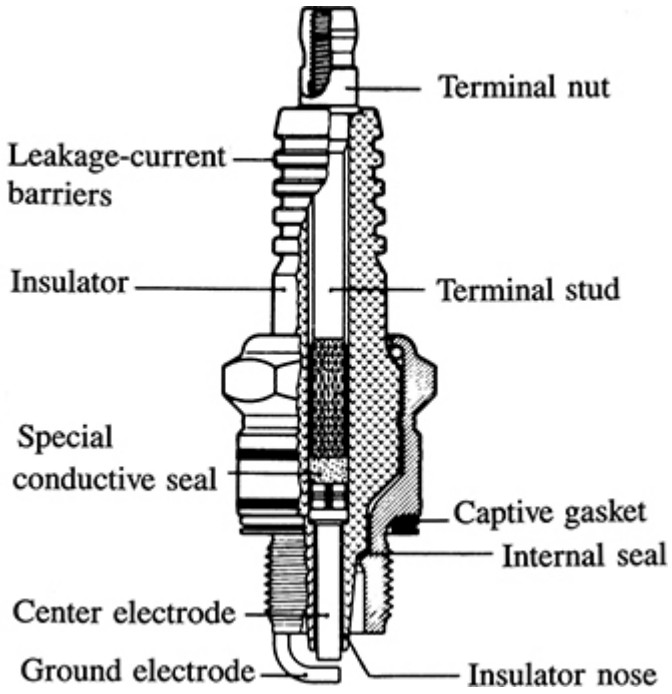


Figure 9.45 Cutaway drawing of standard J-type spark plug. (*Courtesy Robert Bosch Gm$_b$ H and SAE.* [56])

Several aspects of the J-plug electrodes impact its performance in an operating engine. Performance is often characterized as the lean (λ) or dilute (EGR percentage) limit at which the standard deviation (s) or coefficient of variance (COV) in imep reaches a significant level—e.g., a COV $_{imep}$ of 5% or more, a higher value than is tolerable in a production engine. The orientation of the ground electrode (in relation to the mean flow direction at the plug at time of ignition) affects this limit. Convecting the developing flame kernel from the electrode gap across the plug diameter toward the ground electrode increases the combustion variability: the heat losses from

the flame kernel to the ground electrode are highest with this orientation. Also, the cross-sectional shape and size of the ground electrode and center electrode affect the lean limit and combustion variability through the variability in kernel heat loss to the electrodes and the efficiency of the transfer of electrical energy to the spark-generated flame. [58]

The dilution of fuel-air mixture with air (leaner operation), or with exhaust gas recycle, have analogous but not identical effects on ignition. Dilution with EGR degrades ignition system performance (as measured by variability in imep, as a result of combustion variability) more than equivalent dilution with excess air. The dilute operating limits at lighter engine loads with air are almost twice as far beyond stoichiometric in terms of percent dilution, than the extension with exhaust gas recycle beyond zero EGR, due to the greater impact (per unit mass of fuel) of EGR on the mixture burning rate. With modest dilutions ($\lesssim$10% EGR) there is little difference in the ignition system behavior for EGR and air dilution. [59]

The location of the special conductive seal within the plug insulator shell (see Fig. 9.45) affects the *heat rating* of the spark plug. For a "hot" plug, an insulator with a long conical nose is used; for a "cold" plug a short-nosed insulator is used. The length of the heat conduction path from the insulator nose to the shell is changed in this way to vary and control the temperature of the exposed part of the insulator. The spark plug insulator tip temperature increases with increasing engine speed. It is desirable to have the tip hotter than about 500°C to prevent insulator fouling at low speed and load. High-speed high-load tip temperatures must be kept below about 1100°C to prevent surface ignition. Normally, the gap between the center and ground electrodes is 0.8 to 1.2 mm. For extremely dilute mixtures this is usually increased. Magneto ignition systems use smaller gaps (~0.5 mm). High-compression-ratio racing engines use still smaller gaps (0.3 to 0.4 mm).

9.5.3 Alternative Ignition Approaches

A large number of methods for initiating combustion in spark-ignition engines with electrical discharges, in addition to those described in the previous section, have been proposed and examined. These include different designs of spark plug, use of more than one plug, use of higher power, higher energy, or longer-duration discharges, multi-spark systems, and ignition systems that

initiate the main combustion process with a high-temperature reacting jet—plasma-jet and flame-jet ignition systems. The standard ignition systems normally ignite the unburned fuel, air, burned gas mixture within the cylinder and perform satisfactorily under conditions away from the lean or dilute engine stable operating limit. Thus, these alternative ignition approaches have the goal of extending the engine's stability limit (and/or of reducing the cyclic combustion variability near the stability limit), usually by achieving a faster initial burning rate than can be obtained with the more conventional systems. This section describes the more interesting of these alternative ignition approaches.

Alternative Spark-Discharge Approaches

There are many different designs of spark plugs. These use different geometry electrodes, gap widths, and gap arrangements. Electrode arrangements are often categorized as air-gap (e.g., the standard J-type plug design), surface-gap (where the discharge occurs on the insulator surface), and surface-air-gap (a hybrid of these two).

The effects of several plug electrode design features on the engine's stable lean operating limit are illustrated in Fig. 9.46. Ignition system effects are important when misfire due to the quenching effect of the spark plug electrodes determines the stable operating limit. Thus, smaller spark plug center-electrode diameters, larger electrode gap widths, and use of a small diameter platinum center electrode, all extend the lean stability limit to leaner (or more dilute) mixtures for normal spark timings. Figure 9.47 shows how some of the plug designs in Fig. 9.46 affect the standard deviation (and thus COV) of imep as the engine is leaned out. For stoichiometric and up to about 20% lean operation, at this load and speed, the plug and ignition system changes do not affect the combustion variability. The level of dilution at which the imep standard deviation starts to increase rapidly, however, is significantly affected. Note that for the leaner ($> \lambda = 1.3$) mixtures, the CDI system with comparable stored energy (140 mJ) to the TCI system (150 mJ), but a shorter discharge time (0.25 ms instead of 1.7 ms) is not as effective. The plug with thinner electrodes and platinum center electrode has the leanest stable operating limit due to its higher energy-into-the-plasma transfer efficiency, and its lower heat losses to the electrodes due to their smaller surface area in contact with the flame kernel. [59] Multi-gap plugs, designed to produce several discharges which together form a longer arc,

have also been used to generate a larger initial flame kernel and thereby extend the lean limit.

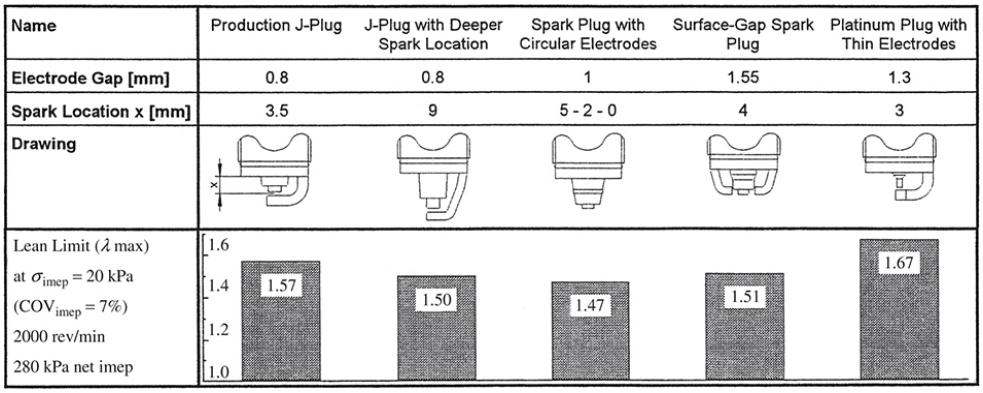


Figure 9.46 Several different spark plug designs (with electrode dimensions) and their lean operating limit (λ_{max}) at 2000 rev/min, 280 kPa net imep, moderate tumble, $COV_{imep} = 7\%$.[59]

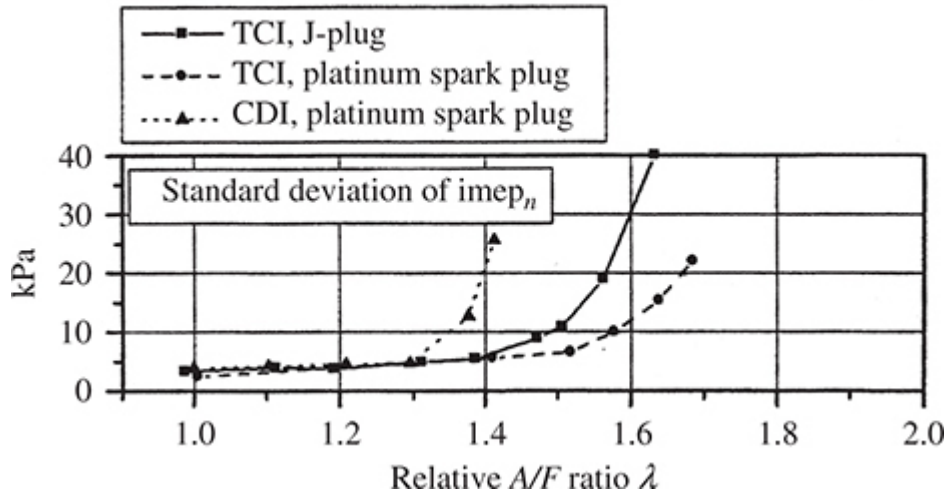


Figure 9.47 Standard deviation of net imep as a function of relative *A/F*, λ, at 2000 rev/min, 280 kPa net imep, and moderate tumble in-cylinder flow: TCI J-type plug, TCI thin platinum-center-electrode plug, and CDI platinum electrode plug.[59]

Use of more than one plug, at separate locations in the combustion chamber and fired simultaneously, is also an option. The advantages are twofold. First, the effective flame area in the early stages of flame

development is increased substantially: by about a factor of two for two widely spaced plugs. Second, the variations in flow velocity and mixture composition in the vicinity of the (multiple) plugs produce less combined variability in the initial mixture-burning rate than occurs with a single plug. Studies of heat-release rates, flame development angles, and rapid-burning angles, and imep and torque fluctuations, have defined the effects of both increasing the number of ignition sites from 1 to 12 and of changing their geometric location. These results confirm that increasing the number of simultaneously developing flame kernels increases the *initial* mixture-burning rate, as anticipated. It also extends the lean stable operating limit and reduces cyclic combustion variability under conditions where slow and occasional partial-burning cycles would occur with fewer spark plug gaps. [60]

Surface gap plugs (or semi-surface gap plugs) are available, and are considered an option, especially for direct-injection spark-ignition engines. An example of such a spark plug is shown in Fig. 9.46. Two (or more) ground electrodes are used, partly overlapping the insulator tip so the discharge occurs along the insulator surface. The surface discharge provides self-cleaning of the insulator tip which in direct-injection engines, especially when these are operated in stratified mode (see Sec. 9.3.5) where carbon fouling of the insulator is more likely to occur. A potential problem with surface plugs is damage to the insulator tip due to the spark current channeling across the surface. With semi-surface multiple-electrode plug designs (where the spark takes both a gas-spark and surface-spark path) channeling problems can be overcome. The self-cleaning and dispersed sparking features of these plugs stabilize stratified-charge combustion. This type of plug is claimed to be advantageous for the more demanding ignition requirements of direct-injection engines, which operate at higher pressures. [61]

Plug and ignition systems that provide multiple spark discharges in sequence are another option. It is thought that each “restrike” of the discharge results in a fresh arc, thus increasing the efficiency with which electrical energy from the ignition coil is transferred to the spark-generated kernel.

In a broader context, the spark-plug electrodes, with modifications to the ignition system electrical circuit, can be used as an ion current sensor. Once the spark-ignition function has ended, a voltage is applied between the plug electrodes and, by measuring the current (carried by ions) in the hot combustion gases in the electrode gap, the electrical conductivity of that gas

can be determined. Thus misfiring engine cycles can be detected since then the gas conductivity is very low due to the low temperature of the unburned mixtures. Also, knock can be detected. The oscillations in peak in-cylinder gas pressure cause oscillations in the temperature of the burned gas that surrounds the spark plug, which cause oscillations in the gas's electrical conductivity, and therefore in the ion current. Thus, there is potential for the ignition system and its spark plug to not only ignite the mixture in the cylinder reliably, but also act as a misfire and knock detector.

Many studies have examined the effects of higher-energy discharges on engine operation near the lean operating limit. It is useful to differentiate between higher current discharges and longer duration discharges: most high-energy (conventional-type) ignition systems have both these features. The results of these studies show that away from the lean or dilute stable operating limit, increasing the discharge current or duration has no significant effect on engine operating characteristics. The higher current does, as would be expected, result in a larger flame kernel during the initial inflammation process and thereby modestly reduces the spark advance required for MBT with a given combustion chamber and set of operating conditions. However, both higher currents and longer duration discharges do extend the lean engine stability limit (and also the dilute, with EGR, stability limit). Hydrocarbon emissions data indicate that longer discharges have a greater impact than higher currents on extending the misfire limit. The observed increase in fuel consumption as the lean operating limit is approached, results from the rapidly increasing cycle-by-cycle combustion variability, which reduces the average imep and torque. Higher breakdown and arc discharge energies increase electrode erosion.

The discussion of discharge fundamentals in Sec. 9.5.1 showed that depositing energy into the discharge during the initial short breakdown phase resulted in faster flame kernel growth than did depositing the same energy over longer times. In such *breakdown* ignition systems, a capacitor is connected in parallel with the spark plug electrodes, and a low-impedance discharge path allows the energy stored in the capacitor to be discharged into the gap very rapidly. The anticipated effects on the engine's combustion process are observed. Away from the lean engine stability limit, the primary impact is a reduction in the flame development period due to the more rapid initial flame kernel growth. Thus MBT timing is less advanced with these breakdown systems than with conventional systems, as shown in Fig. 9.48.

The lean limit can also be extended, and acceptable engine stability obtained (i.e., tolerable cycle-by-cycle combustion variations) for leaner or more dilute engine operation, as shown in Fig. 9.48 *b*. [62, 63]

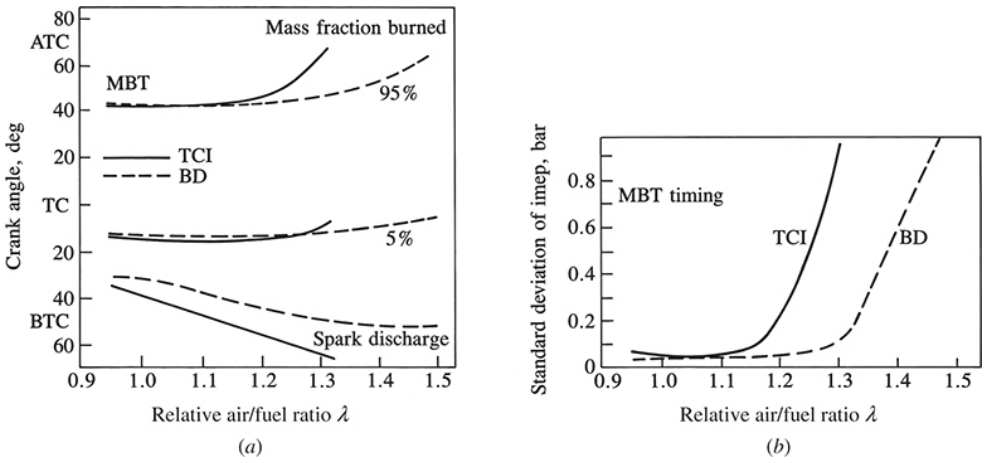


Figure 9.48 (*a*) MBT spark timing and location of 5 and 95% mass fraction burned points for conventional transistorized coil ignition (TCI) system (43 mJ energy, 2 ms duration) and breakdown system (BD) (43 mJ energy, ~10 ns duration) as function of relative air/fuel ratio λ (=1/φ). (*b*) Standard deviation in indicated mean effective pressure as a function of relative air/fuel ratio λ for TCI and BD systems. [62]

Plasma-Jet Ignition

In a plasma-jet ignitor, the spark discharge occurs within a cavity that surrounds the plug electrodes, which connects with the combustion chamber via an orifice (see Fig. 9.49 *a*). The electrical energy supplied to the plug electrodes is substantially increased above values used in normal ignition systems by allowing a capacitor to discharge at a relatively low voltage and high current, through the spark generated in a conventional manner with a high-voltage low-current ignition system. Stored energies of several hundred mJ are used, and this energy is discharged in some 20 μs. This high-power discharge creates a high-temperature plasma so rapidly that the pressure in the cavity increases substantially, causing a supersonic jet of plasma to flow from the cavity into the main combustion chamber. The plasma enters the combustion chamber as a turbulent jet, preceded by a hemispherical blast wave. The gas dynamic effects of the blast wave are dissipated by the time combustion starts. The initial combustion rate is typically an order of

magnitude faster than with the standard TCI system. Ignition in the main combustion chamber takes place in the turbulent plasma jet; the flame starts out as a turbulent flame in contrast to the flame with conventional ignition systems which is initially laminar-like. The penetration of the jet depends on its initial momentum; it thus depends on the amount of energy deposited, cavity size, and orifice area. If the cavity is filled prior to ignition with a hydrocarbon (or a mixture of hydrocarbons), the ignition capabilities are enhanced due to the large increase in hydrogen atoms created in the plasma. [59, 64]

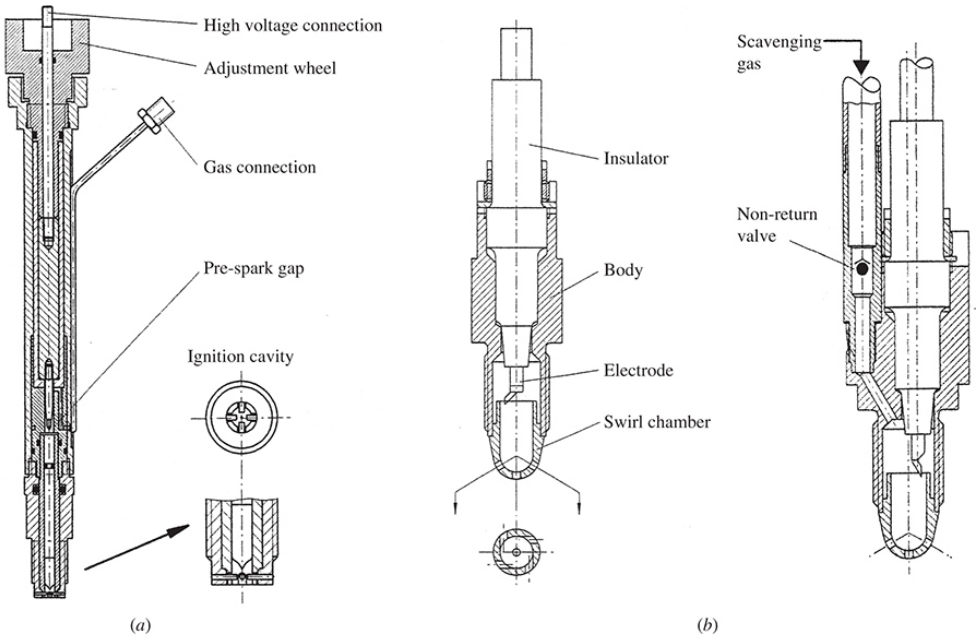


Figure 9.49 (*a*) Plasma-jet ignition system and (*b*) swirl chamber spark plug (unscavenged swirl chamber on left, scavenged swirl chamber on right).[59]

The effects of plasma-jet ignitors on engine combustion are analogous to those of breakdown ignition systems. The flame development period is significantly shortened: thus ignition at MBT timing occurs closer to TC. The lean/dilute mixture levels where cycle-by-cycle combustion variability becomes large are significantly extended, as are the engine's lean or dilute stable operating limit. [59]

Flame-Jet Ignition

With this type of system, ignition occurs in a prechamber cavity, which is physically separated from the main chamber above the piston and is connected to it via one or more orifices or nozzles. See the swirl chamber and scavenged chamber plug designs in Fig. 9.49 *b*. Locating the spark plug gap within a prechamber results in increased velocities and gas temperatures (through heat transfer from the prechamber walls) in the mixture surrounding the plug electrodes. As the flame develops in this cavity, the pressure of the gases in the prechamber rises, forcing gas out into the main chamber through the orifice (or orifices) as one or more burning turbulent jets. The jet, or with a larger orifice the "cloud," penetrates into the main chamber igniting the unburned mixture in the main chamber, thereby initiating the primary combustion process. Ignition within the cavity is usually achieved with a standard spark discharge. The function of the prechamber or cavity is to transform the initial flame around the spark plug electrodes into one or more pressure-driven flame jets (or cloud) into the main chamber, which have a substantial surface area that can ignite extremely lean or dilute mixtures in a repeatable manner. Many different systems for achieving this goal have been developed; some of these have been used in production spark-ignition engines.

In the simpler types of flame-jet ignition concept, the cavity is unscavenged and there is no prechamber fuel metering system. The function of the prechamber cavity is to increase the initial growth rate of the flame immediately following spark discharge. Another approach is to incorporate a (small) cavity into the spark plug. Systems with prechamber volumes varying from 20% of the clearance volume to about 1% have been developed. The flow pattern produced within the prechamber by the flow into the cavity during compression, the location of the spark plug electrodes within the cavity, and of the nozzle or orifice, are critical design issues. A problem with these systems is that the prechamber is never completely scavenged by fresh mixture between cycles, so the burned gas fraction in the unburned mixture within the prechamber is always higher than the burned gas fraction in the unburned main-chamber mixture.

The ignition and combustion characteristics of the swirl-chamber (SC) plugs, such as those in Fig. 9.49 *b*, are shown in Fig. 9.50 where they are compared to the standard J-plug system. The size of prechamber volume is 2% of the cylinder clearance volume. The SC plug center electrode has a platinum pin welded to it. The ignition spark arcs from the pin to the

prechamber wall. The pressure differences at this part-load low-speed condition between pre- and main-chamber, as combustion transitions from the first of these regions to the latter, are 100 to 200 kPa. Comparisons of combustion phasing with the standard ignition system, as the engine is leaned out, show that the optimum (MBT) spark timing for the flame-jet swirl chamber plug is significantly less advanced than MBT spark for a standard J-plug system due to a faster initial burning rate as the flame develops. However, the crank angles of the middle and end of combustion are closely comparable.

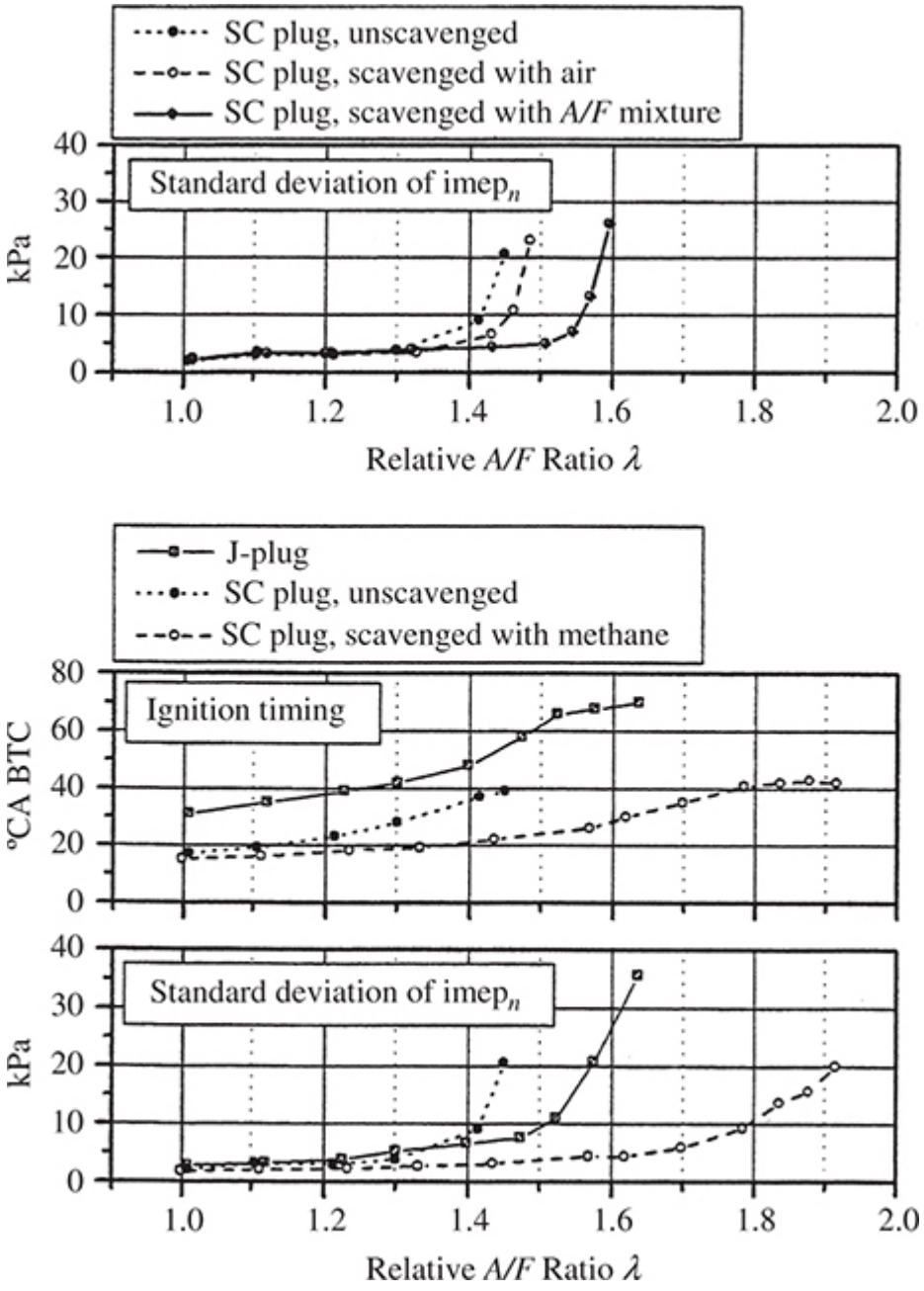


Figure 9.50 Combustion characteristics (standard deviation of net imep and ignition timing) as a function of relative air/fuel ratio of swirl chamber spark plug: unscavenged, and scavenged with air, air-fuel mixture, and methane. J-plug characteristics shown for comparison. 2000 rev/min, 280 kPa net imep, moderate tumble. [59]

The lean operating limits of the different SC plugs depend on whether these plugs are unscavenged, or scavenged with various scavenging gases.

The stable lean operating limit is defined by the rapid rise in standard deviation of imep caused by the rapidly worsening cycle-by-cycle combustion variability as the mixture is increasingly leaned out. In unscavenged SC plug operation, this limit is less lean than with the standard J-plug TCI system. Scavenging improves the lean limit: modestly with air scavenging, more with air-fuel mixture, and most with methane. Scavenging removes the high concentration of residual burned gas left in the swirl chamber after combustion, and the different scavenging gases, air, air-fuel mixture, and fuel, in this sequence produce mixture that is successively easier to ignite and burn in the prechamber.

With close to stoichiometric operation, combustion variability impacts are small. With significantly diluted mixtures, the more rapid and more repeatable combustion initiation process in the pre- and main-chamber does provide benefits. There is a history of prechamber stratified engines where, in addition to the main fuel-injection process which occurs into the fresh charge in the intake system, at the time of spark a readily-ignitable rich-of-stoichiometric mixture was set up in the prechamber. With this approach, the prechamber volume was usually 20 to 25% of the clearance volume. A separate small intake valve fed very rich mixture into the prechamber during the intake process, while the main fuel metering system fed lean mixture to the main intake valve. During intake, the prechamber is completely scavenged by the rich intake stream. During compression, the lean mixture flowing from the main chamber to the prechamber brings the prechamber mixture equivalence ratio to slightly rich-of-stoichiometric at the time of spark discharge.

Summary

Overall, several of these alternative ignition concepts extend the engine's lean stable operating limit relative to conventional engines with standard ignition systems by several air/fuel ratios. However, with stoichiometric operation (even with 10 or so percent EGR) "more powerful" ignition systems have little impact on the stability of the engine's combustion process. The primary reason is that under these conditions, combustion variability, cycle-by-cycle, is modest and "improved ignition" is not needed. That is, if ignition is already satisfactory a better ignition system is of limited value.

9.6 ABNORMAL COMBUSTION: SPONTANEOUS IGNITION AND KNOCK

9.6.1 Description of Phenomena

Abnormal combustion in SI engines reveals itself it many ways. Of the various abnormal combustion processes which are important in practice, two major phenomena are knock and surface ignition. These abnormal combustion phenomena are of concern because: (1) when severe, they can cause major engine damage; and (2), even if not severe, they are regarded as objectionable by the engine or vehicle operator. *Knock* is the name given to the noise which is transmitted through the engine structure when spontaneous ignition of a significant portion of the end-gas—the fuel, air, residual gas, mixture ahead of the propagating flame—occurs. When this abnormal combustion process takes place, there is usually an extremely rapid release of chemical energy in the end-gas, causing very high local pressures and the propagation of pressure waves of substantial amplitude across the combustion chamber. *Surface ignition* is ignition of the fuel-air mixture by a hot spot on the combustion chamber walls such as an overheated valve or spark plug, or glowing combustion chamber deposit, that is by a means other than the normal spark discharge. It can occur before the occurrence of the spark (*preignition*) or after (*postignition*) . Following surface ignition, a flame develops at each surface-ignition location and starts to propagate across the chamber in an analogous manner to what occurs with normal spark ignition.

Because the spontaneous ignition phenomenon that causes knock is governed by the temperature and pressure history of the end gas, and therefore by the phasing and rate of development of the combustion process, various combinations of these two phenomena—localized spontaneous ignition and knock—can occur. These are categorized in Fig. 9.51. When autoignition occurs repeatedly during otherwise normal combustion events, the phenomenon is called *spark-knock.* Repeatedly here means occurring more than occasionally: the knock phenomenon varies substantially cycle-by-cycle, and between the cylinders of a multi-cylinder engine, and does not necessarily occur every cycle (see below). Spark-knock is controllable by the spark advance: advancing the spark increases the knock severity or

intensity and retarding the spark decreases knock. Since surface ignition usually causes a more rapid rise in end-gas pressure and temperature than occurs with normal spark ignition (because the flame either starts propagating sooner, or propagates from more than one source), knock is a likely outcome at higher engine loads following the occurrence of surface or local ignition. To identify whether or not surface ignition causes knock, the terms knocking surface ignition and non-knocking surface ignition are used. Knocking surface ignition usually originates from preignition caused by glowing combustion chamber deposits: the severity of knock generally increases the earlier that preignition occurs. Knocking surface ignition cannot normally be controlled by retarding the spark timing since the spark-ignited flame is not the cause of knock. Non-knocking surface ignition is usually associated with surface ignition that occurs late in the operating cycle.

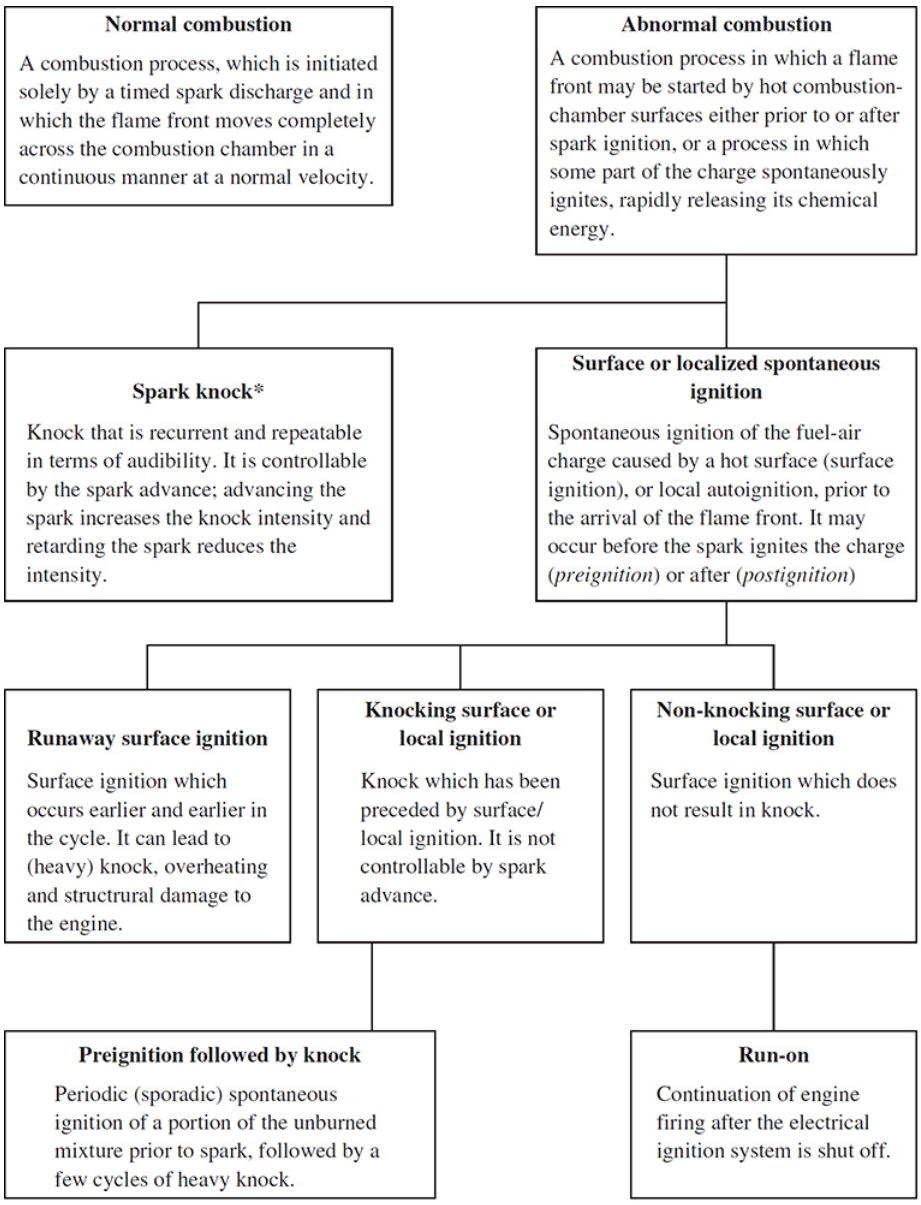


Figure 9.51 Definition of the various normal and abnormal spark-ignition engine combustion phenomena: spark knock, spontaneous surface and local ignition, sporadic preignition, and their relationships.

As in-cylinder pressures have risen due to increasing engine compression ratios and engine turbocharging, a new knock phenomenon has been observed: *sporadic preignition*. Intermittently, a local region of the in-cylinder unburned mixture autoignites toward the end of the compression

stroke before the spark discharge occurs. The subsequent chemical energy release can result in abnormally high pressures and unburned mixture temperatures which can lead to heavy engine knock. Effectively, in a few cycles (once every 10^4 or so cycles), this preignition-initiated combustion is very over advanced, and extremely high cylinder pressures result. This stochastic preignition lasts for one, or a few cycles, and then stops. The probable cause of this phenomenon is that oil from the region between the cylinder liner and the piston top land is periodically transported into the cylinder forming a local region of autoignitable hydrocarbon-air mixture. This localized spontaneous ignition of mixture in the cylinder is a sporadic, seemingly stochastic, preignition process that initiates an over-advanced burning process in the rest of the combustion chamber, which leads to heavy knock.

Runaway surface ignition is surface ignition that occurs earlier and earlier in the cycle. It is usually caused by overheated spark plugs or valves or other combustion chamber surfaces. It is the most destructive type of surface ignition since it can lead to serious overheating and structural damage to the engine.

After some additional description of surface-ignition phenomena, the remainder of Sec. 9.6 will focus on knock. This is because surface ignition is a problem that can be solved by appropriate attention to engine design, and fuel and lubricant quality. In contrast, spark-knock is an inherent constraint on engine performance and efficiency since it limits the maximum compression ratio (and boost levels in turbocharged engines) that can be used with any given fuel.

Of all the engine surface-ignition phenomena identified in Fig. 9.51, preignition is potentially the most damaging. Any process that advances the start of combustion ahead of the timing that gives maximum torque will result in higher unburned mixture temperatures and pressures, and thus heavier knock. It also results in higher heat rejection because of the increasing burned gas pressures and temperatures that result. Higher heat rejection causes higher component temperatures, which in turn, can advance the preignition point (with surface preignition) even further until critical components can fail. The surfaces which can cause preignition are those least well cooled and where deposits build up and provide additional thermal insulation: primary examples are spark plugs, exhaust valves, and deposit build-up on the piston crown and cylinder head. Under normal conditions, using suitable

heat-range spark plugs, preignition is usually initiated by an exhaust valve covered with deposits formed from the fuel, and the lubricant which penetrates into the combustion chamber. Cooler exhaust valves and reduced oil consumption usually alleviate this problem. Engine design features that minimize the likelihood of preignition are: appropriate heat-range spark plugs, removal of sharp asperities, rounded metal edges, and well-cooled exhaust valves with sodium-cooled valves as an extreme option.

Knock primarily occurs at loads approaching wide-open-throttle operating conditions. It is thus a direct constraint on engine performance. It also constrains engine efficiency, since by effectively limiting the temperature and pressure of the end-gas at higher loads, it limits the engine compression ratio. The occurrence and severity of knock depend on the knock resistance of the fuel and on the knock suppressing features of the engine. The ability of a fuel to resist knock is measured by its octane number (ON): higher ONs indicate greater resistance to knock (see Sec. 9.6.3). Gasoline octane ratings can be improved by refining processes such as catalytic cracking and reforming, which convert low-octane hydrocarbons to high-octane hydrocarbons. The octane-number requirement of an engine depends on how its design and the conditions under which it is operating, affect the temperature and pressure of the end-gas ahead of the flame, and on the time required to burn the cylinder charge. An engine's tendency to knock, as defined by its *octane requirement*—the octane rating of the fuel required to avoid knock—is increased by factors that produce higher temperatures and pressures or lengthen the burning time. Thus knock is a constraint that depends on both the quality of available fuels and on the ability of the engine designer to achieve the desired normal combustion behavior while maximizing the engine's performance.

The pressure variation in the cylinder during knocking combustion indicates in more detail what actually occurs. Figure 9.52 shows the cylinder pressure variation in three individual engine cycles, with normal combustion, light knock, and heavy knock, respectively.[65] When knock occurs, high-frequency pressure fluctuations are observed whose amplitude decays with time. Figures 9.52 *a* and *b* have the same operating conditions and spark advance. About one-third of the cycles in this engine at these conditions had normal, smoothly varying, cylinder pressure records as in Fig. 9.52 *a*. Knock of varying severity occurred in the remaining cycles. With light knock, knock occurs late in the burning process, at close to the peak pressure, and the

amplitude of the pressure fluctuations is small (Fig. 9.52*b*). Note that the peak pressure in the center figure is higher and earlier than that in the normal cycle; it is a faster-burning cycle. With heavy knock, illustrated here with more advanced spark timing and by selecting a high intensity knocking cycle, knock occurs closer to top-center earlier in the combustion process and the initial amplitude of the pressure fluctuation is much larger. These pressure fluctuations produce the sharp metallic noise called "knock." They are the result of the essentially spontaneous release of much of the end-gas fuel's chemical energy. This produces a substantial *local* increase in gas pressure and temperature, thereby causing a strong pressure wave to propagate away from the end-gas region across the combustion chamber. This strong pressure wave, the expansion wave that accompanies it, and the reflection of these waves by the chamber walls create the oscillatory pressure versus time records shown in Figs. 9.52*b* and *c*. Note that once knock occurs, the pressure distribution across the combustion chamber is no longer uniform: transducers located at different points in the chamber will record different pressure levels at a given time until the wave propagation phenomena described above have been damped out.

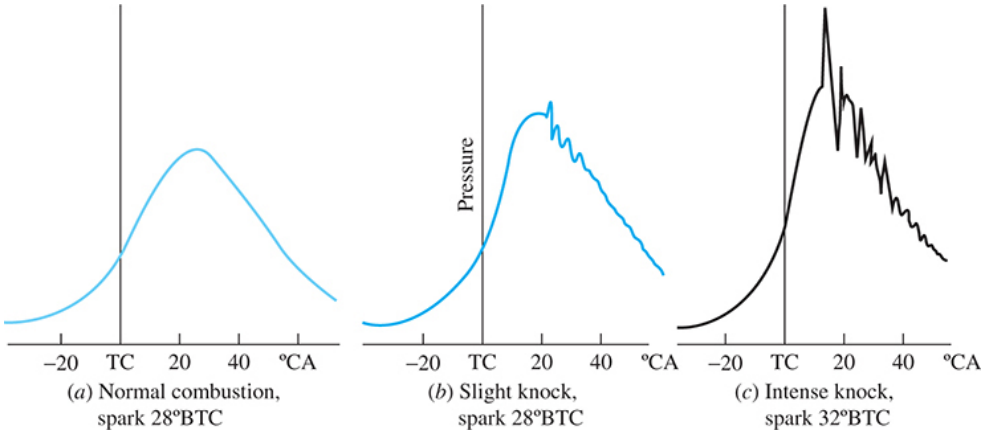


Figure 9.52 Cylinder pressure versus crank-angle traces of individual cycles with (*a*) normal combustion, (*b*) light knock, and (*c*) heavy knock. 4000 rev/min, wide-open-throttle, 381-cm ^{3} displacement single-cylinder engine. [65]

Figure 9.53 illustrates this sequential process—end-gas autoignition leading to audible noise. If the chemical energy release during autoignition of

the end gas is large and rapid enough, then pressure oscillations of a few or more bar in amplitude are set up inside the cylinder. As explained above, these pressure waves originate from the high local pressure generated in the end-gas by autoignition. [i] Pressure measurements at various locations in the combustion chamber vary in amplitude and phasing, and have several different specific frequencies—see below. These pressure waves cause the block (and cylinder head) to vibrate as indicated in Fig. 9.53 by the output from an accelerometer on the block outer wall. The vibrating block (and head) radiates noise which is picked up by a microphone about 10 cm from the cylinder head outer wall (this air gap explains the delay in the knock sound wave arriving at the microphone). The in-cylinder pressure waves take one or a few oscillations to establish themselves: they then decay since the driving process—autoignition of the end gas—is over.

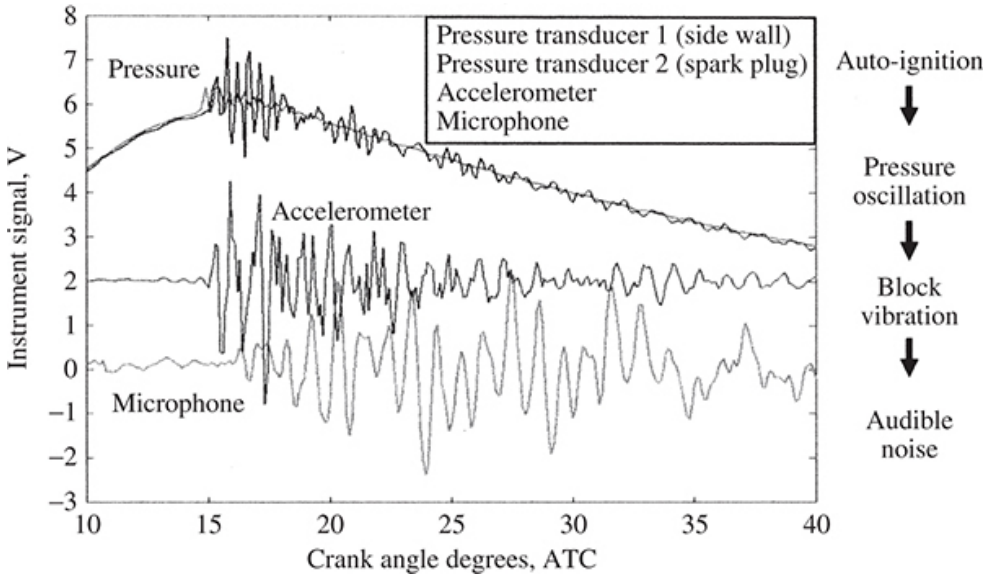


Figure 9.53 How autoignition-produced pressure oscillations within the engine cylinder produce block wall vibration (measured by an accelerometer) which produces noise (microphone, some 10 cm away from block wall).[66]

The interpretation of the in-cylinder pressure signal depends on the pressure transducer location and the acoustic modes excited by the autoigniting end-gas region. Draper [67] analyzed the resonant acoustic modes of a disk-shaped combustion chamber. More recent analysis (e.g., Ref. 68)

adds to his findings. The lowest frequency mode is the first circumferential mode at about 6 kHz for normal automotive size engine cylinders ($B \approx 85$ mm): this is the basic knock frequency that we hear. Next in frequency is the second circumferential mode at about 10 kHz, a frequency that is barely audible to the human ear. Several modes have frequencies in the 18 to 20 kHz range. [68]

Obvious questions at this point are how should we define knock onset, and once knock occurs what is its *severity* or *intensity*? The combined physical and chemical process is complex and many engine and fuel factors are involved, as also are ambient conditions. The transition from normal non-knocking combustion to knocking combustion is just that—a transition. As spark timing is advanced, the pressure levels reached during combustion increase. As a consequence, the temperatures attained in the end gas increase as does their maximum value (at peak pressure, some three quarters of the way through the combustion process). At a certain end-gas temperature and pressure, autoignition starts to occur releasing a small fraction of the chemical energy in the fuel molecules ahead of the flame it as propagates across the combustion chamber. Prior to knock onset, the rate of energy release is slow. As the spark timing continues to be advanced, the autoignition energy release rate and amount increase until it is rapid enough to generate high local pressures that set up strong pressure waves. As the amplitude of these pressure waves increase, block and cylinder head vibrations generate sound waves that become audible, and then louder as the autoignition process speeds up and spreads, further increasing the pressure levels generated in the end-gas region. Thus several measures—amplitude of in-cylinder pressure oscillations, extent of block and head vibration, noise level generated—can be used to define knock onset (often called borderline or audible knock), as described below.

Several methods of knock detection and characterization have been used. The human ear is a surprisingly sensitive knock detector and is routinely used in determining the *octane requirement* of an engine—the required fuel quality the engine must have to avoid knock. Knock detectors used for knock control systems normally respond to the vibration-driven acceleration of parts of the engine block caused by knocking combustion pressure waves. A high-intensity flash is observed when knock occurs; this is accompanied by a sharp increase in ionization. Optical probes and ionization detectors have therefore been used. The spark plug can serve as an ionization detector. For

more detailed studies of knock in engines, the piezoelectric pressure transducer is the most useful monitoring device. Often the transducer signal is filtered so that the pressure fluctuations caused by knock are isolated.

The amplitude of the pressure fluctuation is a useful measure of the intensity of knock because it depends on the amount of end-gas which ignites spontaneously and rapidly, and because engine damage due to knock results from the high gas pressures (and temperatures) in the end-gas region. Use of this measure of knock severity or intensity shows there is substantial variation in the extent of knock, cycle-by-cycle. Figure 9.54 shows this pressure-fluctuation intensity in one hundred consecutive cycles in a given cylinder of a multi-cylinder engine operating at fixed conditions for knocking operation. The intensity varies randomly from essentially no knock to heavy knock. [69] Cylinder-to-cylinder variations are also substantial due to variations in compression ratio, mixture composition and conditions, burn rate, and combustion chamber cooling. One or more cylinders may not knock at all while others may be knocking heavily.

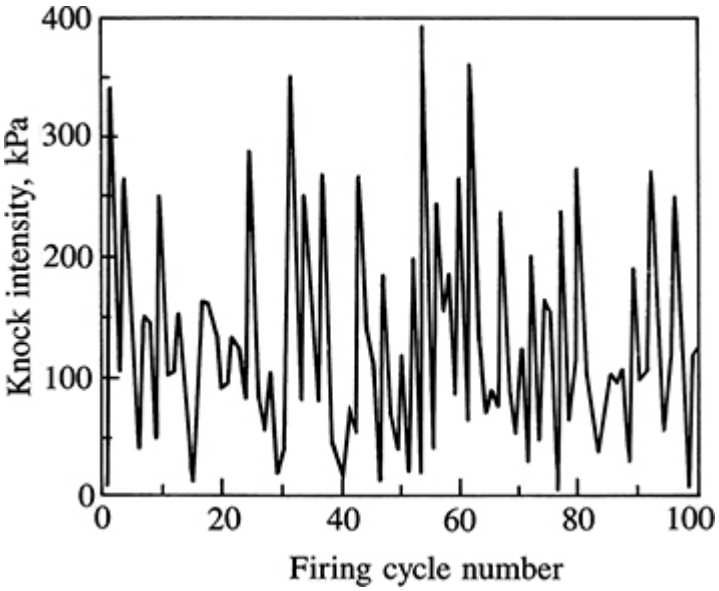


Figure 9.54 Knock intensity (maximum amplitude of bandpass-filtered pressure signal) in 100 individual consecutive cycles. One cylinder of V-8 engine, 2400 rev/min, wide-open throttle. [69]

Since the knock phenomenon produces a nonuniform state in the cylinder, and since the details of the knock process in each cycle and in each cylinder

are different, a fundamental definition of knock intensity or severity is difficult. The ASTM-CFR method for rating fuel octane quality (see Sec. 9.6.3) by the severity of knocking combustion uses the time derivative of pressure during the cycle. Cylinder pressure is determined with a pressure transducer. The low-frequency component of pressure change due to normal combustion is filtered out and the rate of pressure rise is averaged over many cycles during the pressure fluctuations following knock. This approach obviously provides only an average relative measure of knock intensity. The maximum rate of pressure rise has been used to quantify knock severity. An accelerometer mounted on the engine can give indications of relative knock severity provided that it is mounted in the same location for all tests. The most precise measure of knock severity is the maximum amplitude of the pressure oscillations that occur with knocking combustion. The cylinder pressure signal (from a high-frequency-response pressure transducer) is filtered with a band-pass filter so that only the component of the pressure signal that corresponds to the fluctuations occurring after knock remains. The filter is set for the first circumferential mode of gas vibration in the cylinder (in the 3 to 10 kHz range, depending on bore and chamber geometry). The maximum amplitude of the pressure oscillations gives a good indication of the severity of knock. The knock intensities in the individual cycles shown in Fig. 9.54 were determined in this manner. Note that because the pressure fluctuations are the consequence of a wave propagation phenomenon, the location of the pressure transducer in relation to the location of the knocking end-gas, and the shape of the combustion chamber, will affect the magnitude of the maximum recorded pressure-fluctuation amplitude.

The impact of knock depends on its intensity and duration. Trace knock has no significant effect on engine performance or durability. Heavy knock can lead to extensive engine damage. In automobile applications, a distinction is usually made between “acceleration knock” and “constant-speed knock.” Acceleration knock is primarily an annoyance, and due to its short duration is unlikely to cause damage. Constant-speed knock, however, can lead to two types of engine damage. It is especially a problem at high engine speeds where knock is masked by other engine noises and is not easily detected. Heavy knock at constant speed can lead to:

- Preignition, if significant deposits are present on critical combustion chamber components. This could lead to runaway preignition.

- Runaway knock—spark-knock occurring earlier and earlier, and therefore more and more intensely. This soon leads to severe engine damage.
- Gradual erosion of regions of the combustion chamber walls, even if runaway knock does not occur.

The engine can be damaged by knock in different ways: piston ring sticking; breakage of the piston rings and lands; failure of the cylinder head gasket; cylinder head erosion; piston crown and top land erosion; piston melting and holing. Examples of component damage due to preignition and knock are shown in Fig. 9.55. [70–72]

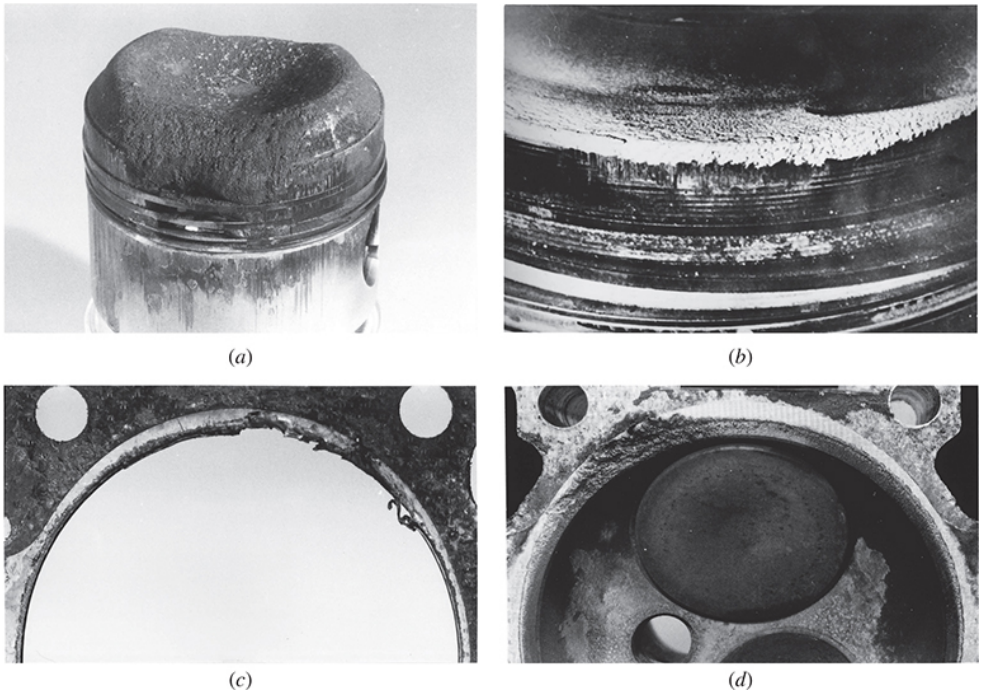


Figure 9.55 Examples of component damage from abnormal engine combustion: (*a*) piston holing by preignition;[71] (*b*) piston crown erosion after 10 hours of high-speed knocking;[70] (*c*) cylinder head gasket splitting failure due to heavy knock;[71] (*d*) erosion of aluminum cylinder head along the top of the cylinder liner due to heavy knock.[71]

The mechanisms that cause this damage are thought to be the following. Preignition damage is largely thermal as evidenced by fusion of spark plugs or pistons. When knock is very heavy, substantial additional heat is

transferred to the combustion chamber walls and rapid overheating of the cylinder head and piston results. Under these conditions, knock is not stable: the overheating increases the engine's octane requirement, which in turn increases the intensity of knock. It becomes heavier and heavier, and the uncontrolled running away of this phenomena can lead to engine failure in minutes. This damage is due to overheating of the engine: the piston and rings seize in the bore. The damage due to heavy knock over extended periods—erosion of piston crowns and (aluminum) cylinder heads in the end-gas region—is due primarily to the oscillating high gas pressures in this region. Extremely high-pressure pulses of up to 180 atm due to heavy knock can occur locally in the end-gas region, at around 6 kHz and in the 15 to 20 kHz frequency range. These high local fluctuating pressures, combined with the higher-than-normal local surface temperatures which occur with the higher knocking heat fluxes, weaken the material. Pitting and erosion due to fatigue with these excessive mechanical stresses, and breakage of rings and lands, can then occur. [70–73]

In practice, while knock onset continues to constrain spark-ignition engine design, and thus engine performance and efficiency (with a given fuel), actual knock damage to engines is increasingly rare. Several factors contribute to this trend; for example, use of knock sensors, more sophisticated control of engine operation, and tighter tolerances on fuel antiknock characteristics. However, knock continues to be a fundamental limitation since knock is viewed by vehicle users as objectionable, and the transition from light knock to heavy knock (which could cause engine damage) can occur with modest changes in engine operating condition (e.g., a few degrees change in spark timing).

9.6.2 Knock Fundamentals

Knock originates through autoignition of portions of the unburned mixture ahead of the flame when one or more localized regions in the end gas are compressed to sufficiently high pressures and temperatures to spontaneously react. Initially, the rate of chemical energy release as the fuel molecules start to oxidize is relatively slow, but that energy release rate then accelerates causing a rapid rise in local pressure and temperature setting up strong pressure waves which impact the surrounding mixture and flow. Thus "autoignition" does not occur uniformly throughout the end-gas, it originates

in local regions where the composition (concentrations of fuel vapor, air, burned residual gases) and temperature are such that the induction times (time for the oxidation chemistry to get started) are shorter due to these local inhomogeneities and variations in mixture thermal state. These local "easier to ignite" regions are termed exothermic centers. The transition of these exothermic centers into knock depends on their size, the thermal and concentration gradients at their boundaries, and their location in the end gas.[23]

Three modes of knocking combustion are observed, following autoignition of these exothermic centers:

1. *Deflagration*, the common mode, generally resulting from small exothermic centers with steep gradients, creating weak pressure oscillations: after ignition, flame propagation from these centers is similar to the normal SI engine flame propagation process.
2. *Thermal explosion,* characterized by larger centers with flatter gradients: pressure oscillations are moderate, and knock damage, if any occurs, is light.
3. *Developing detonation,* the most violent knock mode: occurring rarely, but generating large pressure oscillations with potential for significant surface damage, characterized by medium-sized exothermic centers with intermediate gradients which lead to strong pressure waves driven by the chemical energy release that these waves induce.

This engine phenomenon "knock" includes also the propagation of strong pressure waves across the chamber, chamber resonance, and transmission of sound through the engine structure. That detonation waves may be part of the intense knock phenomenon, has led many to call knock "detonation." However, the more general term "knock" is preferred, since the phenomenon includes more than autoignition-initiated rapid end-gas energy release.

Photographic studies of knocking combustion have been an important source of insight into the fundamentals of the phenomenon over the past 50 years. Figure 9.56 shows two sets of schlieren photographs, one from a cycle with normal combustion and the other from a cycle with knock.[74] Set (*a*) were taken in an overhead valve engine with a disc-shaped combustion

chamber, with a window which permits observation of the chamber opposite to the spark plug as shown. A reflecting mirror on the piston crown permits use of the schlieren technique, which identifies regions where changes in gas density exist. Operating conditions, except for spark advance, were the same for both cycles. In the normal combustion sequence, the turbulent flame front moves steadily through the end-gas as combustion goes to completion. The cylinder pressure varies smoothly throughout this process. When the spark is advanced by 15°, the end-gas temperature and pressure are increased significantly and knock occurs. In the second sequence of photographs, (*b*), the initial flame propagation process (photographs 1 to 3) is like that of the normal combustion sequence: then almost the entire region ahead of the flame appears dark (photograph 4). Between photographs 3 and 4 substantial changes in the density and temperature throughout most of the end-gas region have occurred. Examination of the cylinder pressure trace shows that this corresponds closely to the time when the pressure recorded by the pressure transducer rises rapidly. Immediately after this, pressure oscillations (at 6 to 8 kHz) are detected. In photograph 5 the flame is no longer visible: rapid end-gas expansion has disrupted the flame region and pushed it back out of the field of view. Subsequent photographs are alternatively lighter and darker, indicative of changing local density fields as pressure waves propagate back and forth across the chamber.

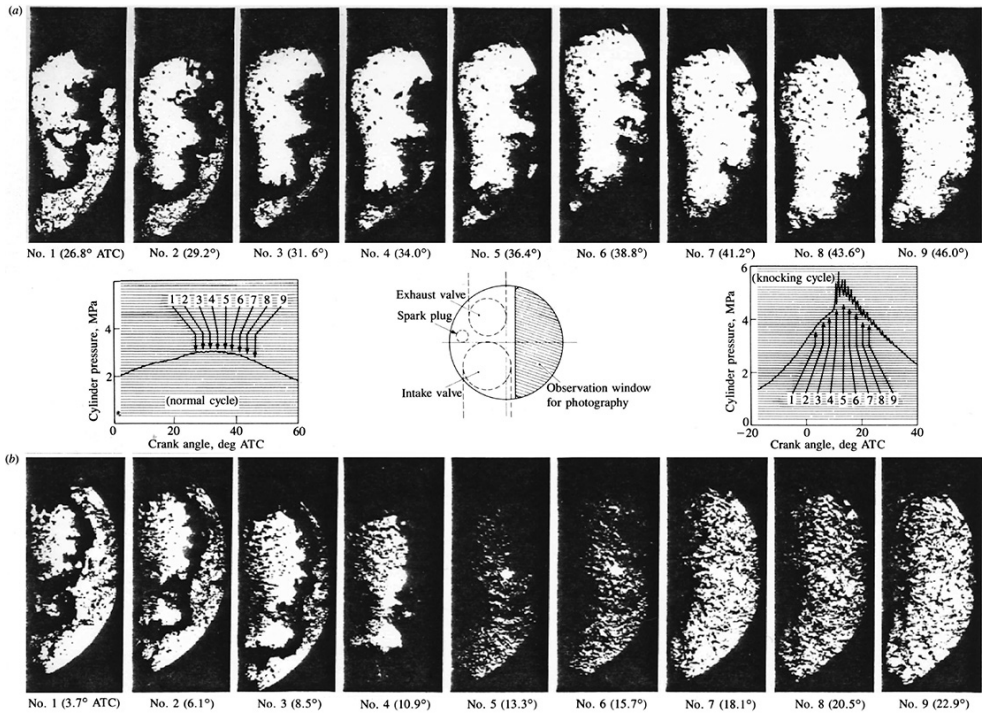


Figure 9.56 Schlieren photographs from high-speed movies of (*a*) normal flame propagation through the end gas and (*b*) knocking combustion (autoignition occurs in photograph 4), with corresponding cylinder pressure versus crank-angle traces. Disc-shaped combustion chamber with details of window shown in insert, 1200 rev/min, 80% volumetric efficiency, (*A/F*) = 12.5. Spark timing: (*a*) 10° BTC, (*b*) 25° BTC. [74]

More extensive studies of this type, which relate photographs from high-speed movies or videos of the combustion process to the cylinder pressure development, indicate that locations where ignition of one or more portions of the end-gas first occur, and the subsequent rate with which the ignition process develops throughout the rest of the end-gas, vary substantially cycle-by-cycle and with the intensity of the knocking process. Figure 9.57 shows five shadowgraph photographs from a knocking engine cycle in a research engine similar to that shown in Fig. 9.56. The photographs arc 33 μs apart; the total sequence shown lasts 1°. The first image shows the flame front prior to onset of knock. The second image shows the onset of autoignition with the appearance of dark regions (developing exothermic centers) near the wall (two identified by arrows) where substantial density gradients resulting from local energy release exist. The third, fourth, and fifth photographs show the spread of these ignited regions with time through the remaining end-gas. The exact location where autoignition occurred was identified with a photo digitizing system, which ranked regions of the photograph by their brightness or darkness. The digitized version of the second photograph is also shown in Fig. 9.57. Additional smaller regions of autoignition are evident near the wall and in the vicinity of the flame front. The location of autoignition sites varied with engine operating conditions. The rate of spread of the autoignited end-gas region also varied significantly. Under heavy knocking conditions, the entire end-gas region became ignited very rapidly and high-amplitude pressure oscillations occurred. Under trace knock conditions, autoignition could occur; yet the spread of the autoignited region be sufficiently slow so that only modest pressure oscillations occur.

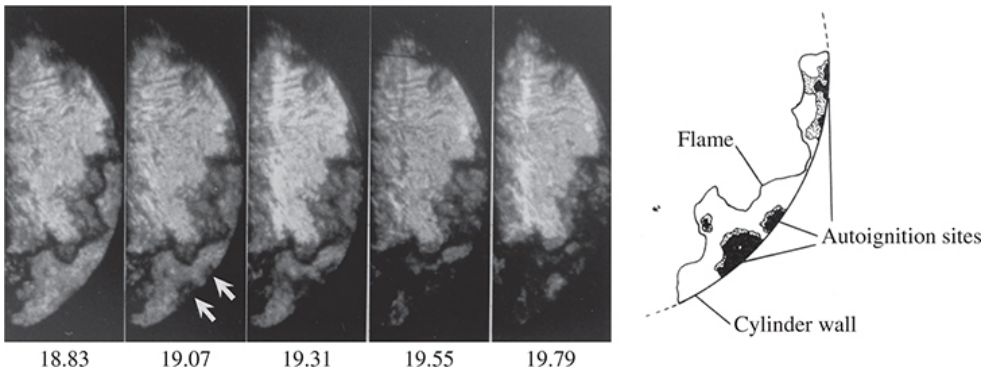


Figure 9.57 Five shadowgraph photographs of a knocking combustion cycle identifying location of autoignition sites (arrows). Crank angle of each photo indicated: 33 μs between frames. Operating conditions and engine details as in Fig. 9.56. Photo digitized picture of second photograph (on right) shows additional details of the autoignition sites in the end-gas region. [75]

The end-gas autoignition in these exothermic centers and the resulting spread of the burning process are illustrated by the data in Fig. 9.58. Data from simultaneous LIF images of formaldehyde, formed in the end-gas during the early stages of autoignition which burns up as exothermic centers (ETCs) develop and release their chemical energy, and images from ultra-high-speed schlieren movies which track the motion of the exothermic center boundaries have been combined. Of the several ETCs shown, most are like B where the center develops into a deflagration propagating outward at subsonic speed. However, center A transitions into a developing detonation propagating outward on its lower right-hand side at supersonic speeds, driven by the shock-wave-induced rapid unburned end-gas mixture chemical energy release.

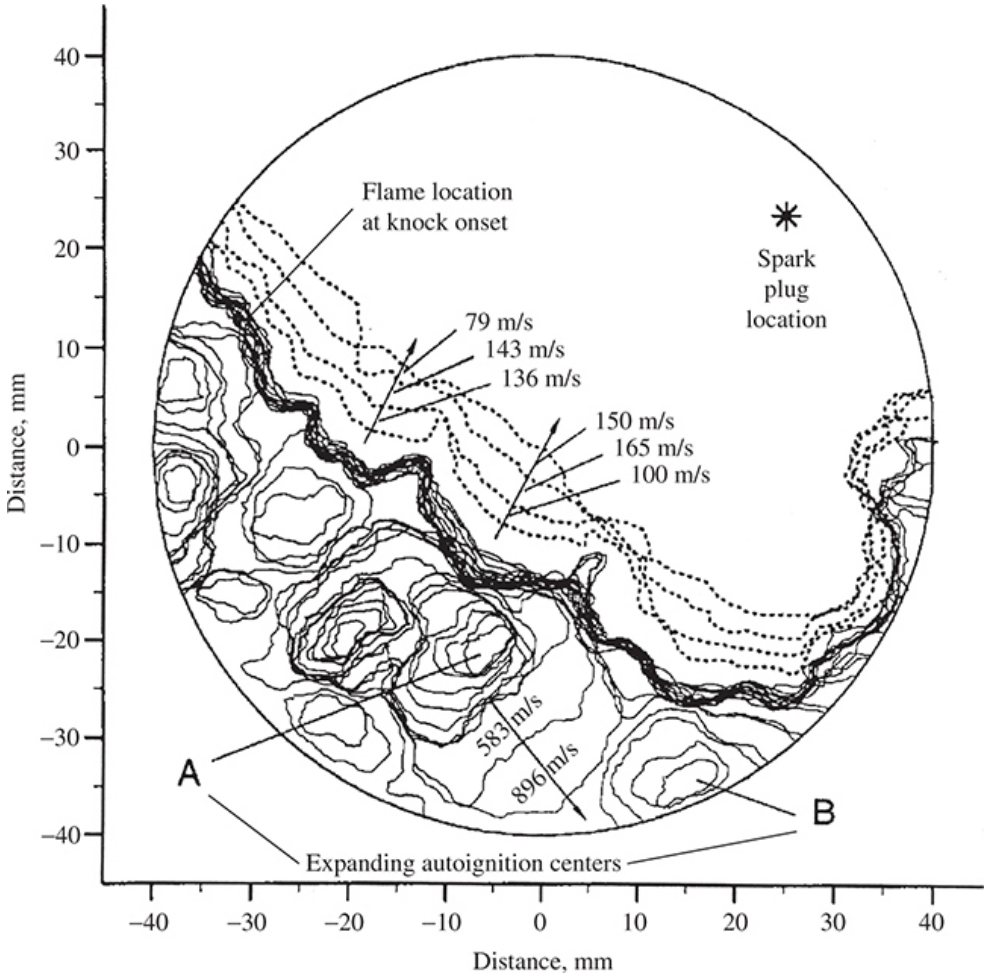


Figure 9.58 Tracings from an ultra-high-speed schlieren movie (750,000 frames/sec) showing formation and outward propagation rates of exothermic autoignition enters in the end-gas. Most centers (e.g., B) develop into deflagrations whereas the lower right-hand side of A transitions into a developing detonation wave. Note the irregular distribution of ECTs, and the pushing back of the regular flame front by the fast end-gas pressure rise. [23]

When the above-described end-gas ignition process occurs rapidly, the gas pressure in the end-gas region rises substantially due to the rapid release of the end-gas fuel's chemical energy. A strong pressure or shock wave propagates from the outer edge of this high-pressure end-gas region across the chamber at supersonic velocity, and an expansion wave propagates into the high-pressure region toward the near wall. The shock wave and expansion wave reflect off the walls of the chamber, producing standing waves of substantial amplitude. The amplitude of the pressure oscillations

builds up as the standing waves are established, and then decays as the wave-induced gas motion is damped out. The frequency of the pressure oscillations decreases with time as the initially supersonic pressure waves decay to sound waves. The pressure signal detected with a transducer during a knocking combustion cycle will depend on the details of the end-gas ignition process, the combustion chamber geometry, and the location of the transducer in relation to the end-gas region.

The pressure variation across the cylinder bore, due to knock, is illustrated in Fig. 9.59 for a the disc-shaped combustion chamber with the spark plug located in the cylinder wall as shown in Fig. 9.59 *a*. Figure 9.59*b* shows the pressure and temperature distribution across the combustion chamber due to the normal flame propagation process, at the time rapid release of the end-gas chemical energy suddenly occurs raising its pressure and temperature as shown. A strong pressure wave will now propagate to the right and an expansion wave to the left, as shown in the distance-time diagram in Fig. 9.59 *c*. These waves reflect off the walls and interact. The pressures at the cylinder wall, in the end-gas region and on the opposite side of the chamber at the spark plug, develop as shown in Fig. 9.59 *d*. In the end-gas region the pressure rises extremely rapidly when knock occurs, to a value considerably higher than that recorded on the opposite side of the chamber where the pressure rises more gradually. Standing waves are then set up and the amplitudes of the oscillations decay as the waves are damped out.

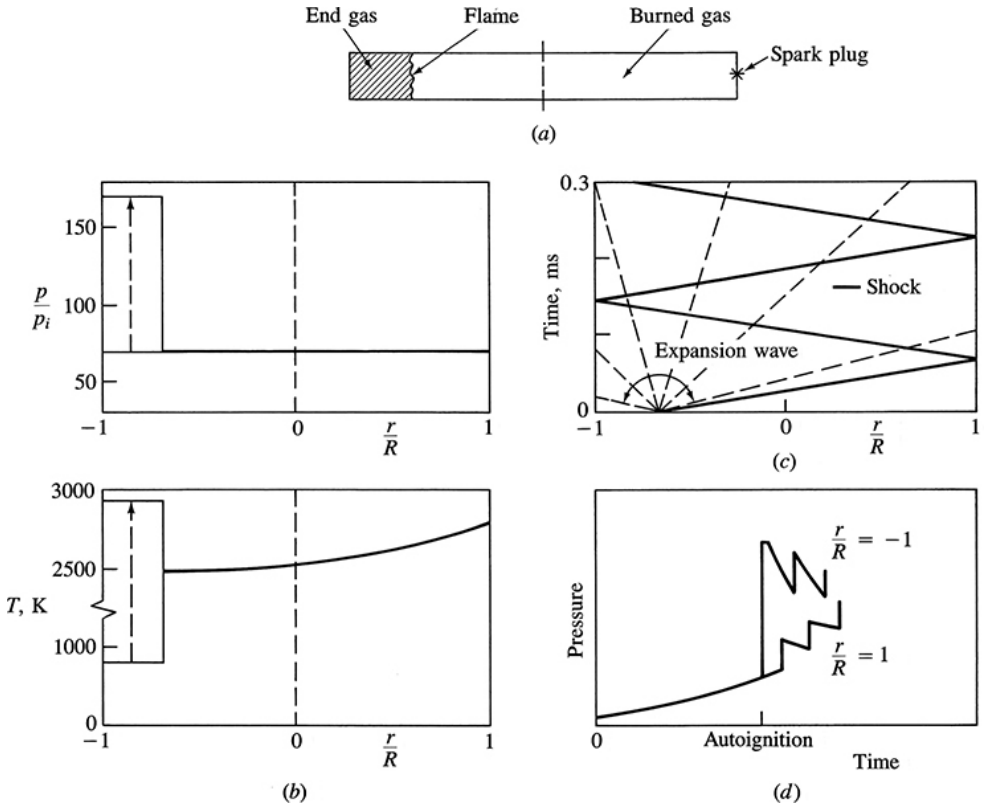
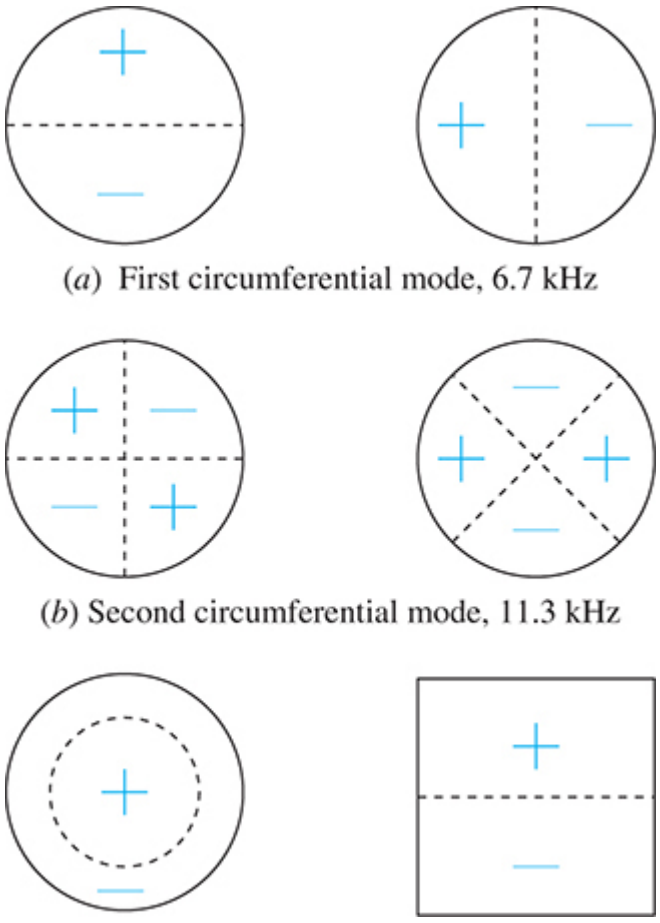


Figure 9.59 Illustration of how cylinder pressure distribution develops following knock onset: (*a*) schematic of disc-shaped combustion chamber at time of knock; (*b*) pressure and temperature across the diameter of a disc-shaped engine combustion chamber, before and after end-gas autoignition (assumed to occur very rapidly, at constant end-gas density); (*c*) schematic of shock and expansion wave pattern following end-gas autoignition on distance-time plot; (*d*) pressure variation with time at cylinder wall in the end-gas region, and at the opposite side of the cylinder.

The resonant acoustic modes of the combustion chamber define the frequencies of these pressure oscillations. Figure 9.60 shows some of these acoustic modes for a simple circular-cylindrical cavity (which approximates an engine combustion chamber) for which analytical solutions are known.[67] The first and second circumferential modes and the first radial and first axial modes of such a cylindrical cavity are depicted in Fig. 9.60*a*, *b*, *c*, and *d*, respectively. The dashed lines indicate nodes and the plus and minus signs identify the relative phases of the anti-nodes. The frequencies of the axial modes depend on crank angle, but the circumferential and radial modes do

not. The lowest frequency (for automobile engines, about 6 kHz) is the first circumferential mode. The circumferential modes are always paired, and the paired nodal lines are orthogonal and for a right circular cylinder have the same frequency. With actual engine cylinder-head geometries, there is an asymmetry and the two frequencies can differ by up to 10%. Linear combinations of the modes in a pair can be used to create a mode with the same shape and frequency as either, but with the node lines rotated. The majority of the pressure wave energy is in the lowest mode.



(*a*) First circumferential mode, 6.7 kHz

(*b*) Second circumferential mode, 11.3 kHz

Figure 9.60 Schematic diagram of some of the acoustic modes of a right cylindrical cavity. Typical frequencies for a standard gasoline SI engine are shown. [68]

Figure 9.61 shows examples of cylinder pressure traces for individual cycles in a gasoline SI engine at fixed operating conditions, (1500 rev/min, high-load) at different spark timings. The engine had a pent-roof, four-valve,

central spark-plug location cylinder head. The piston was flat topped. The bore was 83 mm, the compression ratio 9.8. Figure 9.61 *b* shows a cycle at a spark timing corresponding to borderline knock. Figures 9.61 *a* and *c* show cycles with spark timings retarded and advanced by 2° CA, respectively. Figure 9.61 *d* shows the fast Fourier transform (FFT) power density spectrum for these three cycles. As the engine transitions through borderline knock, two ranges of frequencies are excited. The lower frequency is around 6 kHz and corresponds to the frequency of the first circumferential mode of the chamber ($f \approx a/(2B)$ where a is the sound speed in the burned gases, and B is the effective cylinder diameter). The single peak at this frequency is well defined, and the peak power density increases as knock becomes more severe. A second range of frequencies is observed between 15 kHz and 22 kHz, and there is no single well-defined peak. This could be the first axial mode whose frequency varies with piston position, and thus crank angle. It could be higher order circumferential modes. [66] The higher frequency modes do not contribute to the audible noise knock generates because these frequencies are above the range of human hearing.

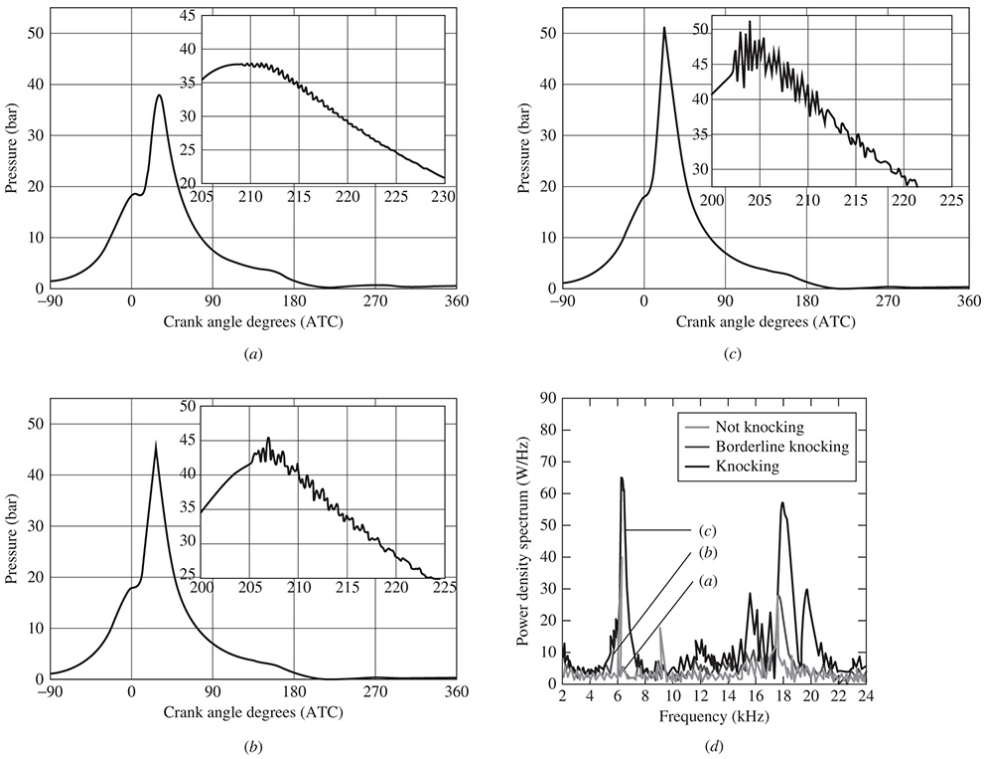


Figure 9.61 Cylinder pressure traces for typical cycles of a gasoline [toluene reference fuel (95 RON)], SI engine operating at 1500 rev/min and

high load, with a stoichiometric mixture, with a spark timing corresponding to borderline knock (*b*); also with spark timing retarded by 2° such that the engine is not knocking (*a*); and with spark timing advanced by 2° so it knocks more heavily (*c*). The region around peak pressure for each cycle is enlarged to show the extent of the pressure oscillations. (*d*) The fast Fourier transform of the three cycles shown. [66]

In summary, the fundamental explanation of knock described above is based on the concept of local autoignition of the fuel-air mixture in the end-gas. *Autoignition* is the term used for a rapid combustion reaction which is not initiated by any external ignition source. In the basic combustion literature this phenomenon is sometimes called an "explosion." Autoignition of a gaseous fuel-air mixture occurs when the chemical energy released by reactions as thermal energy is larger than the thermal energy lost to the surroundings by heat transfer: as a result, the temperature of the mixture increases rapidly due to the exponential temperature dependence of the rates of the chemical reactions involved. The state at which such spontaneous ignition occurs is often characterized by the *self-ignition temperature,* and the resulting self-accelerating event where the gas pressure and temperature increase rapidly is termed a *thermal explosion.*

In complex reacting systems such as exist in combustion, the "reaction" is not a single- or even a few-step process; the actual chemical mechanism consists of a large number of simultaneous, interdependent reactions called *chain reactions.* In such chains there is an *initiating* reaction where highly reactive intermediate species or *radicals* are produced from stable molecules (fuel and oxygen). This step is followed by *propagation* reactions where radicals react with the reactant molecules to form products and other radicals to continue the chain. The process ends with *termination* reactions where the chain propagating radicals are removed. Some propagating reactions produce two reactive radical molecules for each radical consumed. These are called *chain-branching* reactions. When, due to chain-branching, the number of radicals increases sufficiently rapidly, the reaction rate becomes extremely fast and a *chain-branching explosion* occurs. While the terms thermal and chain-branching explosions have been introduced separately, in many situations the self-accelerations in temperature and radicals occur simultaneously and the two phenomena are combined. [76]

Let us now turn to the autoignition chemistry of hydrocarbon-air mixtures. The process by which a hydrocarbon is oxidized can exhibit four different types of behavior, or a sequential combination of them, depending on the pressure and temperature of the mixture: slow reactions; single or multiple cool flames (slightly exothermic reactions); two-stage ignition (cool flame followed by a hot flame); single-stage ignition (hot flame). Slow reactions are a low-pressure, low-temperature (<200°C) phenomenon not normally occurring in engines. At 300 to 400°C one or more combustion waves can form, accompanied by faint blue light emission; the reactions are quenched, however, when only a small fraction of the reactants have reacted and the temperature rise is only tens of degrees. These are called *cool flames.* Depending on conditions and the fuel, a cool flame may be followed by a "hot flame" or high-temperature explosion where the reaction accelerates rapidly after ignition. This is termed *two-stage ignition.* As the temperature of the mixture increases, a transition from two-stage to single-stage ignition occurs. While all hydrocarbons exhibit induction intervals which are followed by a very rapid reaction rate, some hydrocarbon compounds do not exhibit the cool flame or two-stage ignition behavior.

Figure 9.62 shows these ignition limits for isooctane, methane, and benzene. For isooctane, ignition in the low-temperature region is by a two-stage process: there is a first time interval before the cool flame appears and then a second time interval from the appearance of the cool flame to the hot-flame combustion process. Ignition in the high-temperature region is by a continuous one-stage process. Cool flame phenomena vary significantly with hydrocarbon structure. Normal paraffins give strong cool flames, whereas branched-chain paraffins are more resistant. Olefins give even lower luminosity cool flames with longer induction periods. Methane shows only the high-temperature ignition limit, as indicated by Fig. 9.62. Benzene, also, does not exhibit the cool flame phenomenon, and other aromatics give hardly detectable luminosity. It is thought that some compounds knock by a low-temperature two-stage ignition mechanism, some via a high-temperature single-stage ignition mechanism, and for some fuels both mechanisms may play a role.

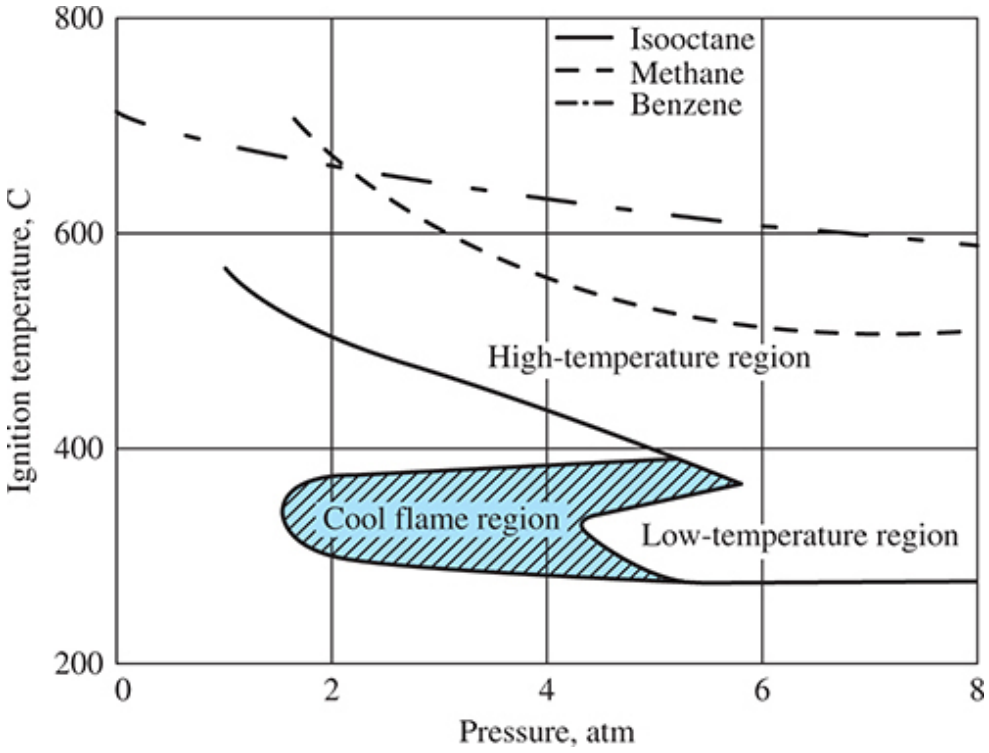


Figure 9.62 Ignition diagrams for isooctane, methane, and benzene. Two-stage ignition occurs in the low-temperature region; the first stage may be a cool flame. Single-stage ignition occurs in the high-temperature region.

Examples of these two mechanisms in rapid-compression machine experiments, where a homogeneous isooctane-air mixture was compressed to different final conditions in a piston-cylinder apparatus and allowed to autoignite, are shown in Fig. 9.63. $A \rightarrow B \rightarrow C$ is the piston-motion-produced compression. The top trace shows a well-defined cool flame at *D,* preceding hot ignition at *E.* The lower trace, at a higher temperature, shows a single-stage ignition process.

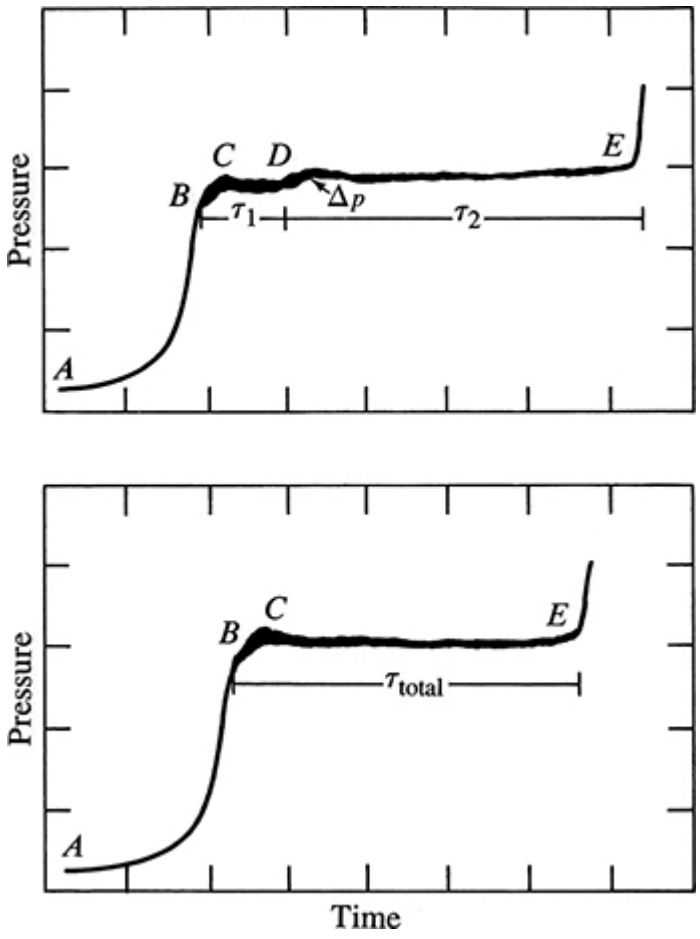


Figure 9.63 Pressure records of the autoignition of isooctane-air mixture ($\varphi = 0.9$) in a rapid compression machine. $A \rightarrow B \rightarrow C$ is the compression process. Top: two-stage ignition (at *D,* then at *E*) at post-compression pressure of 1.86 MPa and temperature of 686 K. Bottom: single-stage ignition at *E* at post-compression conditions of 2.12 MPa and 787 K. Vertical scale: 690 kPa/division. Horizontal scale: 1 ms/division. [77]

An outline of the basic hydrocarbon (RH) oxidation process due to Semenov is as follows: [76] it serves as a framework for discussing autoignition chemistry.

$$
\begin{aligned}
&\text{(R1)} && \mathrm{RH + O_2 \rightarrow \dot{R} + H\dot{O}_2} && \left.\vphantom{\mathrm{RH + O_2 \rightarrow \dot{R} + H\dot{O}_2}}\right\} \text{Chain initiation} \\
&\text{(R2)} && \mathrm{\dot{R} + O_2 \rightarrow RO_2} && \left.\vphantom{\begin{matrix}a\\b\\c\\d\\e\end{matrix}}\right. \\
&\text{(R3)} && \mathrm{R + O_2 \rightarrow olefin + HO_2} && \\
&\text{(R4)} && \mathrm{O + RH \rightarrow ROOH + \dot{R}} && \left.\vphantom{\begin{matrix}a\\b\\c\\d\\e\end{matrix}}\right\} \text{Chain propagation} \\
&\text{(R5)} && \mathrm{RO \rightarrow R'CHO + R''O} && \left.\vphantom{\begin{matrix}a\\b\\c\\d\\e\end{matrix}}\right. \\
&\text{(R6)} && \mathrm{O + RH \rightarrow H_2O_2 + \dot{R}} && \left.\vphantom{\begin{matrix}a\\b\\c\\d\\e\end{matrix}}\right. \\
&\text{(R7)} && \mathrm{ROOH \rightarrow R\dot{O} + \dot{O}H} && \left.\vphantom{\begin{matrix}a\\b\end{matrix}}\right\} \text{Degenerate branching} \\
&\text{(R8)} && \mathrm{R'CHO + O_2 \rightarrow R'\dot{C}O + H\dot{O}_2} && \left.\vphantom{\begin{matrix}a\\b\end{matrix}}\right. \\
&\text{(R9)} && \mathrm{R\dot{O} \rightarrow \text{destruction}} && \left.\vphantom{\mathrm{R\dot{O} \rightarrow \text{destruction}}}\right\} \text{Chain termination}
\end{aligned}
\tag{9.65}
$$

The dot denotes a radical; each prime denotes a free bond on the organic radical R.

Reaction R1 is slow and explains the induction period in hydrocarbon combustion. R2 is fast and of near-zero activation energy. R3 leads to olefins known to occur in the oxidation of saturated hydrocarbons. R4 and R5 yield the main intermediates. The degenerate branching comes about from the delay in decomposition of the reactive species in R7 and R8. As one radical is used up to form the reactants in R7 and R8, the multiple radicals do not appear until these reactants decompose. [76, 78]

The following indicates the relevance of the above mechanism to knock in engines. End-gas sampling studies have identified products of slow combustion reactions of isooctane; these principally include olefins, cyclic ethers, aldehydes (R′CHO), and ketones (R″CO). [79] Such studies have shown increasing concentrations of peroxides (predominantly H_2O_2 with traces of organic peroxides) with isoparaffinic fuels which show two-stage ignition behavior. Higher temperature, single-stage ignition fuels such as benzene and toluene gave no detectable peroxide. Aldehydes and ketones have been measured in significant and increasing concentrations in motored engines where the peak cycle temperature was steadily increased. In motored engines, the occurrence of cool flames, the two-stage nature of the autoignition ignition process at intermediate compression temperatures, and the transition to a single-stage ignition process at very high compression ratios (i.e., motored-engine gas temperatures with a peak much higher than normal to simulate end-gas conditions in a firing engine) have also been demonstrated. [80]

Two types of models of this autoignition process have been developed and used: (1) empirical induction-time correlations and (2) chemical mechanisms which embody many or all of the features of the "full" hydrocarbon oxidation process outlined in Eq. (9.65) .

Induction-time correlations are derived by matching an Arrhenius function to measured data on induction or autoignition times, for given fuel-air mixtures, over the relevant mixture pressure and temperature ranges. It is then assumed that knock-initiating autoignition occurs when

$$\int_{t=0}^{t_i} \frac{dt}{\tau} = 1 \tag{9.66}$$

where t is the induction time at the instantaneous temperature and pressure for the mixture, t is the elapsed time from the start of the end-gas compression process ($t = 0$), and ti is the time at autoignition. This equation [j] can be derived by assuming that the overall rate of production of the critical species in the induction period chemistry, for a given mixture, depends only on the gas state and that the concentration of the critical species required to initiate autoignition is fixed (i.e., independent of the gas state). [81]

A number of empirical relations for induction time for individual hydrocarbons and blended fuels have been developed from fundamental or engine studies of autoignition. These relations have the form

$$\tau = Ap^{-n} \exp\left(\frac{B}{T}\right) \tag{9.67}$$

where A, n, and B are fitted parameters that depend on the fuel. The most extensively tested correlation is that proposed by Douaud and Eyzat: [82]

$$\tau = 17.68\left(\frac{\mathrm{ON}}{100}\right)^{3.402} p^{-1.7} \exp\left(\frac{3800}{T}\right) \tag{9.68}$$

where τ is in milliseconds, p is absolute pressure in bar, and T is in kelvins. ON is the research octane number of the fuel (see Sec. 9.6.3). If the temperature and pressure time history of the end-gas during an individual cycle are known, Eqs. (9.64) and (9.66) together can be used to determine whether autoignition occurs before the normally propagating flame consumes

the end-gas. Use of this approach for predicting knock onset can be surprisingly accurate. [83]

An important requirement with any model of the end-gas autoignition process is characterizing the end-gas temperature. During intake, the combustion chamber walls are hotter than the entering gases: thus, heat is transferred from the walls to the in-cylinder mixture. Midway during the compression stroke, the mixture temperature rises to levels above the wall temperature and a thermal boundary layer will build up adjacent to the wall, as heat is now transferred to the wall. Unburned mixture away from the wall, however, will be compressed essentially adiabatically (see Sec. 12.6.5). In addition, any unburned mixture which for some portion of time during intake and compression has been in close proximity to the exhaust valve and piston (which run at higher temperatures than the water-cooled wall regions) could be at higher temperatures than adiabatically compressed mixture due to substantial heat transfer to the unburned mixture. Thus, the end-gas temperature is not uniform and the distribution of temperature is complex. Often, the *mean* unburned mixture temperature is used to characterize its state. Alternatively, the *core* temperature corresponding to adiabatic compression of mixture from conditions at the start of compression is used. In the absence of substantial heating by the exhaust valve and piston, the core temperature is a better representation of the maximum unburned mixture at any point in the cycle.

More complex and complete chemical models of the autoignition process have been developed for specific paraffinic hydrocarbon and alcohols, which are important components in real fuels. However, detailed models are not yet available for fuels with multiple components such as gasolines. A generalized kinetic model for hydrocarbon oxidation based on a degenerate branched-chain mechanism, known as the Shell model, has been developed and tested with some success. The model uses generic chemical entities representative of a variety of individual species which undergo a set of generalized reactions. This is justified by the broadly similar (though complex) ignition behavior of a variety of different fuel molecules and the similar kinetics exhibited by the organic radicals of the same type in the hydrocarbon oxidation process [Reaction set (9.65)]. [77, 84]

The Shell model is based on a generic eight-step degenerate chain-branching reaction scheme. The scheme involves the fuel (RH), oxygen, radicals formed from the fuel (R), products (P), intermediate product (Q),

and degenerate-branching agent (B). The rate constants are either fixed at values consistent with the literature or fitted so that measured induction times (such as those illustrated in Fig. 9.63) are adequately predicted. An example of results obtained with this scheme is shown in Fig. 9.64. It shows the calculated pressure, temperature, and species concentrations of R, Q, and B in the end-gas region of an operating spark-ignition engine, leading up to knock at 209° ABC. In this example, the autoignition model has been incorporated in a multidimensional model of the flow and flame propagation processes within the combustion chamber (see Sec. 15.5).[85]

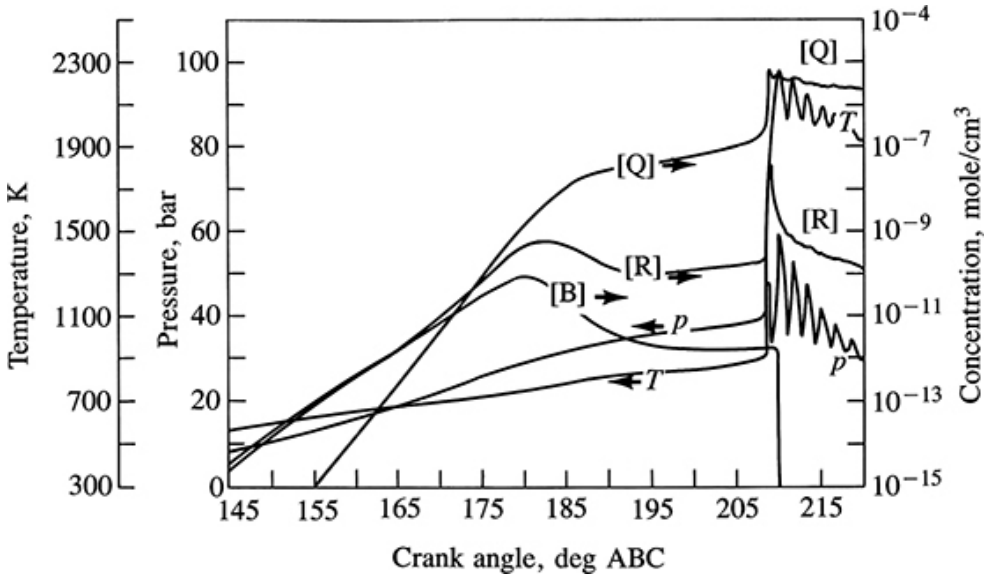


Figure 9.64 Pressure, temperature, and composition in the end-gas, before, during, and after autoignition, predicted by the Shell model in a spark-ignition engine combustion process. B, Q, R defined in text. [85]

9.6.3 Fuel Factors

The tendency to knock depends on engine design and operating variables which influence end-gas temperature, pressure, and the time spent at high values of these two properties before flame arrival. Thus, for example, the tendency to knock is decreased through reductions in the end-gas temperature that follow from decreasing the inlet air temperature, or retarding the spark from MBT timing. However, knock is a phenomenon that is governed by both

engine and fuel factors: its presence or absence in an engine depends significantly on the "antiknock" quality of the fuel.

Individual hydrocarbon compounds vary substantially in their ability to resist knock, depending on their molecular size and structure. Their tendency to knock has been characterized by the *critical compression ratio* of an engine: i.e., the compression ratio at which, under specified operating conditions, the specific fuel compound will exhibit incipient knock. Knocking tendency is related to molecular structure [k] as follows: [86, 87]

Paraffins

- Increasing the length of the carbon chain increases the knocking tendency.
- Compacting the hydrocarbon molecule by incorporating side chains (thereby shortening the length of the basic chain) decreases the tendency to knock.
- This adds methyl groups (CH_3) to the side of the basic carbon chain, in the second from the end to the center position, reducing the number of weaker C-H bonds: see below.

Olefins

- The introduction of one double bond has little antiknock effect; two or three double bonds generally result in appreciably less knocking tendency.
- Exceptions to this rule are acetylene (C_2H_2), ethylene (C_2H_4), and propylene (C_3H_6), which knock much more readily than the corresponding saturated hydrocarbons.

Napthenes and aromatics

- Napthenes have significantly greater knocking tendency than have the corresponding size aromatics.
- Introducing one double bond has little antiknock effect; two and three double bonds generally reduce knocking tendency appreciably.
- Lengthening the side chain attached to the basic ring structure increases the knocking tendency in both groups of fuels, whereas branching of the side chain decreases the knocking tendency.

Figure 9.65 identifies the magnitude of these trends on a plot of the critical compression ratio against the number of carbon atoms in the

molecule. The strong dependence of knocking tendency on fuel molecular size and structure is apparent. The location of the various hydrocarbon compounds on Fig. 9.65 can be explained in part through the hydrogen to carbon atom bond arrangements for each molecule. The weakest hydrogen bonds (and thus the most easily broken as the hydrocarbon molecule starts to react by losing hydrogen atoms) are those between a hydrogen atom and a carbon atom with two C-to-C bonds. The n-heptane molecule (which has a critical compression ratio of 2.2) has ten of these weaker bonds. Isooctane (2-2-4, trimethylpentane), with only two of these bonds, has a critical compression ratio of 6.4, and resists autoignition well.

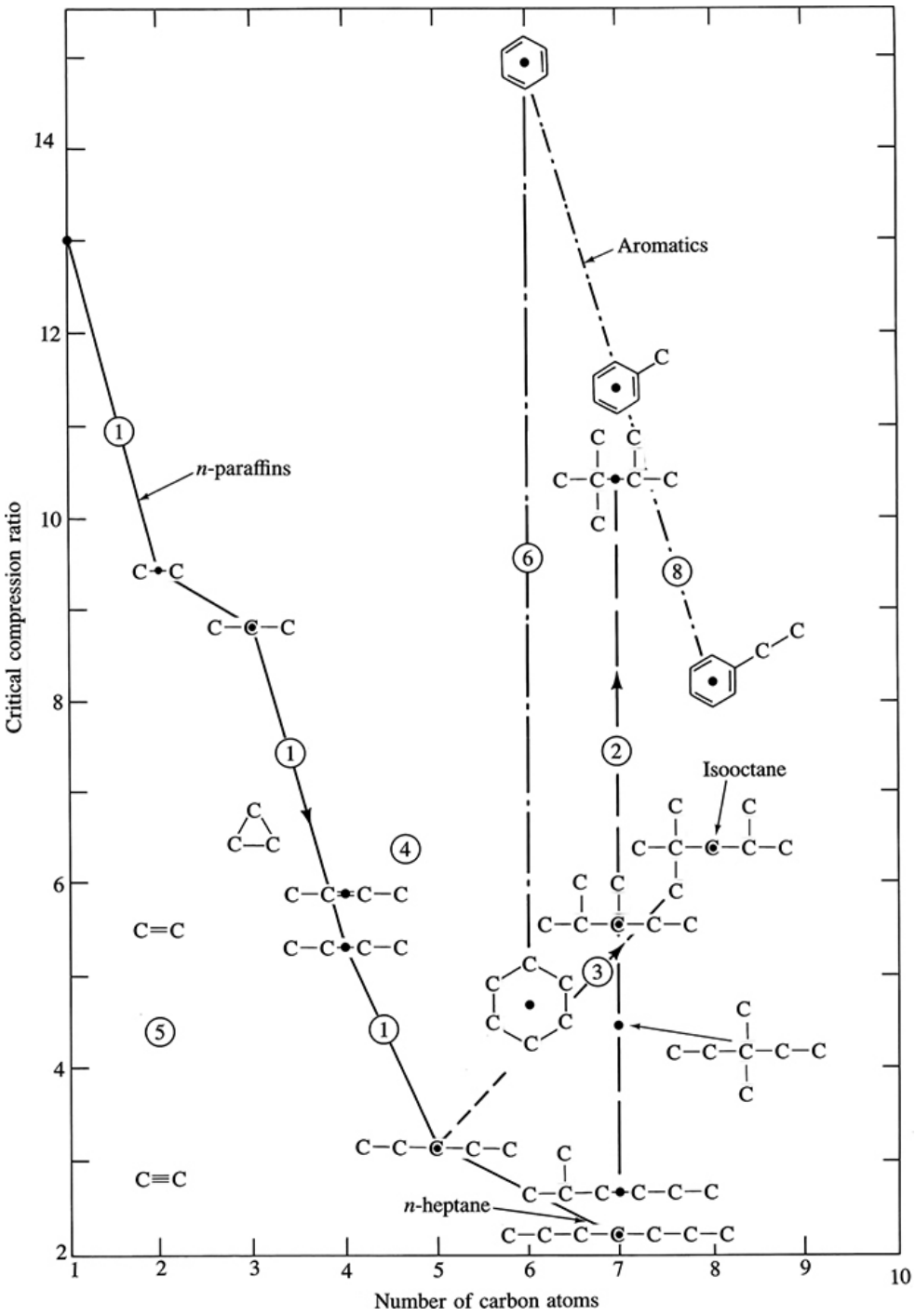


Figure 9.65 Critical compression ratio (for incipient knock at 600 rev/min and 450 K coolant temperature) as a function of number of carbon atoms in hydrocarbon molecule illustrating the effects of changes in molecular structure. (*Developed from Lovell.* [86])

Practical fuels are blends of a large number of individual hydrocarbon compounds from all the classes of hydrocarbons: alkanes (paraffins), cyclanes (napthenes), alkenes (olefins), and aromatics (see Sec. 3.3 , and alcohols). A more practical measure of a fuel's resistance to knock is obviously required: such a property is the fuel's *octane number.* It determines whether or not a fuel will knock in a given engine under given operating conditions: the higher the ON, the higher the resistance to knock. ON is not a single-valued quantity since it depends on engine design and operating conditions during testing. The standard ON scale is based on two hydrocarbons which define the ends of the scale. By definition, normal heptane (*n*-C_7H_{16}) has a value of zero and isooctane (C_8H_{18}, 2,2,4-trimethylpentane) has an ON of 100. These hydrocarbons were chosen because of the great difference in their ability to resist knock and the fact that isooctane had a higher resistance to knock than the gasolines available at the time the scale was established. Blends of these two hydrocarbons to form *Primary Reference Fuels* define the knock resistance of intermediate ONs: for example, a blend of 10% *n*-heptane and 90% isooctane has an ON of 90. A fuel's ON is measured by determining the PRF, the blend of these two hydrocarbons that matches that fuel's knock resistance in a specific spark-ignition engine under the same operating conditions.

Several standard octane rating methods for fuels have been developed. Two of these—the research method (ASTM D-2699) [l] and the motor method (ASTM D-2700)—are carried out in a standardized single-cylinder engine. In the motor method, the engine operating conditions are more severe; that is, the conditions are more likely to produce knock. In addition, road octane rating methods have been developed to define the antiknock quality of fuels in cars operated on the road or on chassis dynamometers. The engine used in the ASTM research and motor methods is the single-cylinder engine developed under the auspices of the Cooperative Fuel Research Committee in 1931—the CFR engine. [m] This test engine is a robust four-stroke overhead-valve engine with an 82.6-mm (3.25-in.) bore and 114.3-mm (5-in.) stroke. The compression ratio can be varied from 3 to 30 while the engine is operating, with a mechanism which raises or lowers the cylinder and cylinder head assembly relative to the crankcase. The engine is equipped with several fuel reservoirs, with a selector valve, so the engine can be operated on any of the three fuels: two reference fuels (usually blends of *n*-heptane and isooctane) and the fuel being rated. The engine operating

conditions of the research and motor methods are summarized in Table 9.7. The test conditions were chosen to represent the engine operating regime where knock is most severe.

TABLE 9.7 Operating conditions for research and motor methods

	Research method	Motor method
Inlet temperature	52°C (125°F)	149°C (300°F)
Inlet pressure	Atmospheric	
Humidity	3.6–7.2 g H_2O/kg dry air	
Coolant temperature	100°C (212°F)	
Engine speed	600 rev/min	900 rev/min
Spark advance	13°BTC (constant)	19–26° BTC (varies with compression ratio)
Air/fuel ratio	Adjusted for maximum knock	

With the fuel under test, the fuel/air ratio is adjusted for maximum knock. The compression ratio is then adjusted to produce knock of a standard intensity. The level of knock obtained with the test fuel is bracketed by two blends of the reference fuels not more than two octane numbers apart (with one knocking more and one less than the test fuel). The octane number of the gasoline is then obtained by interpolation between the knock-meter scale readings for the two reference fuels and their octane numbers. For fuels below 100 ON, the *primary reference fuels* are blends of isooctane and *n*-heptane; the percent by volume of isooctane in the blend is the octane number. For fuels above 100 ON, the antiknock quality of the fuel is determined in terms of isooctane plus milliliters of the antiknock additive, tetraethyl lead, per U.S. gallon. [n]

The octane ratings of several individual hydrocarbon compounds and common blended fuels are summarized in App. D , Table D.4. Practically all fuels exhibit a difference between their research and motor octane numbers. The motor method of determining ON uses more severe operating conditions than the research method (higher inlet mixture temperature, higher speed, more advanced spark timing). Thus, the motor octane number (MON) is

lower than the research octane number (RON). The numerical difference between these two octane numbers is called the *fuel's sensitivity*:

$$\text{Fuel sensitivity } S = \text{RON} - \text{MON} \tag{9.69}$$

Fuel sensitivity varies with the composition of the fuel, and thus the source of crude petroleum and the refining process details. The primary reference fuels themselves (mixtures of isooctane and *n*-heptane), by definition have the same octane numbers by both the research and motor methods. Since the primary reference fuels are paraffins, we would expect other paraffins to have little or no sensitivity. In contrast, olefins and aromatics have high sensitivity, as do alcohols. Generally, therefore, straight-run gasolines containing high percentages of saturated hydrocarbons have low sensitivity, while cracked or reformed gasolines containing larger percentages of unsaturated hydrocarbons have high sensitivity. Regular (U.S.) grade unleaded gasoline typically has a RON of at least 91 and a MON of about 83, giving a sensitivity of 8 to 10 octane numbers. Standard grade European gasolines have higher RON and MON values, 95 and 85, respectively.

Research and motor octane number fuel ratings are made in a single-cylinder engine run at constant speed, wide-open throttle, and fixed spark timing. These methods do not always predict how a fuel will behave in an automobile engine operated under a variety of speed, load, and weather conditions. Several methods of rating a gasoline in actual vehicles, either on the road or on chassis dynamometers which duplicate outdoor road conditions have, therefore, been developed. These methods determine the fuel's *road octane number.* More broadly, a measurable engine parameter z related to autoignition [o] can be correlated at fixed engine operating conditions with a fuel's knock resistance through [88]

$$z = a(\text{RON}) + b(\text{MON}) + c$$

Note that a, b, and c are experimentally derived constants that depend on the engine used and its operating conditions (e.g., speed and load). More generally, the propensity of a fuel to autoignite and knock under real world conditions is quantified by its *octane index*, OI. The OI of a fuel is obtained in an engine operating under specified conditions, by determining the primary

reference fuel composition that matches the knock onset behavior of the actual fuel. This OI can be related to the fuel's RON and MON using a weighting factor K: [88, 89]

$$\mathrm{OI} = (1-K)\mathrm{RON} + K\,\mathrm{MON} = \mathrm{RON} - KS \tag{9.70}$$

K is assumed to be independent of the fuel, and depends only on the engine operating conditions. Note that if $K = 0$ the OI corresponds to the fuel's RON: $K = 1$ corresponds to OI = MON. In the United States, the OI posted on retail gasoline pumps is (RON + MON)/2 termed the *antiknock index*. Thus $K = 0.5$ is used to characterize a fuel's "on-the-road" ON.

Engines, vehicles, and fuels have all changed significantly since the mid-1900s. Recent analyses have evaluated this octane weighting factor K over the past 60 years using data from vehicle-fleet studies where, usually, the engine-in-vehicle fuel octane requirement to avoid knock was determined from maximum vehicle acceleration tests. The basic RON test was adopted in 1930. It was found that it did not adequately predict the antiknock performance of fuels in actual use. The MON test, with its thermally more demanding conditions, was then developed to better reflect the actual antiknock behavior of fuels. Thus, around 1930, K was close to unity. A historical review [90] indicates that K, on average, has steadily decreased over time to about 0.5 in 1950, 0.25 in 1970, and is now close to zero (with slightly negative values for some engines and operating conditions). Thus RON is now a better predictor of actual fuel antiknock behavior than combinations of RON and MON, and is used in Europe for this purpose.

It has also become clear that gasolines with higher sensitivity S [see Eq. (9.69)] with the same RON value usually have better knock resistance (a higher OI) than fuels of lower sensitivity. The key to understanding this is the unburned mixture temperature behavior, often characterized by the engine's intake air temperature at knock onset. For the RON and MON tests this air temperature is 52 and 149°C, respectively: for modern naturally-aspirated engines it is about 30°C, and for turbocharged engines about 40°C, significantly lower than the standard test values.

Most fuel components with high sensitivities have more complex molecular structures compared to paraffins. However, there is no simple relationship between sensitivity and the fuel's molecular structure. Figure 9.66 plots MON values against RON values for several different classes of

individual hydrocarbons. Non-sensitive fuels lie on the 45° line MON ≈ RON. These tend to be paraffins. Sensitive fuels lie below this line—MON is lower than RON. The higher sensitivity fuels are olefins, aromatics, diolefins, and napthenes. Alcohols (not included in the plot) have high sensitivities. [p] Note the large variations in RON and MON for each class of fuels. [91]

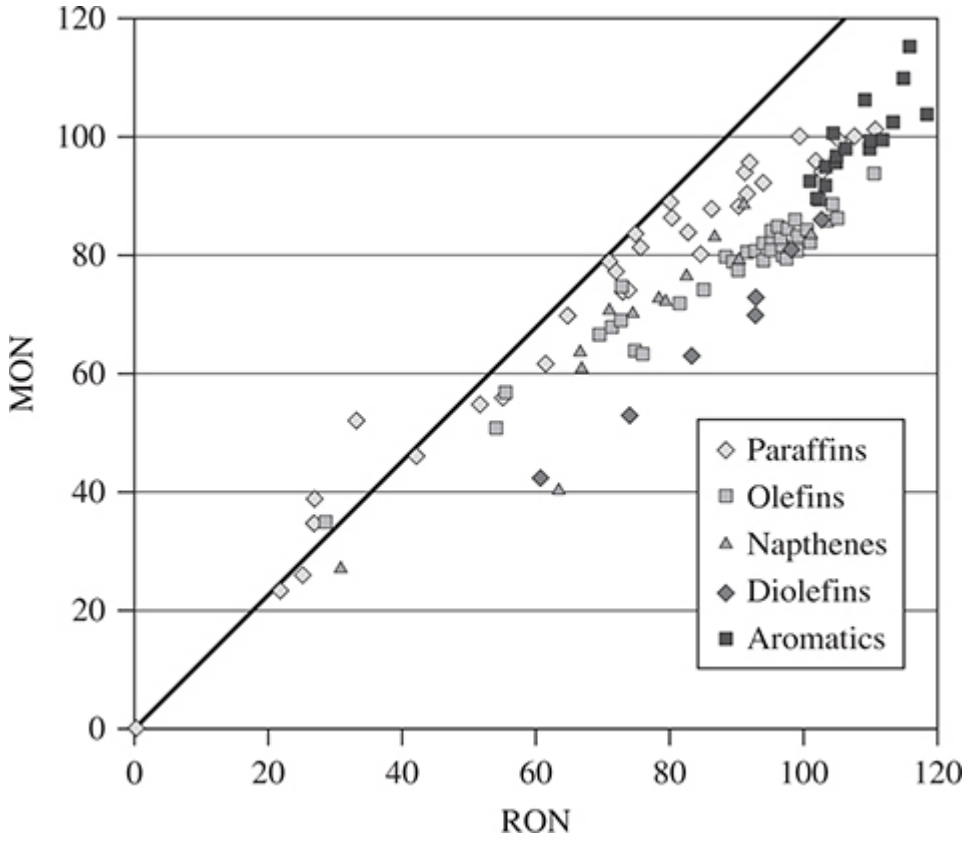


Figure 9.66 Motor octane number (MON) plotted versus research octane number (RON) for a variety of hydrocarbons including paraffins, olefins, napthenes, diolefins, and aromatics. Non-sensitive fuels are located on the diagonal black line and sensitive fuels are located below that line. Note there are large variations in RON and MON for each class of hydrocarbons. [91]

Studies of the effects of mixture temperature on the autoignition fuel chemistry explain why increasing fuel sensitivity raises the fuel's knock resistance in gasoline engines. Models with detailed chemical kinetic mechanisms for isooctane, n-heptane, toluene, and ethanol oxidation (with large numbers of species and reactions; see Sec. 14.5.6) were coupled with

engine operating cycle simulation-generated end-gas temperature and pressure profiles to explore the autoignition behavior of several fuels of increasing sensitivity blended from the above compounds. The crank angle at which autoignition occurred was computed as a function of intake air temperature. Results for three 96 RON fuels with different sensitivities are shown in Fig. 9.67. The air temperature in the RON test is 52°C and in the MON test is 149°C. Modern engines operate with air temperatures in the 30 to 40°C range. With air temperature below the RON test value (where the behavior of the different sensitivity fuels is closely comparable), the highest sensitivity fuel ($S = 17$) shows greater resistance to knock: the zero sensitivity fuel (PRF 96) has the lowest knock resistance. At air temperatures above 50°C or so, the sensitivity trend reverses—the least sensitive fuel shows the greatest knock resistance. [91]

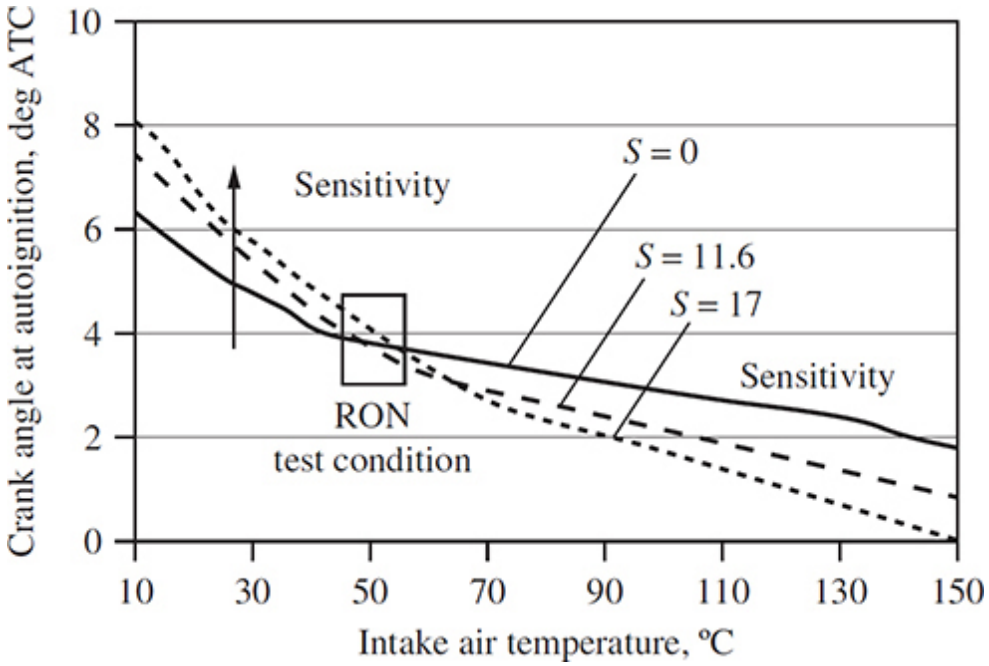


Figure 9.67 The computed crank angle of autoignition (in degrees ATC) as a function of intake air temperature for three fuels of different sensitivity, *S*: a primary reference fuel (RON 96), a toluene reference fuel (78% toluene, 22% *n*-heptane mixture with RON = 96 and $S = 11.6$) and a 63.5% ethanol, 36.5 *n*-heptane blend with RON = 96 and $S = 17$. The RON octane rating test condition is indicated by the box. [91]

For modern gasoline engines, the maximum end-gas temperature under knock-limited conditions is in the 800 to 950 K range. At the low end of this range, in the low-temperature combustion region where the autoignition chemistry is slower, higher sensitivity fuels perform better. At the higher end of this range ($Tu_{,max} \geq 900$ K), the high-temperature combustion region where chemistry is significantly faster, less sensitive fuels perform slightly better at resisting knock. In the center of the transition regime, in between (the critical knock onset region with today's gasolines) which corresponds to the "negative" temperature gradient ignition regime (see Fig. 9.62), autoignition induction times of fuels with different sensitivities but the same RON are similar. At end-gas temperatures below this mid-point (≈825 K) the steeper induction time dependence on temperature of the more sensitive gasolines, as the autoignition chemistry shifts from the negative-temperature region to the low temperature region, results in slower chemistry and thus greater knock resistance for the more sensitive fuels. [91]

Refiners and automobile manufacturers are interested in the octane requirement of engines in vehicles on the road. The *octane requirement* (usually abbreviated to OR) of an engine or vehicle-engine combination is defined as the minimum fuel octane number that will resist knock, at MBT spark timing, over a given engine speed and load range.

9.6.4 Sporadic Preignition and Knock

As in-cylinder pressures have risen due to increasing engine compression ratios and turbocharging, an additional knock phenomenon has been observed, initiated by a seemingly random preignition process. Sporadically, a local region of the in-cylinder unburned mixture spontaneously ignites toward the end of the compression stroke well before the spark discharge occurs. This preignition initiates combustion well before the intended start of the flame propagation process at time of spark discharge. As a result, in-cylinder gas pressures (and temperatures) rise sooner, attaining much higher values close to TC piston position than with a normal combustion event. The "end gas" under these conditions can autoignite rapidly leading to extremely heavy knock. [q] Due to this preignition, the combustion process in that engine cycle is very "over-advanced." This phenomenon has been broadly observed as in-cylinder pressures have risen, and its avoidance necessitates

constraining the pressure levels at which gasoline spark-ignition engines (especially turbocharged engines) can operate.

The phenomenon is sporadic, occurring once every 20,000 to 40,000 cycles, and lasts for only one or a few sequential cycles. The knock intensity produced (peak to peak amplitude of the knock pressure oscillations) can be up to one- to two-hundred bar, much higher than the knock intensity resulting from spark knock.

Figure 9.68 shows an example of this preignition leading to heavy knock phenomenon.[92] In-cylinder pressure records from a turbocharged gasoline engine, operating at high-load, 2000 rev/min and 2 bar intake pressure, with significantly retarded spark timing (2.3° CA before TC) are shown. A normal non-knocking cycle (N) without preignition has a low peak (and late) pressure (80 bar) as shown that is close to knock onset conditions for this engine. The next cycle shown (K) is one with fairly heavy knock (KI ~ 24 bar) with preignition occurring at 6° CA BTC. The third cycle (HK) shows preignition occurring much earlier at 19° CA BTC leading to "super knock" with a knock intensity (peak to peak) of about 150 bar.

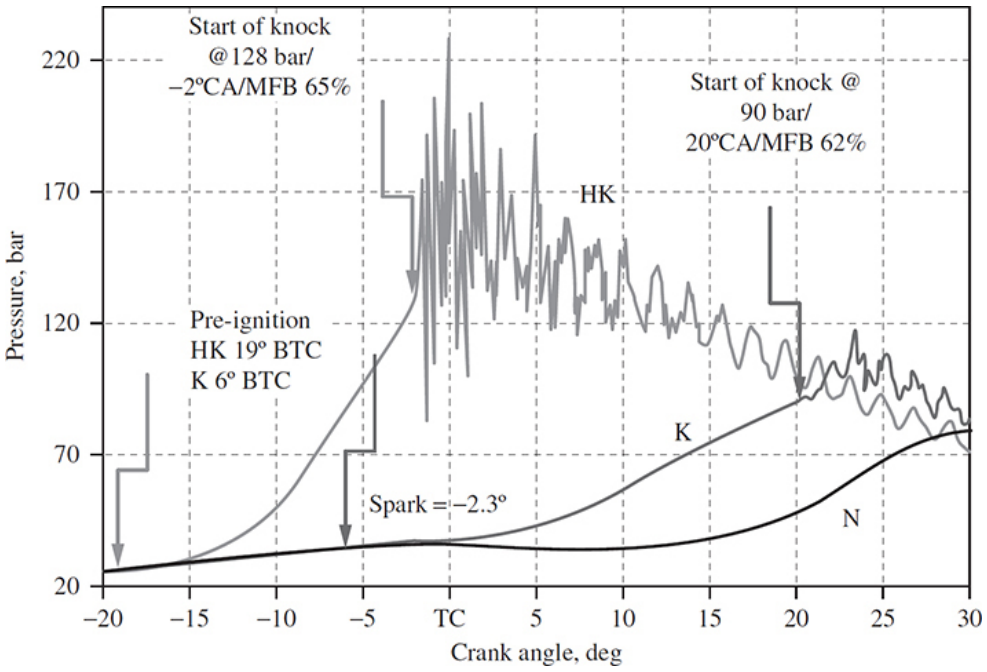


Figure 9.68 Three individual in-cylinder pressure cycles in a turbocharged gasoline engine operating at high load (2 bar manifold air pressure) and 2000 rev/min with retarded spark timing (2.3° CA before TC). A normal combustion cycle (N) with trace knock has a low (≈80 bar) and late (30°

ATC) peak pressure. A mid-intensity knocking cycle (K), where preignition occurs at 6° BTC, has a pressure at knock onset of 90 bar, and a heavy knock (with early preignition at 19° BTC) cycle, HK, reaches a pressure of 128 bar at knock onset. MFB is mass fraction burned. [92]

The pressures and unburned mixture (end-gas) temperatures at the autoignition point that produces knock for these three cycles are shown in Table 9.7. The difference in knock intensities between these three cycles cannot be explained by the different unburned mixture mass fractions available for autoignition. Rather they depend on the process and thus speed by which the autoignition pressure waves propagate outward, causing other unburned gas regions to autoignite and establish the strong pressure waves of these intense knock phenomena. The normal cycle (where knock just occurs) is a relative slow deflagration pressure wave process, the normal knock initiation process. The mid-knock intensity cycle is a developing-detonation pressure wave process where the chemical energy release the wave initiates is driving and intensifying the pressure wave. The very high knock intensity cycle is the result of a strong developing detonation wave where the high pressure wave initiates an increasingly large chemical energy release which drives the wave faster and increases its magnitude.

A plausible explanation for the preignition process that initiates this super-knock phenomenon is that it results from vaporized oil in the in-cylinder fuel-air mixture. Oil vapor when mixed with gasoline and air has a lower self-ignition temperature than a gasoline/air mixture and thus will autoignite at lower unburned mixture temperatures. Oil is present as a thin film on the cylinder liner, and between the liner and piston. Gasoline, impinging on the liner absorbs oil from the film, and when the piston travels upward during the compression stroke, fuel-air mixture can collect in the crevice volume above and below the top ring in the piston lands. Thus oil-fuel mixture in vapor or liquid form can accumulate in this region: eventually enough mixture may accumulate and exit the crevice, and be convected into the unburned in-cylinder gases away from the combustion chamber wall. That mixture in that specific cycle (or few cycles, until this accumulated oil plus fuel mixture is gone), then spontaneously ignites. The origin of this early preignition phenomenon is the formation of a local region of oil vapor, gasoline vapor, air, and burned gas (likely during compression) that

autoignites due to its low octane rating. This early initiation of combustion results in much higher-than-normal unburned mixture pressures and temperatures that then autoignite the end-gas when only half to two-thirds of the in-cylinder gasoline and air has burned, resulting in very heavy knock as seen in Fig. 9.68. Experiments designed to test this explanation (such as reducing crevice volumes to increase the frequency of this preignition initiated super-knock phenomenon, use of less volatile oils) provide confirmation. [93]

9.6.5 Knock Suppression

Knock suppression would allow SI engine compression ratios, and boost levels in turbocharged engines, to be increased thereby improving engine performance and efficiency. Strategies for suppressing or delaying knock onset include use of fuel additives, high-octane fuel blending compounds, and engine modifications, as summarized below. Note that fuel additives are widely used to counteract fuel-related deterioration in engine performance. Detergent additives are used to ensure that the intake system (fuel injectors, intake valves) and the combustion system (spark plugs, combustion chamber walls) remain free of deposits and function as intended. Corrosion inhibitors are included in gasolines to form a thin protective film on metal surfaces to prevent water/moisture induced corrosion. Antioxidants are added to fuels to improve their storage capability. [51]

Knock-Suppressing Fuel Additives and Blending Compounds

Modern gasolines contain a number of chemical additives designed to improve fuel quality. Gasoline base stocks are also blended with oxygen containing organic compounds to raise their octane rating. The most effective antiknock agents are lead alkyls. Tetraethyl lead (TEL), $(C_2H_5)_4Pb$, was first introduced in 1923. Tetramethyl lead (TML), $(CH_3)_4Pb$, was introduced in 1960. Since TML boils in the mid-range of a gasoline (110°C) (whereas TEL boils at the high end, 200°C), the introduction of TML permitted better distribution of octane amongst the cylinders of an engine. In 1959, a manganese antiknock compound (methylcyclopentadienyl manganese tricarbonyl), known as MMT, was introduced as a supplementary antiknock agent for TEL. It is also an antiknock agent in its own right. In the 1970s,

unleaded gasolines were introduced and now dominate. Two factors influenced the elimination of lead alkyls: concern about the toxicological aspects of lead in the urban environment, and the use of catalytic devices for emission control that are poisoned by lead. MMT is no longer used as an antiknock additive in unleaded gasoline due to its role as a deposit in plugging exhaust catalytic converters.

Two (sometimes three) grades of (unleaded) gasoline with different octane ratings are usually available at refueling stations. In the United States the lower and higher rated fuels are labeled "regular" and "premium" with antiknock indexes of 87 and 93. In Europe average gasoline octane numbers are two to three ONs higher. [r] Gasolines with higher octane ratings than these could be produced at the refinery, but historically the cost of the additional refining required to do this, relative to the in-vehicle benefit has been judged marginal.

This refining limitation on petroleum-derived (unleaded) gasolines has increased interest in other methods of boosting the octane rating of gasolines. The use of oxygenates—alcohols and ethers—as gasoline extenders, which due to their excellent antiknock quality increase the fuel's octane rating, is common. A brief review of the mechanism by which these additives and blending compounds improve the knock resistance of gasoline follows.

Lead is the most effective antiknock element known. In the form of lead alkyls, it is stable and fuel-soluble. It is generally agreed that the alkyls decompose before they exert their antiknock action. The decomposed material—lead oxide, PbO—either as a vapor or as a fog (a dispersion of fine particles), inhibits the preflame chain-branching reactions which lead to autoignition of the fuel-air charge, thus slowing the overall reaction rate. However, lead has little effect on two-stage ignition until after the cool flame. The individual alkyls vary in antiknock behavior as a function of fuel composition and combustion conditions. The average effect of various amounts of TEL in a large number of regular gasolines is summarized in Fig. 9.69: the effectiveness of each successive increment of TEL, added steadily, decreases. The addition of about 0.8 g lead per liter (3 g Pb per U.S. gallon), provides an average gain of about 10 octane numbers, though lead's effectiveness varies with the chemical composition of the base fuel. When a gasoline containing a lead alkyl is burned in a spark-ignition engine, it produces nonvolatile combustion products. These deposit on the walls of the combustion chamber and on the spark plug, causing lead-fouling of the spark

plug electrodes and tracking across the plug insulator, and corrosion of the hot exhaust valve. Commercial lead-containing antiknock fluids, therefore, contain scavenging agents—combinations of ethylene dibromide and ethylene dichloride—which transfer the lead oxides that would otherwise deposit into volatile lead-bromine compounds which are largely exhausted with the combustion gases.

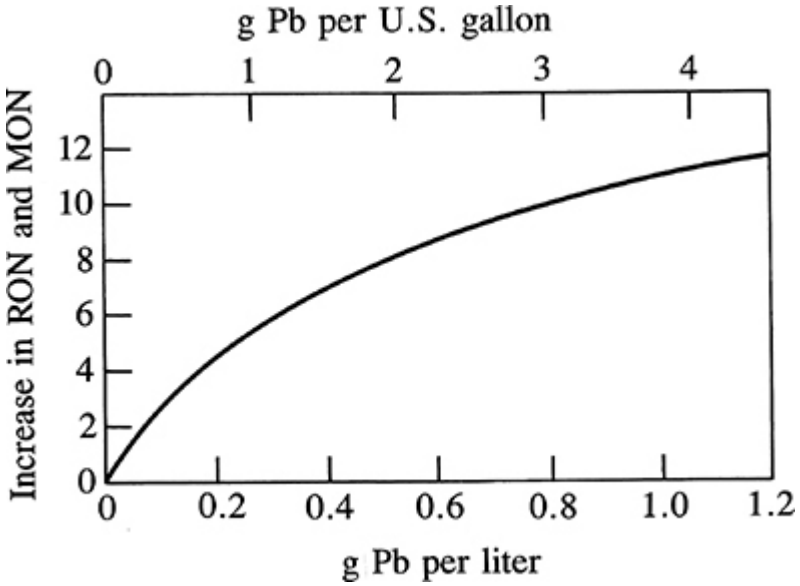


Figure 9.69 Gasoline octane number increase resulting from use of antiknock additive tetraethyl lead. Increase varies with fuel composition: average values shown.

The use of oxygenates as extenders or substitutes for gasoline is increasing. In some cases, this is because the oxygenate can be produced from nonpetroleum sources (e.g., ethanol from biomass) and thus may offer strategic or economic benefits. In other cases, the good antiknock blending characteristics of oxygenates can aid in meeting octane quality requirements where regulations prevent or limit lead alkyl use. Several oxygenates have been used as automotive fuels: those of greatest interest are ethanol (C_2H_5OH), methanol (CH_3OH), tertiary butyl alcohol (TBA, C_4H_9OH), and methyl tertiary butyl ether (MTBE).

Ethanol is being made from biomass, especially sugar cane and corn grain. Table 9.8 lists the knock-suppressing characteristics of these compounds and their physical and chemical characteristics relative to

gasoline. The *blending octane value* of these oxygenated compounds is not necessarily the same as the compound's research octane number and depends on the composition (and thus the octane rating) of the gasoline with which the compound is blended. MTBE-gasoline blends have good stability in the presence of water, and MTBE has little effect on fuel vapor pressure, and materials compatibility. TBA is susceptible to water extraction and loss. Ethanol is technically attractive as a high-octane supplement or substitute for gasoline and is used in 5 or 10% blends in gasolines, as well as a stand-alone fuel as E85 (nominally 85% ethanol, 15% gasoline).

TABLE 9.8 End-gas conditions at knock onset (see Fig. 9.68)

	End-gas autoignition			
Cycle	CA	p_a, bar	T_a, K	Mass fraction burned
Knock onset N	25° ATC	70	800	Unknown
Knock onset K	20° ATC	90	850	0.62
Early preignition/ heavy knock HK	2° BTC	128	926	0.65

Fuel blending of these oxygenated hydrocarbon compounded with a base gasoline can be illustrated as follows with ethanol. Because ethanol raises the octane and volatility of blends, it can be blended with a lower octane, reduced volatility base gasoline to ensure that the final gasoline-ethanol blend meets the fuel's specifications and regulations. Figure 9.70 shows how the blending of ethanol with various base gasolines (referred to as blendstock for oxygenate blending, BOB) increases the research octane number of the final fuel. Note that as the RON of the blendstock increases, the slope of each increasing RON line decreases. The blending RON of ethanol derived from the linear portion of the curves in Fig. 9.70 (up to about 15% ethanol) decreases from about 145 to 125 as the base gasoline octane increases from 84 to 98 RON. Note also that a saturation effect occurs with ethanol percentages above about 30%.

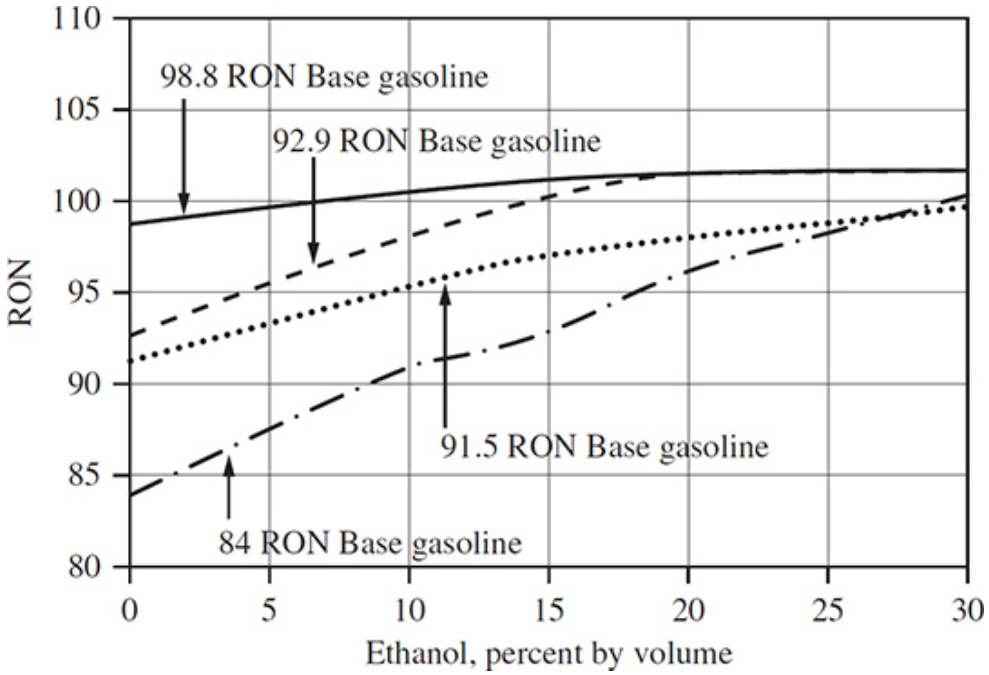


Figure 9.70 Research octane number as a function of volumetric content of ethanol for four different base gasolines with RON values of 84, 91.5, 92.9, and 98.8.[83]

Methanol, because it can be made from natural gas, coal, or biomass materials, has potential as an additive and alternative fuel. Its high blending-octane quality (about 130 RON, 95 MON), when used in low-concentration (~5%) methanol-cosolvent-gasoline blends, means it could be used to upgrade low octane gasoline. Problems with methanol include poor solubility in gasoline in the presence of water, high toxicity, an energy content per unit volume about half that of gasoline; high latent heat of vaporization (which can contribute to poor drivability), incompatibility with many commonly used metals and elastomers, blending effects on gasoline volatility which may force the displacement of significant volumes of butane. Some of these problems can be partially reduced by using cosolvents such as TBA or isobutanol. Use of ethanol and methanol as stand-alone fuels in specially designed engines permits advantage to be taken of their high octane rating via higher compression ratios and boost levels. As stand-alone fuels, however, their significantly lower energy content than gasoline is a disadvantage.

The octane requirement of an engine-vehicle combination usually increases somewhat during use, due primarily to the buildup of deposits on the cylinder head and piston crown. While these deposits increase the engine's compression ratio modestly, their largest effect is to increase the temperature of the surface of the combustion chamber in contact with the inducted mixture. This *increases* heat transfer *to* the fresh mixture during induction and *decreases* heat transfer from the unburned charge during compression. End-gas temperatures are therefore higher, thus increasing the likelihood of knock. As the combustion chamber deposits stabilize (over some 15,000 km of driving), the engine's octane requirement typically increases by up to about 5 octane numbers before it stabilizes. The advent of cleaner gasolines and improved deposit controlling additives is reducing this deposit related octane requirement increase.

Engine Knock Control Strategies

Autoignition in the end-gas ahead of the flame, the initiating process for engine knock, depends on the maximum unburned mixture temperature and pressure, and on the time available before flame arrival, for a given fuel octane rating. Here we focus on those spark-ignition engine variables that affect the unburned mixture's temperature and pressure, and time available for autoignition to develop, that can be changed and thus affect knock onset. As is well known, knock constrains engine compression ratio in all spark-ignition engines, and also the boost level in turbocharged SI engines, and thus limits the maximum engine mean effective pressure and fuel conversion efficiency.

The engine variables that determine knock onset, and that can be used to control knock are: spark timing (relative to MBT timing), relative air/fuel ratio, intake air temperature, coolant temperature, dilution of the mixture with cooled exhaust gas recirculation, and cooling of the in-cylinder charge through evaporation of the liquid fuel. A review of the literature supplemented by studies of these effects in more modern engines [95] provides a guide to the effects of several important parameters on the octane requirement (the minimum octane number fuel needed to avoid knock) of gasoline-fueled spark-ignition engines. The results are given in Table 9.10.

TABLE 9.10 Effect of engine operating variables on engine's octane requirement

Variable	Octane number requirement change per unit change in variable
Compression ratio	+ 5 ONR/unit r_c increase
Spark retard	− 1 ONR/1° retard
Air/fuel ratio*	− 2 ONR/unit *A/F*
Intake air pressure†	+ 2–4 ONR/10 kPa increase
Exhaust back pressure	+ 1 ONR/30 kPa increase
Intake air temperature	+ 1 ONR/7 deg C increase
Coolant temperature	+ 1 ONR/10 deg C increase
Dilution cooled EGR	− 3–4 ONR/10% mass dilution
Type of fuel injection	− 4 ONR/DI relative to PFI
Altitude	− 1.4 ONR/300 m rise (0–1800 m elevation)
	− 2.5 ONR/300 m rise (1800–3600 m elevation)

*Octane requirement peaks at 5% (relative air/fuel ratio) rich of stoichiometric and decreases for richer (and leaner) mixtures.
†Higher value for naturally-aspirated gasoline engines; lower value for turbocharged engines.

Spark retard is the most commonly used control approach to reduce the engine's octane requirement. Retarding the phasing of the combustion process by retarding spark from its MBT timing, decreases the peak cylinder pressure (which occurs later ATC), and as a consequence, decreases the maximum unburned mixture temperature. Many studies have indicated that each crank-angle degree of spark retard decreases the fuel RON required to avoid knock by 1 octane number. Since the loss in torque, bmep, and efficiency is modest for up to 5 degrees of spark retard from MBT timing (see Fig. 9.16), and only becomes significant as retard increases beyond about 10 degrees, spark retard is widely used. Also its effect on knock avoidance is immediate.

Spark-ignition engines are often operated about 20% fuel rich of stoichiometric at wide-open throttle to increase torque, lower the thermal loading on the exhaust catalyst system (and the turbine in turbocharged engines), and reduce the engine's octane requirement. Figure 9.71 shows the effect of relative air/fuel ratio changes on an engine's octane requirement, based on a review of the relevant literature.[95] The octane requirement is highest at around 0.95 relative air/fuel ratio, about 5% fuel-rich of stoichiometric. The octane requirement decreases on either side of this maximum—both for richer mixtures, and for less-rich mixtures through stoichiometric to lean mixtures. The sensitivity of these decreases is about 3 ON per 0.1 relative air/fuel ratio, or 2 ON per unit air/fuel ratio. Note that while WOT operation with rich mixtures has historically been used in

gasoline engines for many years, it does result in high air pollutant emissions. For this reason, many modern engines operate stoichiometric at WOT so the three-way catalyst continues to be effective. However, this results in higher exhaust gas and catalyst temperatures (due to HC, CO, and H_2 oxidation within the catalyst), and higher turbine inlet temperatures.

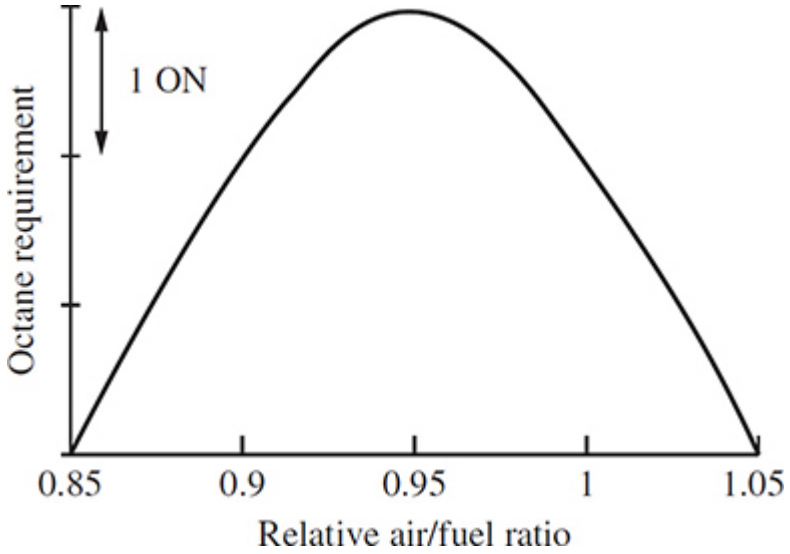


Figure 9.71 Effect of relative air/fuel ratio (λ) on engine octane requirement. $\lambda < 1$, rich mixture.[95]

Unburned in-cylinder mixture temperature is a critical variable in the autoignition process leading to knock. It is affected by the intake air temperature, the amount and temperature of the in-cylinder residual gas, and coolant temperature (which affects cylinder head, piston and liner temperatures, and thus heat transfer to the unburned mixture—which is into the mixture through the first half of the compression process). Table 9.9 gives the octane sensitivity of these temperature-related variables: the most important is the intake air temperature.

TABLE 9.9 Oxygenate properties

	Methanol	Ethanol	TBA	MTBE	Gasoline
Research octane no. (RON)	106	109	~105	~114	91–98
Blending octane no. (RON)	130	129	~110	~125	NA
Weight percent oxygen	50	35	22	18	NA
Stoichiometric (*A/F*)	6.5	9.0	11.2	11.7	14.5
Specific gravity	0.796	0.794	0.791	0.746	0.74
Lower heating value, MJ/kg	20.0	26.9	32.5	35.2	42
Latent heat of vaporization, MJ/k†	1.16	0.84	0.57	0.34	0.35
Boiling temperature, °C	65	78	83	55	27–227

TBA, tertiary butyl alcohol; MTBE, methyl tertiary butyl ether.
Note that blending octane number values depend on the octane of the base blendstock gasoline and amount of oxygenate. Source: Table D.4; Motor Gasolines Technical Review; Chevron Products Company, Motor Gasolines *Technical Review* , Chevron Corporation, 1996 and 2009.94
†Depends on temperature.

Direct fuel injection into the cylinder helps suppress knock. The OR decreases by about 4 relative to the equivalent PFI value. As the fuel spray evaporates with DI, its heat of vaporization is drawn from the in-cylinder charge, thus cooling that charge. With gasoline, the maximum in-cylinder charge cooling effect is just under 20 K. Studies that compared the charge cooling effect of gasoline direct in-cylinder fuel injection with intake-port fuel injection indicate that for a gasoline (without any ethanol) the difference in charge cooling is 13 to 14 K which is about 70% of the maximum gasoline evaporative cooling effect. This suggests that some in-cylinder fuel-evaporative charge cooling occurs with port-fuel-injection systems—about a quarter of the maximum cooling effect. Direct injection of gasoline would be expected to realize close to the maximum cooling effect, since close to full evaporative of the fuel occurs prior to combustion.

Intake manifold pressure is also an important variable. Increased in-cylinder pressure accelerates the autoignition chemical reactions. Data indicate that the octane requirement sensitivity to manifold pressure is higher at WOT for naturally-aspirated engines (some 3 or 4 ON per 10 kPa change in MAP) than for turbocharged engines with MAP values approaching 2 bar absolute (about 2 ON per 10 kPa change in MAP). (In part this may be due to maximum pressure constraints in TC engines.)

Note that the octane requirement sensitivities in Table 9.10 will depend somewhat on engine design and operating details and also on whether the

engine is turbocharged. In the engine development and design process the many variables that affect the knock requirement of the engine are traded-off against each other so as to optimize part-load engine brake fuel-conversion efficiency, and wide-open-throttle torque (or bmep), for the intended fuel octane rating.

One strategy used to suppress knock onset is the use of *cooled exhaust gas recycle* (EGR). This is mostly used in turbocharged gasoline engines where knock significantly constrains engine compression ratio and boost level. The use of EGR at high load dilutes the engine's in-cylinder fuel/air/burned gas mixture, which normally reduces the torque a given displacement engine produces. However, this knock suppression effect allows a higher compression ratio and boost level to be used which more than offsets the charge dilution effect.

Dilution of the unburned mixture with EGR has the following effects on engine operation. The inert component of the fuel/air/burned gas mixture—nitrogen plus CO_2 plus H_2O, with a stoichiometric mixture—is increased thus slowing down the flame development and rapid burning angles (see Fig. 9.15) due to the lower laminar flame speed (see Sec. 9.3.3). The slower burn requires earlier spark timing to obtain maximum brake torque, but with MBT spark timing, the impact on knock onset is modest.

The explanation of the knock suppressing effect of cooled EGR is based on the fundamental physical variables—unburned mixture pressure and temperature—in the autoignition induction-time correlation, Eq. (9.68) . The highest values of *Tu* and *p* depend on both the piston-motion-produced compression process coupled with the combustion-produced pressure rise which additionally compresses the unburned mixture. [96] To a good approximation the unburned mixture is compressed adiabatically by the prevailing pressure, thus

$$\frac{\dot{T}_u}{T_u} = \frac{(\gamma_u - 1)}{\gamma_u}\frac{\dot{p}}{p} \tag{9.71}$$

The pressure rise rate may be obtained from the "heat" or chemical-energy release First Law analysis of the engine combustion process where the engine cylinder is treated as a uniform one-zone system [Eq. (9.27) with $V_{cr} \rightarrow 0$]:

$$\frac{\dot{p}V + p\dot{V}}{(\gamma - 1)} = \dot{Q} - p\dot{V}$$

which, when combined with Eq. (9.71) gives

$$\frac{\dot{T}_u}{T_u} = \frac{(\gamma_u - 1)}{\gamma_u}\left[(\gamma - 1)\frac{\dot{Q}}{pV} - \gamma\frac{\dot{V}}{V}\right] \tag{9.72}$$

Note that γ is the average ratio of specific heats for the assumed uniform gases in this one-zone energy release approach: $\dot{Q}$ is the net heat release, $\dot{Q}_{ch} - \dot{Q}_{ht}$.

The effect of increasing EGR on γ_u, the ratio of specific heats of the unburned mixture, is shown in Fig. 9.72: when EGR is increased from 0 to 36%, the (γ_u - 1)/ γ_u factor in Eqs. (9.71) and (9.72) increases by 2.5%. However, with a stoichiometric mixture and constant fuel mass in the cylinder, as the amount of cooled exhaust gas recirculated increases, the in-cylinder pressure increases since, with increasing EGR for fixed fuel mass, the intake manifold pressure must increase. Cycle simulation results show that *prior* to spark, the higher γ_u increases the unburned mixture temperature; but that after combustion starts, the higher pressures decrease the first term in the square brackets in Eq. (9.72) [s] and more than offset the effect of increasing in (γ_u - 1)/ γ_u. So peak unburned mixture temperatures are reduced by cooled EGR. Thus the autoignition induction time is longer and knock onset is suppressed. Figure 9.73 shows the change in peak unburned mixture temperature as a function of EGR for two different loads—one relevant to naturally-aspirated, the other to turbocharged engines. The decrease in peak end-gas temperature is 5 K at 10% EGR and 12 K at 20% EGR. [96]

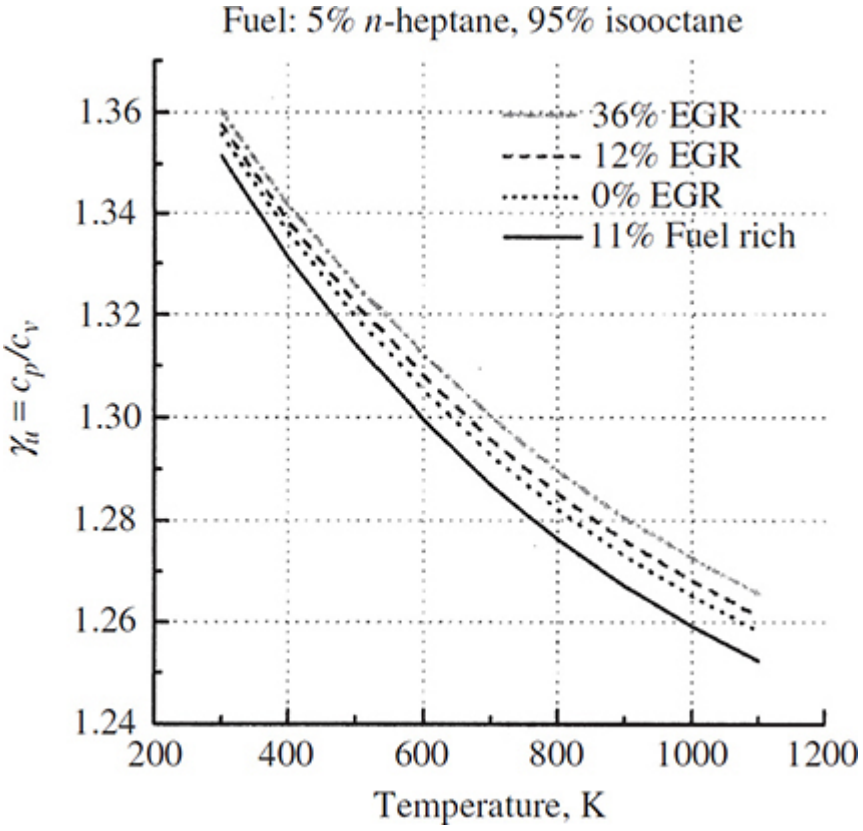


Figure 9.72 Ratio of unburned mixture specific heats (c_p/c_v) as a function of dilution and gas temperature.[97]

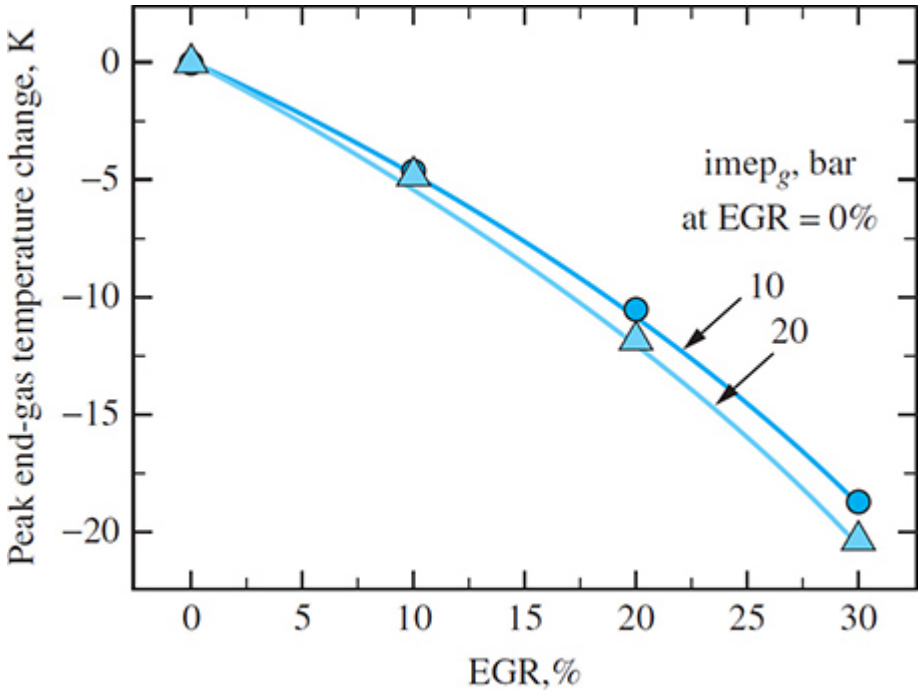


Figure 9.73 Change in maximum end-gas (unburned mixture) temperature as a function of amount of cooled EGR: at constant fuel mass flow rate. Two curves are for imep $_g$ = 10 and 20 bar at EGR = 0%. [96]

Use of EGR for knock suppression in turbocharged gasoline engines requires that the recycled gases are cooled in a heat exchanger and fed into the intake via a control valve upstream of the compressor. Since the exhaust gases contain unburned hydrocarbons and fine particulates with some 12% water, as well as trace amounts of lubricating oil, deposit build-up and corrosion in these EGR system components are issues.

PROBLEMS

9.1 The table gives relevant properties of three different spark-ignition engine fuels. The design and operating parameters of a four-cylinder 1.6-liter displaced volume engine are to be optimized for each fuel over the engine's operating load and speed range. You may assume that for each engine-fuel combination, the gross indicated fuel conversion efficiency and imep at any operating condition are given by 0.85 times the fuel-air cycle efficiency at those conditions. Also, assume that for every five research octane number increase above 95 (the gasoline value) in fuel antiknock rating, the compression ratio can be increased by one unit. For gasoline, the engine compression ratio is 9, so if the octane number of the fuel is 100, a compression ratio of 10 can be used.

(a) At part-throttle operation—at an intake pressure of 0.5 bar and a speed of 2500 rev/min—estimate the gross indicated fuel conversion efficiency and specific fuel consumption for each engine-fuel combination. Each engine-fuel combination operates at the lean limit given.

(b) At the appropriate equivalence ratio for maximum power, with 1 bar inlet pressure at the same speed (2500 rev/min), the volumetric efficiencies are as shown. Explain these volumetric efficiency values. Each fuel is fully vaporized in the inlet manifold and the inlet mixture temperature is held constant for all fuels. Manifold and valve design remains the same.

(c) Estimate the ratios of the maximum indicated mean effective pressure, at the same conditions as in (*b*), of the natural gas and methanol fueled engines to the maximum imep of the gasoline engine.

(d) lf the methanol engine burns 33% faster than the gasoline engine (i.e., its combustion event takes three-quarters of the time), sketch a *carefully drawn qualitative* cylinder pressure versus crank angle curve for the two engines, from halfway through the compression stroke to halfway through the expansion stroke. Put both curves on the same graph. Conditions are as in (*b*) . Show the motored *and* firing pressure curves for each engine. The spark timing should be set for maximum brake torque for each engine. Show the location of spark timing, the location of maximum cylinder pressure, and the approximate location of the end of combustion.

Fuel properties

Fuel	Formula	Research octane number	λ at lean operating limit	Stoichiometric air/fuel ratio	Heating value, MJ/kg	η_v
Natural gas	CH_4	120	1.4	17.2	50	0.78
Methanol	CH_3OH	105	1.25	6.4	20	0.75
Gasoline	$(CH_2)_n$	95	1.1	14.9	44	0.85

λ = relative air/fuel ratio.

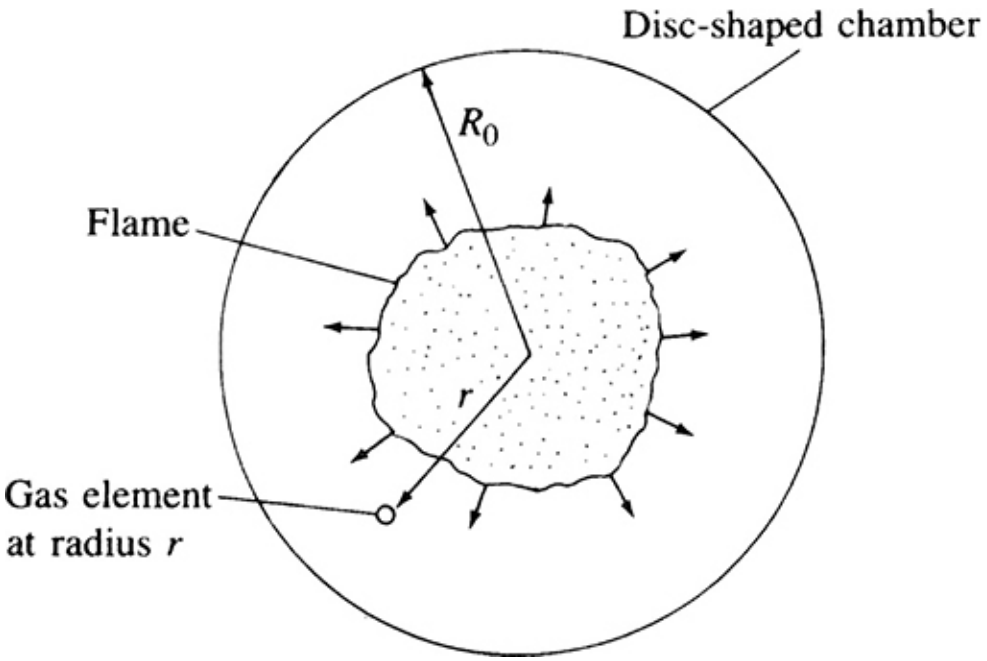
9.2 The figure shows the flame propagating radially outward from the center of a disc-shaped combustion chamber. Combustion in such a device has many features in common with spark-ignition engine combustion. The chamber diameter is 10 cm and the height is 1.5 cm. For this problem, the flame can be thought of as a thin cylindrical sheet. Its radius increases approximately linearly with time. The volume of the chamber is constant. The fuel is a typical hydrocarbon fuel; the mixture is stoichiometric; the initial temperature is room temperature.

(a) Sketch qualitative but *carefully proportioned* graphs of the following quantities versus time from the start of combustion to the end of combustion:

1. The ratio of actual pressure in the chamber to the initial pressure
2. The ratio of average density of the gas *ahead* of the flame to the initial density
3. The ratio of the average density of the gas *behind* the flame to the initial density

(b) On a qualitative but *carefully proportioned* graph of r/R_0 versus time show how the radial positions of gas elements, initially at r/R_0: 0, 0.5, and 1.0 before combustion, change during the combustion process as the flame propagates radially outward from the center ($r = 0$) to the outer wall ($r = R_0$).

Note: Accurate numerical calculations are not required to answer this question. You will be graded on the amount of physical insight your diagrams and the supporting brief explanations of your logic communicate. You should write down any equations or approximate numerical values for relevant quantities that help explain your reasoning.



9.3 In a spark-ignition engine, a turbulent flame propagates through the uniform premixed fuel-air mixture within the cylinder and extinguishes at the combustion chamber walls.

(a) Draw a carefully proportioned qualitative graph of cylinder pressure p and mass fraction burned x_b as a function of crank angle q for $-90° < q < 90°$ for a typical SI engine at wide-open throttle with the spark timing adjusted for maximum brake torque. Mark in the crank angles of spark discharge, and of the flame development period (0 to 10% burned) and end of combustion, on both p and x_b versus q curves relative to the top-center crank position.

(b) Estimate approximately the fraction of the cylinder volume occupied by burned gases when the mass fraction burned is 0.5 (i.e., halfway through the

burning process).

(c) A simple model for this turbulent flame is shown on the left in the figure below. The rate of burning of the charge $dm_b/\ dt$ is given by

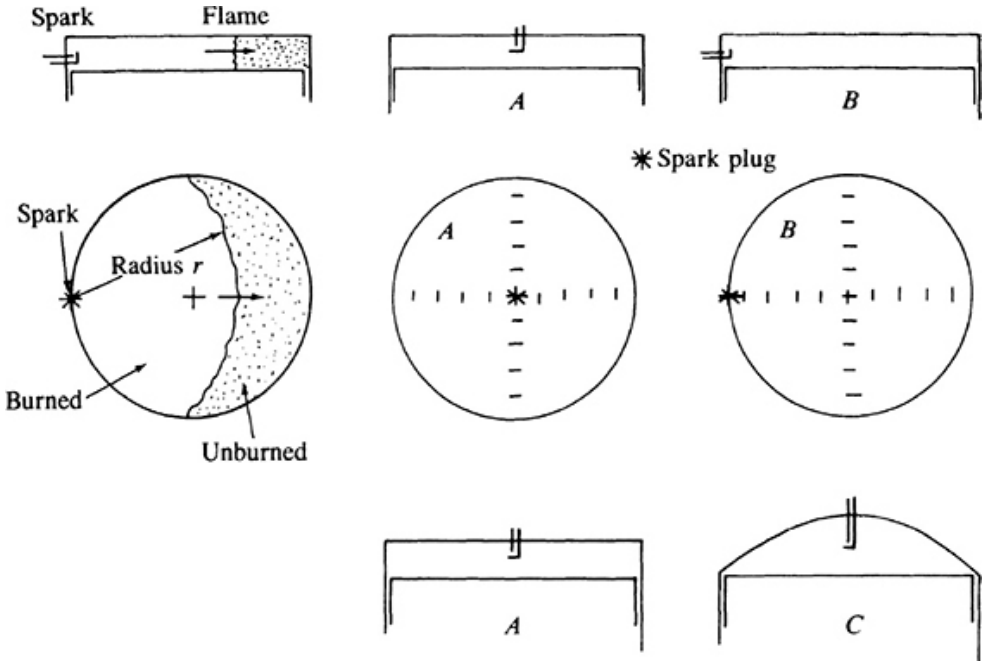
$$\frac{dm_b}{dt} = A_f \rho_u S_T$$

where A_f is the area of the flame front (which can be approximated by a portion of a cylinder whose axis is at the spark plug position), ρu is the unburned mixture density, and S_T is the turbulent flame speed (the speed at which the front moves relative to the unburned mixture ahead of it). The rate of mass burning is influenced therefore by combustion chamber geometry (through A_f) as well as those factors that influence S_T (turbulent intensity, fuel/air ratio, residual gas fraction, and EGR). Compare combustion chambers *A* and *B* shown in the figure. Sketch the *approximate* location of the flame front when 50% of the mass has been burned. (A careful qualitative sketch is sufficient; however, provide a quantitative justification for your sketch.) Sketch the mass fraction burned versus crank angle curves for these two combustion chambers on the same graph, each with its spark timing set for maximum brake torque. You may assume the value of S_T is the same for *A* and *B*.

(d) Compare combustion chambers *A* and *C* in the figure which have the same flame travel distance but have different chamber shapes. Which chamber has

1. The faster rate of mass burning during the first half of the combustion process?
2. The faster rate of mass burning during the second half of the combustion process?
3. The more advanced timing for maximum brake torque?

Explain your answers.

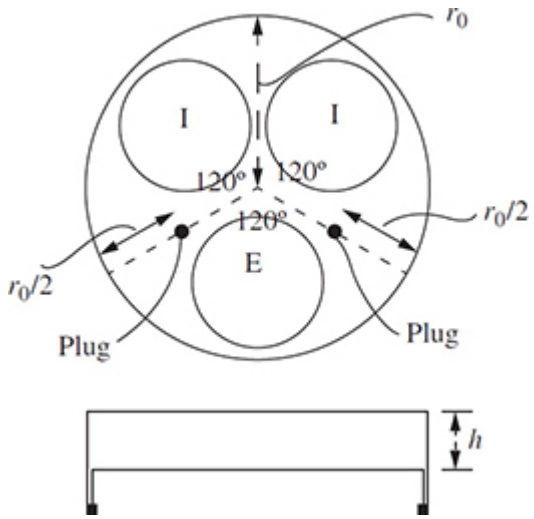


9.4 Compare the three-valve two-plug combustion chamber shown, with a four-valve chamber with one spark plug on the cylinder axis. Make appropriate simple assumptions about the flame shape, as it grows outward from the spark(s).

(a) For the four-valve center plug, plot flame envelope area (flame outer boundary) $A_f/2ph_0$ against r_f/r_0 through the combustion process. On the same graph, sketch carefully the equivalent curve for the three-valve two-plug design assuming both plugs fire at the same time: simple geometric calculations are required to short out the physics.

(b) Plot the mass fraction burned curves versus CAD for these two combustion chambers on a qualitative but carefully proportioned graph from 45 CAD BTDC to 45 CAD ATDC. Show on the graph the spark discharge location, the TDC, the location of maximum in-cylinder pressure and the approximate end of combustion, with the spark timing for both chambers set at MBT timing.

(c) With the two-plug chamber, the burn rate can be slowed down by firing one plug later than the other. Estimate approximately the mass fraction burned after which further delaying the second plug firing will have no impact on the burn rate.



9.5 In the disk-shaped combustion chamber of a spark-ignition engine, the flame propagates radially outward from the spark plug at the center (on the cylinder axis). The flame can be modeled as a thin smooth sheet at the location within the wrinkled turbulent flame, which on average separates the unburned and burned gases.

(a) Derive an equation for the mass fraction burned $x_b = m_b/m$ as a fraction of r_f/r_0 (r_f = flame radius and r_0 = cylinder radius) and the density ratio $r_d = ru/r_b$ Plot this curve for an appropriate value of r_d.

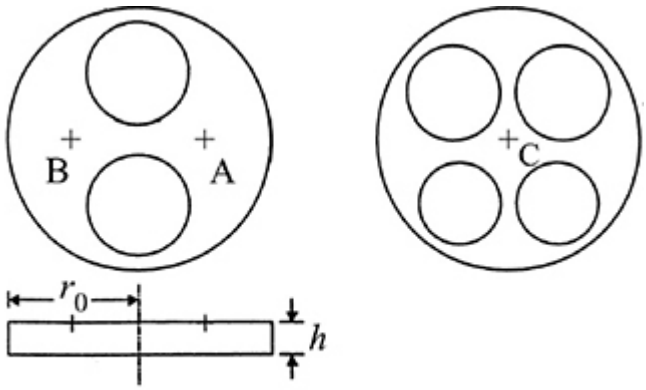
(b) Draw a carefully proportioned graph of x_b versus CAD for the engine operating at part-load with MBT spark timing (30 CAD BTC), indicating the approximate mass fraction burned at TC and 10 CAD ATC. Note that x_b at $\theta_{p\ max}$ is about 0.75.

Make an appropriate estimate of $p_{\ max}$. The pressure in the cylinder at IVC (50 CAD ABC) is the inlet pressure, i.e., 0.5 bar. The cylinder volume ratio, volume at IVC to volume at $p_{\ max}$ is 7.3. The mixture is stoichiometric and the compression ratio is 10. Make reasonable assumptions of any other quantities you may need.

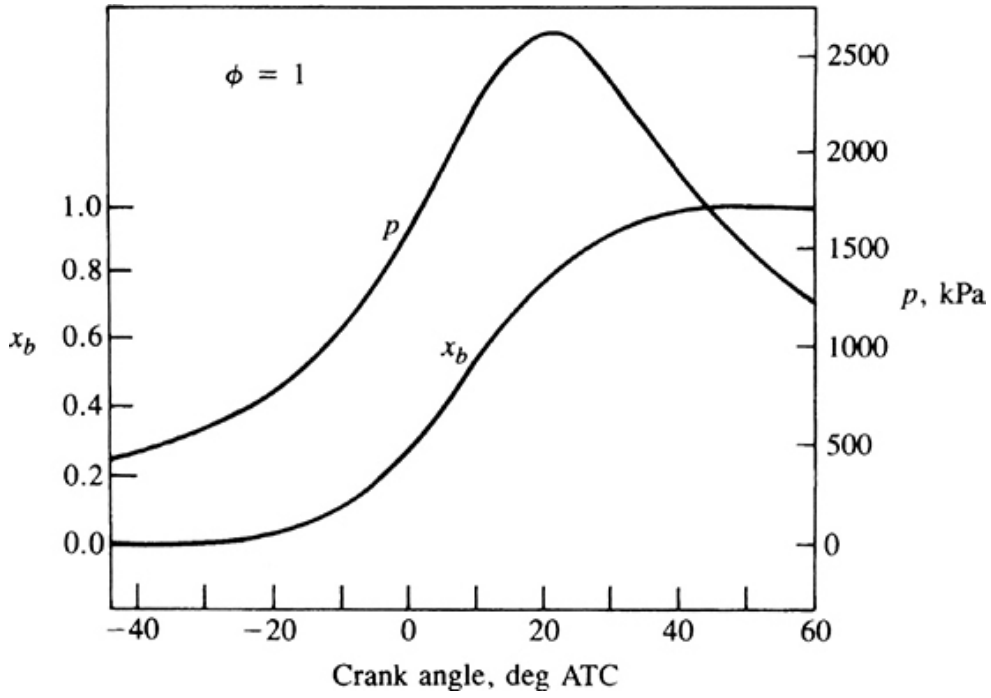
9.6 The cylinder head of a two-valve-per-cylinder spark-ignition engine with a single spark plug [at A, $r_0/2$ from the cylinder axis, in Fig. (i)] is to be modified to increase its burn rate. The options are to add a second spark plug at location B, or use a 4-valve head with a central spark plug [Fig. (ii)]. Assume that the clearance height h is constant, and is the same for both chambers (they are disk-shaped chambers).

(a) Assuming that each flame propagates outward in all directions at constant velocity from each spark plug, plot a carefully proportioned graph of flame outer (spherical) envelope area A_f divided by $(2\pi r h_0)$ $(2\pi r_0 h)$, versus flame radius rf/r_0 for the original one plug, the two plug, and the center plug options. R_0 is the cylinder radius. Identify 1.0 on each axis. Sketches and simple geometric calculations are helpful; complex calculations are not required.

(b) Based on these graphs, explain how the optimum spark timing will differ for these three chambers. For the original two-valve chamber (spark plug at A), the optimum spark timing is 30° before TC and the 50% burned point is 10° after TC. Will the two-plug chamber (A and B), and center plug chamber C, have more advanced or less advanced timing? Explain your logic. Assume the local flame velocity for each chamber is constant and the same for all chambers.



9.7 The graph gives the pressure-crank angle curve for a spark-ignition engine running at $\varphi = 1.0$. The mass fraction burned x_b is also shown. Estimate the temperature of the reactants at a number of crank angles and plot a graph of T_u versus θ. Assume the reactants in the cylinder are at 333 K and 1 bar pressure at the start of compression. It is necessary to make simplifications in order to do this and you should explain clearly what other assumptions you make; they should be compatible with Prob. 9.8 below.



9.8 If the combustion takes place progressively through the burning of a large number of very thin zones of gas (see Fig. 9.7) and there is no mixing between the zones, determine:

(a) the temperature at -30°, just after combustion, of the zone which burns at −30°;

(b) the temperature at 0°, just after combustion, of the zone which burns at 0°;

(c) the temperature at +30°, just after combustion of the zone which burns at +30°;

(d) the temperature of the products in these three zones at +30°.

Plot your results versus crank angle to show whether there is a spatial distribution of temperature in the cylinder after combustion.

Note: Each thin unburned gas element burns at essentially constant pressure and is subsequently compressed and/or expanded. Use charts (Chap. 4) or an equilibrium computer code. The unburned gas state is given by Prob. 9.7.

9.9 Explain why SI engine torque varies, at fixed speed and inlet mixture conditions, as the spark timing is varied from very advanced (e.g., 60° before TC) to close to TC. What is the "best" spark timing? Explain how it varies with engine speed and load.

9.10 **(a)** Explain the causes of the observed variation in cylinder pressure versus crank angle and imep in spark-ignition engines, cycle-by-cycle.

(b) What impacts do these cylinder pressure variations have on engine operation?

9.11 **(a)** Describe briefly what occurs when a spark-ignition engine "knocks."

(b) SI engine knock is primarily a problem at wide-open throttle and lower engine speeds. Explain why this is the case.

(c) With a knock sensor, the normal knock control strategy is to retard the spark timing when knock is detected, until knock no longer occurs. Explain why this strategy is effective and is preferred over other possible approaches (e.g., throttling the inlet, adding EGR).

(d) In a knocking engine, the crank angle at which autoignition occurs and the magnitude of the pressure oscillations which result vary substantially, cycle-by-cycle. Suggest reasons why this happens.

9.12 Knock in spark-ignition engines is an abnormal combustion condition. Almost everyone who rides a motorcycle or drives a car experiences this phenomenon at some time, and usually changing into a lower gear will take the engine away from this condition. Use your experience of what changes in other variables do, and consult this text and other sources to complete a table with the dependent variables shown at the top of the columns.

The independent variables are: speed, compression ratio, chamber surface/volume ratio, spark plug distance from cylinder axis, percent EGR, inlet mixture temperature, inlet mixture pressure, (F/A), wall temperature, air swirl, squish motion, fuel octane number. In these columns, show the corresponding influence on the dependent variables by a "+" for an increase and a "–" for a decrease. Show the effect of increase in engine system independent variables on: cylinder pressure and temperature, flame speed, total burn time, autoignition induction period, tendency to knock. Provide in the extreme right-hand column brief comments to explain your answers.

Effect of increase in engine independent variable	Dependent variables						
	Cylinder		Flame speed	Total burn time	Induction period	Tendency to knock	Explanation
	Pressure	Temperature					
Speed, rev/min							
Etc.							

9.13 **(a)** The electrical energy stored in a typical ignition system coil is 50 mJ. Almost all this energy is transferred from the coil during the glow discharge phase. If the glow discharge lasts for 2 ms, use the data in Fig. 9.39 to estimate the glow discharge voltage and current.

(b) Only a fraction of this energy is transferred to the fuel-air mixture between the spark plug electrodes. Estimate the energy transferred during the breakdown and glow phases of the discharge, using the data in Fig. 9.39 and Table 9.5.

(c) Overall about one-tenth of the coil energy is transferred to the fuel-air mixture. What fraction of the cylinder contents' chemical energy ($m_f Q_{\text{LHV}}$) does this correspond to at a typical part-load condition ($p_i = 0.5$ atm, $\varphi = 1.0$)? Assume 500 cm^3 per cylinder displaced volume. If the average burned gas temperature within the flame kernel just after spark is 3500 K and the cylinder pressure is 6 atm, what radius of kernel has fuel chemical energy equal to the electrical energy transferred to the kernel?

9.14 An approximate way to calculate the pressure in the end-gas just after knock occurs is to assume that all the end-gas (the unburned gas ahead of the flame) burns instantaneously at constant volume. We assume that the inertia of the burned gases prevents significant gas motion while the end-gas autoignites.

For the pressure data in figure in Problem 9.7, assume autoignition occurs at 10 crank angle degrees after the top-center position. Determine the maximum pressure reached in the end-gas after knock occurs. From the mass fraction burned and approximate average burned gas conditions at this time, estimate the volume occupied by the end-gas as a fraction of the cylinder volume just before autoignition occurs. Use the charts (Chap. 4) or an equilibrium gas composition computer code.

9.15 At spark timing (30° BTC) in an automobile spark-ignition engine (with bore = stroke = 85 mm and $r_c = 9$) at 2000 rev/min, operating on gasoline, the cylinder pressure is 7.5 atm and the mixture temperature is 650 K. The fuel-air mixture is stoichiometric with a residual gas fraction of 8%. The rapid burning angle $\Delta\theta_b$ (10 to 90% mass burned) is 35°. Estimate (a) the mean piston speed; (b) the average flame travel speed based on $\Delta\theta_b$ (the spark plug is located 15 mm from the cylinder axis); (c) the turbulence

intensity at TC [see Eq. (8.25)]; (*d*) the laminar flame speed at spark; (*e*) the turbulent burning speed S_b at TC using Eqs. (9.56) and (9.58) (appropriate firing and motored pressures are given in Fig. 9.4 *a*); (*f*) the mean expansion speed of the burned gases u_b at TC. Discuss briefly the relative magnitudes of these velocities.

9.16 The following combustion chamber design changes increase the mass burning rate in a spark-ignition engine at fixed compression ratio, bore, speed, and inlet mixture conditions. Explain how each change affects the burning rate.

(a) Reducing the amount of EGR.

(b) Using two spark plugs per cylinder instead of one.

(c) Generating swirl within the cylinder using a mask on the cylinder head (see Fig. 8.4).

(d) Using a combustion chamber with higher clearance height near the spark plug and a more central plug location.

9.17 The following factors contribute to cycle-by-cycle variations in the combustion process of an internal combustion engine, and hence in each cycle's cylinder pressure profile and torque pulse. Provide a *brief* description of why variations in each factor result in variations in the combustion process and how that causes variations in cylinder pressure cycle-by-cycle.

Causes of cycle-by-cycle combustion variations:

1. Flame kernel motion
2. Electrical energy deposited in spark-generated kernel
3. Heat losses from kernel to spark plug
4. Local turbulence characteristics near plug
5. Amount of fuel/air/EGR fed to cylinder
6. Local mixture composition near plug
7. Large scale features of the in-cylinder flow
8. Average turbulence in the cylinder

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[a] This model applies to burned gas regions of the chamber *away* from the walls. Heat transfer to the walls results in a thermal boundary layer on the walls, which grows with time as shown in Fig. 9.6. The gas in the boundary layer is not isentropically compressed and expanded.

[b] This angle is sometimes called the *ignition delay.* Since the flame starts to propagate outward immediately following the spark discharge there is no delay, and the terminology used here is preferred (see Sec. 9.3).

[c] An alternative definition for $\Delta\theta b$ uses the maximum burning rate to define an angle or time characteristic of the bulk charge burning process[4] (see Fig. 9.15). $\Delta\theta_b$(or $\Delta\theta_b^*$) are usually closely comparable.

[d] COV of a data set is the standard deviation divided by the mean value.

[e] The water in actual residual and exhaust gas was omitted, and the CO*2* level augmented to compensate. A mixture of 80% N_2 and 20% CO_2, by volume, was used.

[f] Controlled autoignition combustion is discussed in Sec. 10.7 .

[g] Variations in the total amount of air, recycled exhaust gas, and residual, in the chamber could also, for some operating regimes, be significant.

[h] This limit has often been called the *lean operating limit.* Since what limits engine operation in practice is excessive torque fluctuations, cycle-by-cycle, and high hydrocarbon emissions, resulting from the use of mixtures made overly dilute with either air or burned gases (or with both), *stable operating limit* is a more appropriate term.

[i] Note that during flame propagation, ρ_u / ρ_b is about 4: thus the volume of the end-gas region corresponding to say 20% of the mixture unburned is usually crescent shaped, and its location is defined by the flame propagation process.

[j] Often called the Livengood and Wu autoignition integral after its originators.[81]

[k] See Sec. 3.3 for a review of hydrocarbon structure and its nomenclature.

[l] ASTM denotes American Society for Testing and Materials; the letter and number define the specific testing code.

[m] The Cooperative Fuel Research Committee is now the Coordinating Research Council, Inc.

[n] The octane number of the fuel is calculated from ON = 100 + 28.28T/[1.0 + 0.736T + (1.0 + 1.472T - 0.0352l6T^2)$^{1/2}$], where T is milliliters of tetraethyl lead per U.S. gallon. Tetraethyl lead, $(C_2H_5)_4Pb$, contains 64 weight percent lead; 1 ml of TEL contains 1.06 grams of lead.

[o] For example, z can be the knock limited spark advance or knock limited torque.

[p] Ethanol, RON = 109, MON = 90, S = 19; methanol, RON = 109, MON = 89, S = 20.

[q] This phenomenon is sometimes called “super, or mega knock.”

[r] Research octane numbers are a more accurate predictor of a fuel’s resistance to knock. In the United States, regular and premium gasolines have RON values of close to 92 and 98, respectively. In Europe (where RON rather than AKI is used to label gasoline or petrol), the standard and premium fuels have RON values of 95 and 98.

[s] $\dot{Q}$ remains essentially unchanged.

CHAPTER 10

Combustion in Compression-Ignition Engines

10.1 ESSENTIAL FEATURES OF PROCESS

Combustion in compression-ignition or diesel engines occurs through a different sequence of processes than does combustion in a spark-ignition engine. The essential features of the diesel combustion process can be summarized as follows. Fuel is injected by the fuel-injection system into the engine cylinder toward the end of the compression stroke, just before the desired start of combustion (SOC). Figures 1.26 to 1.30 illustrate the major components of diesel fuel-injection systems. The liquid fuel, injected at high velocity, usually as several jets through small orifices or nozzles in the injector tip, atomizes into small drops and penetrates into the high-temperature high-pressure air in the combustion chamber. The fuel vaporizes and the fuel vapor mixes with air. The high-temperature fuel-vapor air mixture thus produced ignites spontaneously, after a delay period of a few crank-angle degrees, and starts burning. The cylinder pressure increases as combustion of the evolving fuel-vapor air mixture continues. The consequent compression of the unburned portion of the charge shortens the delay before ignition for the fuel and air which has mixed to within combustible limits, which then burns rapidly. It also reduces the evaporation time of the remaining liquid fuel. Injection continues until the desired amount of fuel has entered the cylinder. Atomization, vaporization, fuel-air mixing, and combustion continue until essentially all the fuel has passed through each process. In addition, mixing of the air remaining in the cylinder with burning and already burned gases continues throughout the combustion and expansion

processes. Thus, for much of the diesel's combustion process, the fuel-burning rate is controlled by the rate at which fuel mixes with air.

Such "non-premixed flames" are usually termed *diffusion flames* because the (gaseous) fuel and the air must diffuse into each other and then react. Most of the fuel chemical energy is released in a thin reaction sheet close to where the fuel vapor and air reach the stoichiometric mixture proportions (as shown in Fig. 9.2). Because the in-cylinder flow is turbulent, the diesel combustion process occurs primarily as an unsteady diffusion flame. Spontaneous ignition, resulting from the high air temperature and pressure in the cylinder when injection commences, and the fact that fuel-air mixing controls the burning rate, are the distinguishing features of the diesel combustion process.

It will be clear from this summary that the compression-ignition combustion process is extremely complex. [a] The details of the process depend on the characteristics of the fuel, on the design of the engine's combustion chamber and fuel-injection system, and on the engine's operating conditions. It is an unsteady, heterogeneous, three-dimensional combustion process, with complex chemistry. While a well-grounded conceptual understanding of diesel engine combustion has been developed, to date an ability to describe many of the critical individual processes in a quantitative manner is incomplete.

Some important consequences of this combustion process on engine operation are the following:

- Since injection commences just before combustion starts, there is no knock limit as in the spark-ignition engine resulting from spontaneous ignition of the premixed fuel and air in the end-gas. Hence a higher engine compression ratio can be used in the compression-ignition engine, improving its fuel conversion efficiency relative to the SI engine.
- Since injection timing is used to control combustion timing, the delay period between the start of injection (SOI) and start of combustion (SOC) must be kept short (and reproducible). A short delay is also needed to hold the maximum cylinder gas pressure below the maximum the engine can tolerate. Thus, the spontaneous ignition characteristics of the fuel-air mixture must be held within a specified range. This is done by requiring that diesel fuel have a cetane number (a measure of the

ease of ignition of that fuel in a typical diesel environment; see Sec. 10.6.2) above a certain value.

- Since the combustion process occurs within and around the fuel sprays, engine torque can be controlled by varying the amount of fuel injected per cycle, with the engine's airflow essentially unchanged. Thus, the engine can be operated unthrottled, so pumping work requirements are low: this improves part-load mechanical efficiency relative to the SI engine.
- As the amount of fuel injected per cycle is increased, problems with air utilization during combustion eventually lead to the formation of excessive amounts of soot which cannot be burned up prior to exhaust. This excessive soot or black smoke in the exhaust constrains the fuel/air ratio at maximum engine power to values some 30% lean of stoichiometric. Hence, the maximum indicated mean effective pressure (in a naturally-aspirated engine, or at a given boost level in a turbocharged engine) is lower than values for an equivalent spark-ignition engine operating at stoichiometric.
- Because the diesel always operates with lean fuel/air ratios (and at part load with very lean fuel/air ratios), the effective value of γ (= c_p/c_v) during the expansion process is higher than in a spark-ignition engine. This gives a higher fuel conversion efficiency than the spark-ignition engine, for a given expansion ratio (see Sec. 5.5.3).

A major challenge in diesel combustion chamber design is achieving sufficiently rapid mixing between the injected fuel and the air in the cylinder to complete combustion in the appropriate crank-angle interval relatively close to top-center. The foregoing discussion indicates (and more detailed analysis will confirm) that mixing rates control the fuel-burning rate. Commercial diesel engines are made in a range of cylinder sizes, with cylinder bores varying from about 70 to 900 mm. The mean piston speed at maximum rated power is approximately constant over this size range (see Sec. 2.2), so the maximum rated *engine* speed will be inversely proportional to the stroke [see Eq. (2.9)]. For a fixed crank-angle interval for combustion (of order 40 to 50 degrees CA to maintain high fuel conversion efficiency), the time available for combustion will, therefore, scale with the stroke. Thus, at the small end of the diesel engine size range, the mixing between the injected fuel and the air must take place on a time

scale some 10 times shorter than in engines at the large end of this range. It would be expected, therefore, that the design of the engine combustion chamber, the inlet port and valve, and the fuel-injection system would have to change substantially over this size range to provide the fuel and air motions inside the cylinder required to achieve the desired fuel-air mixing rate. As engine size decreases, more vigorous air motion is required while less fuel jet penetration is necessary. It is this logic, primarily, that leads to the different diesel combustion chamber designs and fuel injection systems found in practice over the large size range of production diesel engines.

10.2 TYPES OF DIESEL COMBUSTION SYSTEMS

Traditionally, diesel engines have been divided into two basic categories according to their combustion chamber design: (1) *direct-injection* (*DI*) *engines,* which have a single open combustion chamber into which fuel is injected directly and (2) *indirect-injection* (*IDI*) *engines*, where the chamber is divided into two regions and the fuel is injected into the "prechamber" which is connected to the main chamber above the piston crown via a nozzle, or one or more orifices. IDI engine designs were used extensively in smaller engine sizes, but have largely been replaced by high-speed DI engine designs. Within each category there are several different chamber geometry, air-flow, and fuel-injection arrangements.

10.2.1 Direct-Injection Systems

Figure 10.1 shows three basic DI combustion systems, categorized by their chamber geometries, air motions, and fuel spray arrangement. In the largest-size engines, where mixing rate requirements are least stringent, quiescent DI systems of the type shown in Fig. 10. 1 *a* are used. The momentum and energy of the injected fuel jets are sufficient to achieve adequate fuel distribution and rates of mixing with the air. Additional organized air motion is not required. The combustion chamber shape is usually a shallow bowl in the crown of the piston, and a central multi-hole injector is used.

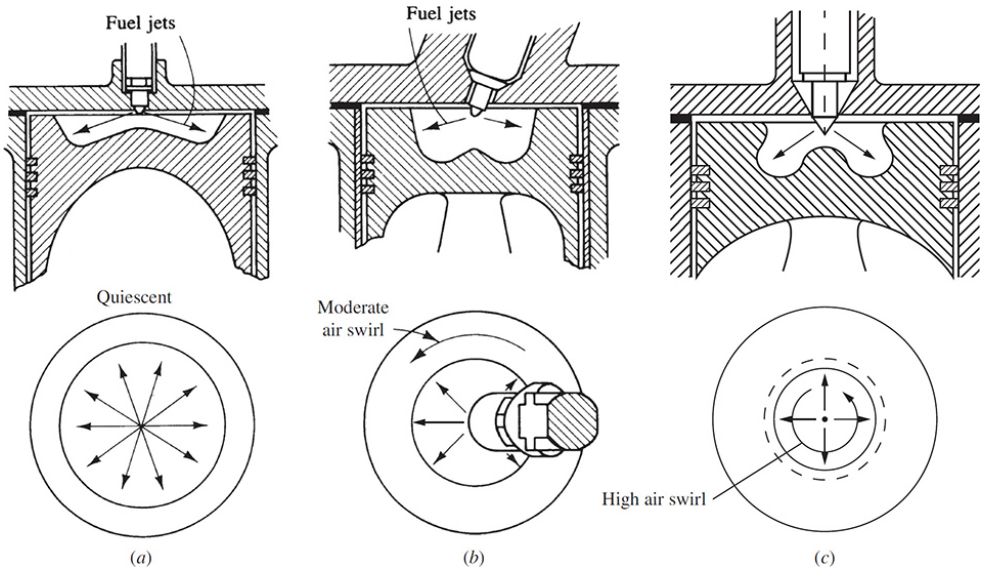


Figure 10.1 Common types of direct-injection diesel engine combustion systems: (*a*) quiescent chamber with multi-hole nozzle typical of larger engines; (*b*) bowl-in-piston chamber with moderate swirl and multi-hole nozzle; (*c*) re-entrant bowl-in-piston chamber with high swirl and multi-hole nozzle: (*b*) and (*c*) are used in medium and smaller DI engine size ranges, respectively.

As engine size decreases, increasing amounts of air swirl are used to achieve faster fuel-air mixing rates. Air swirl is generated by suitable design of the inlet port (see Sec. 8.3): the swirl rate can then be increased as the piston approaches TC by forcing the air toward the cylinder axis into a bowl-in-piston type of combustion chamber. Figures 10. l *b* and *c* show the two types of DI engine with swirl, in common use. Figure 10. l *b* shows a medium-sized DI engine with swirl, with a centrally located multi-hole injector nozzle. Here, the design goal is to hold the amount of liquid fuel which impinges on the piston bowl walls to a minimum. Figure 10. l *c* shows a small high-speed DI engine with a re-entrant bowl-in-piston combustion system, which uses high swirl and a multi-hole injector nozzle. These two types of designs are used in medium-size (10- to 15-cm bore) diesels and, with increased swirl, in small-size (7- to 9-cm bore) diesels.

10.2.2 Other Diesel Combustion Systems

Figure 10.2 *a* shows the M.A.N. wall-distribution combustion system that was developed for commercial vehicle engines to control the harsh combustion noise produced by earlier direct-injection diesels. With this Meurer or M-system, the single-hole fuel-injection nozzle sprays the fuel against the wall of the deep "hemispherical" bowl-in-piston combustion chamber, tangentially with the highly swirling in-bowl air motion. This results in a significantly slower fuel evaporation rate than that occurs in comparable multi-spray air distribution DI approach. With the now common use of turbocharging, and the advances in DI fuel-injection technology that have taken place, any benefits of this M system have largely disappeared.

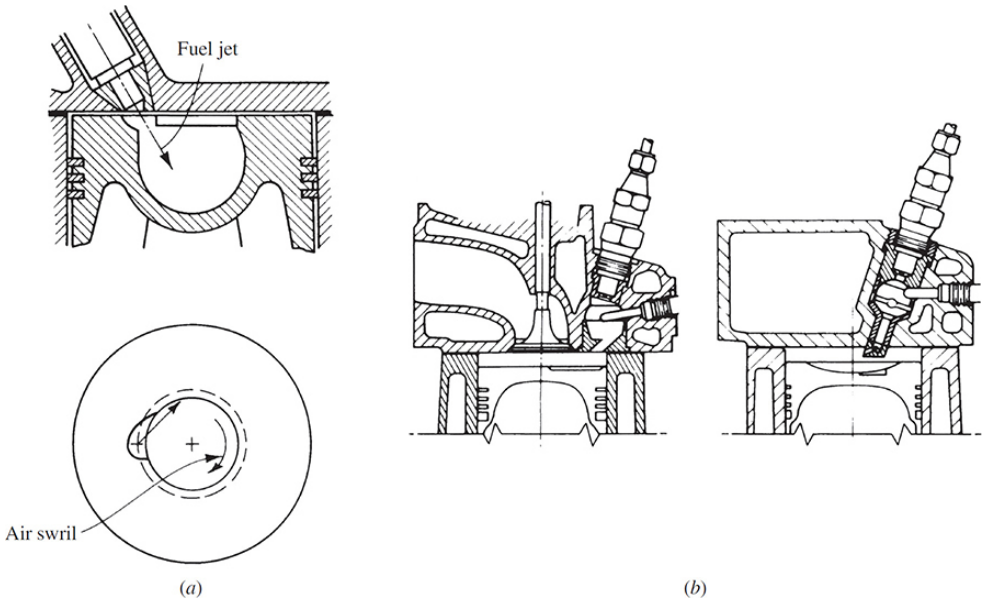


Figure 10.2 (*a*) Direct-injection: MAN, Meurer (or M) combustion system with deep bowl in piston, air swirl, and single-hole nozzle. (*b*) Two common types of small indirect-injection diesel engine combustion systems: left, swirl prechamber; right, turbulent prechamber.

Historically, as diesel combustion systems developed, use of air swirl to enhance fuel-air mixing rates, despite amplification as the air is forced into the piston bowl during compression, did not provide sufficiently high fuel-air mixing rates for small high-speed diesels such as those used in automobiles. Indirect-injection or divided-chamber combustion systems were used instead, where the vigorous charge motion required during fuel injection is generated in a prechamber, during the compression stroke. Two broad

classes of IDI systems evolved: (1) swirl chamber systems and (2) prechamber systems, as illustrated in Fig. 10.2 *b*. During compression, air is forced from the main chamber above the piston into the auxiliary chamber, through the nozzle or orifice (or set of orifices). Thus, toward the end of compression, a vigorous flow in the auxiliary chamber is set up; in swirl-chamber systems, the connecting passage and chamber are shaped so that this flow within the auxiliary chamber rotates rapidly.

Fuel was injected into the auxiliary chamber at lower injection-system pressure than was then typical of DI systems, through a pintle nozzle as a single spray: see Fig. 1.18. Combustion starts in the auxiliary chamber; the pressure rise associated with combustion forces fluid back into the main chamber where the jet issuing from the nozzle entrains and mixes with the main chamber air. The glow plug shown on the right of the prechambers in Fig. 10.2 *b* is a cold-starting aid. The plug is heated prior to and during the engine's starting process to ensure ignition of fuel early in the engine cranking process. Such indirect-injection, swirl or pre-chamber, engines are no longer mainstream diesel engine technologies.

10.2.3 Comparison of Different Combustion Systems

The number of different types of combustion chambers proposed and tried since the beginnings of diesel engine development is substantial. Over the years, however, through increased understanding of the physical and chemical processes involved and competition in the marketplace, only a limited number of different designs have survived. The important characteristics of those types of diesel engines now most commonly used are summarized in Table 10.1. The numbers for dimensions and operating characteristics are typical values for each different type of diesel combustion system.

TABLE 10.1 Characteristics of diesel combustion systems

System	Direct injection systems				Other combustion chambers	
	Quiescent multi-spray	Medium swirl multi-spray	High swirl multi-spray	High swirl "M"	Swirl chamber	Prechamber
Size	Largest	Medium	Smallest	Medium	Smallest	Smallest
Cycle	2-/4-stroke	4-stroke	4-stroke	4-stroke	4-stroke	4-stroke
TC/S/NA*	TC/S	TC/NA	NA/TC	TC/NA	NA/TC	NA/TC
Maximum speed, rev/min	120–2100	1800–3500	2500	4000–4500	3600–5000	4500
Bore, mm	900–150	150–100	100–75	130–100	95–70	95–70
Stroke/bore	3.5–1.2	1.3–1.0	1.1–0.9	1.2–0.9	1.1–0.9	1.1–0.9
Compression ratio	12–15	15–17	18–22	16–18	20–24	22–24
Chamber	Open or shallow dish	Bowl-in-piston	Re-entrant deep bowl-in-piston	Deep bowl-in-piston	Swirl pre-chamber	Single/multi-orifice prechamber
Air-flow pattern	Quiescent	Medium swirl	Highest swirl	High swirl	Very high swirl in prechamber	Very highly turbulent in prechamber
Number of nozzle holes	Multi 6–10	Multi 4–8	Multi 4	Single	Single	Single
Injection pressure	Very high	High	High	Medium	Lowest	Lowest

*Turbocharged/supercharged/naturally-aspirated.

The larger slowest speed engines used for power generation or marine applications utilize open quiescent chambers in the piston crown, which are close to disc shaped: the motion of the fuel jets is responsible for distributing and mixing the fuel with air. The largest of these are often two-stroke cycle engines. In the next size range, in large truck and locomotive engines, a quiescent chamber consisting of a shallow dish or bowl in the piston crown is used. The air utilization in these engines is low, but they are invariably supercharged and/or turbocharged to obtain high power density.

In the DI category, as engine size decreases and maximum speed rises, swirl is increasingly used to obtain high-enough fuel-air mixing rates. The swirl is generated by suitably shaped inlet ports, and is amplified during compression by forcing most of the air toward the cylinder axis into the deep bowl-in-piston combustion chamber. High-speed smaller versions of the DI diesel have been successfully developed over the past 35 or 50 years for the automobile market. These use high intake-generated swirl, augmented by the radial inward displacement of the air, and the vigorous squish flow the deep re-entrant bowl-in-piston combustion chamber produces, to achieve fast enough fuel-air mixing and combustion rates to operate up to engine speeds of 4500 to 5000 rev/min (lower than maximum speeds for gasoline SI engines, but producing acceptable specific maximum engine power by turbocharging to high boost levels).

Table 10.1 lists the characteristics of three other diesel combustion systems for historical reasons: they are no longer widely used. An alternative to the multi-hole-nozzle swirl system used in medium and heavy-duty truck

engines is the M system (or wall-wetting system), where most of the fuel from the single-hole pintle nozzle is placed on the wall of the spherical bowl in the piston crown. Historically, in the smallest engine sizes, IDI systems were used to obtain the vigorous auxiliary-chamber air motion required for high fuel-air mixing rates. Two different types of auxiliary-chamber geometries were used: a swirl chamber and a turbulence-generating prechamber as shown in Fig. 10.2 *b*.

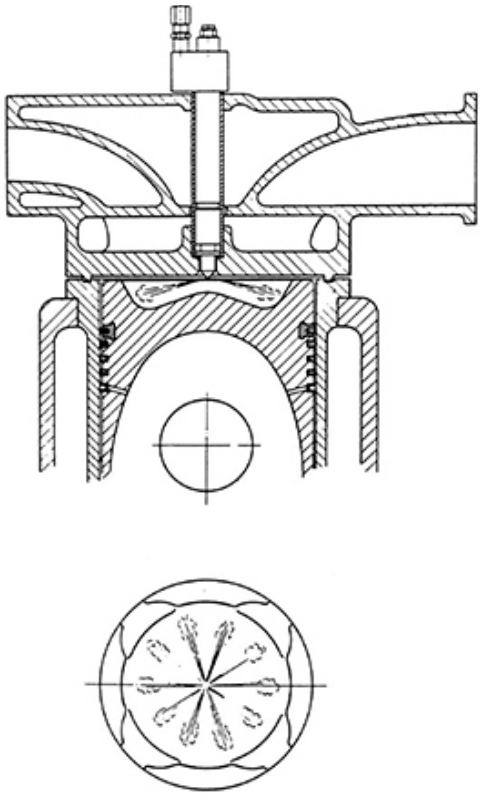
10.3 DIESEL ENGINE COMBUSTION

Studies of diesel engine combustion using optical diagnostics to visualize fuel spray and flame behavior, combined with analyses of engine cylinder pressure data, have led to a widely accepted descriptive model of the compression-ignition engine combustion process. The concept of *heat-release rate* is important to understanding diesel combustion. Its basic definition is the rate at which the chemical energy of the fuel is released by the combustion process. It is usually calculated from cylinder pressure versus crank-angle data, as the energy release rate required to create the measured pressure, using the techniques described in Sec. 10.4 . [b] The combustion model defines several distinct phases of diesel combustion, each controlled by different physical and chemical processes. Although the relative importance of each phase does depend on the combustion system used, and on engine operating conditions, four phases are common to all diesel engines as discussed below.

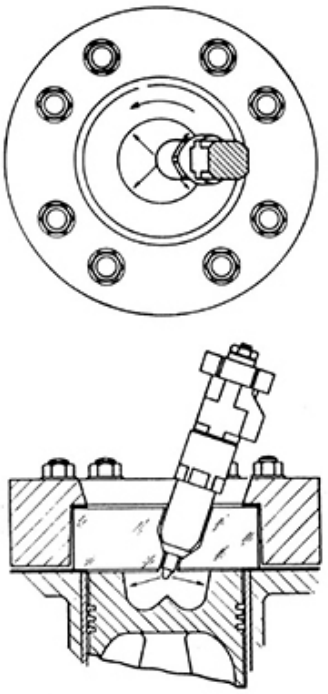
10.3.1 Optical Studies of Diesel Combustion

Various optical techniques have been used to visualize the diesel combustion process. Many of these studies have been carried out in combustion chambers close to those used in practice, under normal engine operating conditions. Sequences of images over time provide valuable information on the evolution of the combustion process in the different types of engines. Figure 10.3 shows four combustion chamber systems that were studied photographically several decades ago which helped develop earlier combustion models. These are: (*a*) a quiescent chamber typical of the large

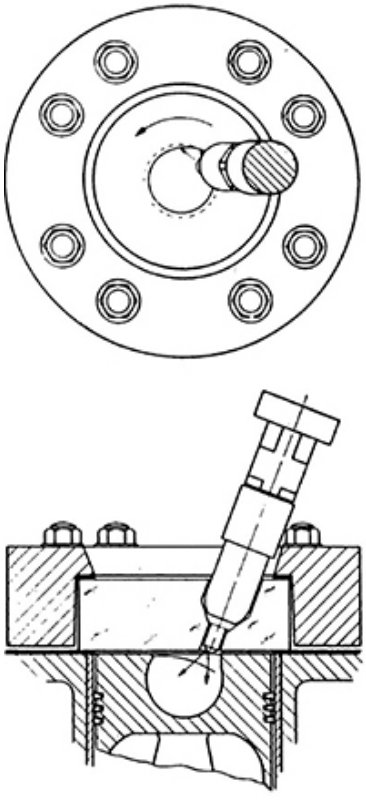
diesel engines used for industrial, marine, and rail traction applications (only the burning of a single fuel spray of the multi-spray combustion system could be studied); (*b*) a medium-speed DI engine with swirl and four fuel jets centrally injected; (*c*) an M.A.N. “M” DI system; and (*d*) a Ricardo Comet V swirl-chamber IDI system. [1]



(*a*)



(*b*)



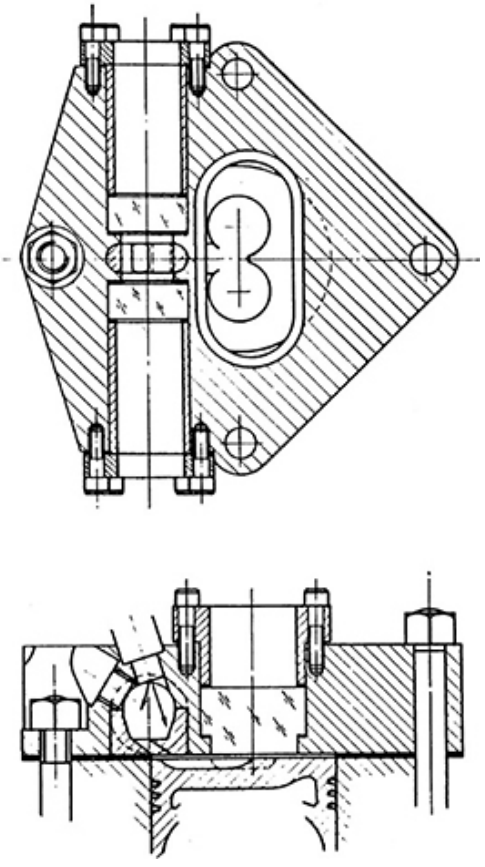




Figure 10.3 Four diesel combustion chambers used to obtain the photographs of the compression-ignition combustion processes shown in Fig. 10.4: (*a*) quiescent DI chamber; (*b*) multi-hole nozzle DI chamber with swirl; (*c*) M.A.N. "M" DI chamber; (*d*) Ricardo Comet IDI swirl chamber.[2, 3]

The combustion sequences in Fig. 10.4, originally recorded on color film, show the following features:

Fuel sprays . The fuel droplets reflect light from spot lamps and define the extent of the liquid fuel spray prior to complete vaporization.

Premixed burning. These regions are of too low a luminosity to be recorded with the exposure level used. The addition of a copper additive to the fuel gives these normally blue flames a green color bright enough to render them visible.

Diffusion flame. The burning high-temperature carbon particles in this flame (which separates fuel on the inside of the flame from the air outside) provide more than adequate luminosity and appear as yellow-white. As the flame cools, the radiation from the oxidizing particles changes color through orange to red.

Over-rich mixture chemistry. The appearance of brown regions, usually surrounded by a white diffusion flame, indicates excessively rich mixture regions where substantial soot particle production is occurring. Where this fuel-rich soot-laden cloud contacts air, there is a hot (white) diffusion flame.

Table 10.2 summarizes the characteristics of these different regions, discernable in the photographs shown in the Fig. 10.4.

TABLE 10.2 Interpretation of diesel engine combustion photographs [1, 2]

Shade/color*	Interpretation
Dark grey	Background; the gas (air in early stages, combustion products later) is transparent and not glowing
Medium grey (green)	Early in combustion process; low luminosity "premixed" rendered visible by copper added to fuel. Later; burned gas above about 1800°C
Bright white (yellow-tinged)	Carbon particle burnup in diffusion flame, 2000–2500°C
Less-bright white (yellow, orange-red)	Carbon burnup in diffusion flame at lower temperatures; last visible in film at 1000°C
Darker grey regions within white regions (brown)	Soot clouds forming in very fuel-rich mixture regions. Where these meet air (grey) there is always a white fringe of hot flame

*Original photos are in color.[1]

Figure 10.4 *a* shows a sequence of photographs from one combustion event of the single spray, burning under conditions typical of a large quiescent DI engine. The fuel spray is shown penetrating into the chamber. Ignition occurs at −8° in the fuel-air mixture left behind on the edge of the spray not far from the injector. A flame then spreads rapidly along the outside of the spray to the spray tip where some of the fuel, which has had a long residence time in the chamber, burns with a low-luminosity flame (colored green by the copper fuel additive in the original photos). The diffusion flame engulfing the remainder of the spray is brilliant white with a yellow tinge from the burning of the soot particles which have been formed in the fuel-rich spray core. At this stage (−1°), about 60% of the fuel has been injected. The remainder is injected into this enflamed region, producing a very fuel-rich zone apparent as the brown cloud (11°). This soot cloud moves to the outer region of the chamber (11 to 20°): white-yellow flame activity continues near the injector, probably due to combustion of ligaments of fuel, which issued from the injector nozzle as the injector needle was seating. Combustion continues well into the expansion stroke (31°).

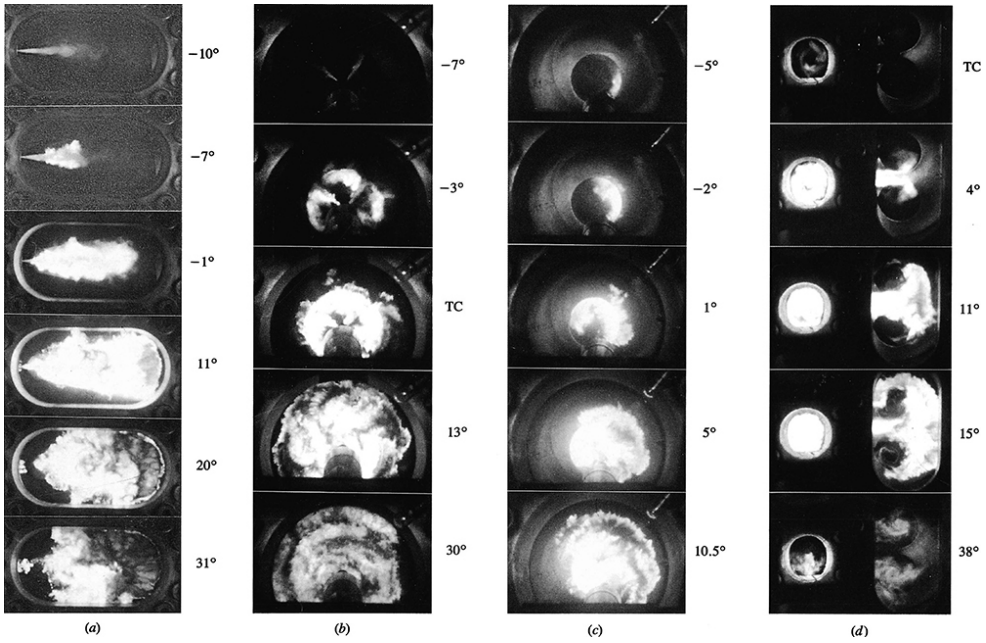


Figure 10.4 Sequence of photographs (originally in color) from high-speed movies taken in the special visualization diesel engines shown in Fig. 10.3: (*a*) combustion of single spray burning under large DI engine conditions; (*b*) combustion of four sprays in DI engine with counterclockwise swirl; (*c*) combustion of single spray in M.A.N. "M" DI diesel; (*d*) combustion in prechamber (on left) and main chamber (on right) in Ricardo Comet IDI swirl chamber diesel. 1250 rev/min, imep = 827 kPa (120 lb/in.) [1,2] (*Courtesy: Ricardo Consulting Engineers*.)

This sequence shows that fuel distribution is highly nonuniform during the combustion process in this type of DI engine. Also, the air between the individual fuel sprays of the quiescent open-chamber diesel mixes with each burning spray relatively slowly, contributing to poor air utilization with this type of combustion chamber.

Figure 10.4 *b* shows a combustion sequence from the DI engine with swirl (the chamber shown in Fig. 10.3*b*). The inner circle corresponds to the deep bowl in the piston crown, the outer circle to the cylinder liner. The fuel sprays (of which two are visible without obstruction from the injector) first appear at −13°. At −7° they have reached the wall of the bowl; the tips of the sprays have been deflected slightly by the anticlockwise swirl. The frame at −3° shows the first ignition. Bright luminous flame zones are visible, one on each spray. Out by the bowl walls, where fuel vapor has been blown around

by the swirl, larger burning regions (tinged green in the original photos) indicating the presence of premixed combustion can be seen. The fuel downstream of each spray is next to ignite, burning yellow-white due to the soot formed in the richer mixture. Flame propagation back to the injector follows extremely rapidly and at TC the bowl is filled with flame. At 5° ATC the flame spreads out over the piston crown toward the cylinder wall due to combustion-produced gas expansion and the reverse squish flow (see Sec. 8.5). The darker grey (brown) regions (13°) are soot-laden fuel-rich mixture originating from the fuel which impinges on the wall. The last frame (30° ATC) shows the gradual diminution of the soot-particle-laden regions as they mix with the excess air and burn up. The last dull-red burning visible on the film is at about 75° ATC, well into the expansion stroke.

Figure 10.4 *c* shows the combustion sequence for the M-type DI engine.[4] In the version of the system used for these experiments, the fuel was injected through a two-hole nozzle which produces a main jet directed tangentially onto the walls of the spherical cup in the piston crown, and an auxiliary spray which mixes a small fraction of the fuel directly with the swirling airflow. At −5° the fuel spray is about halfway round the bowl. Ignition has just occurred of fuel adjacent to the wall, which has mixed sufficiently with air to burn. The flame spreads rapidly (−2°, 1°) to envelop the fuel spray and is convected round the cup by the high swirl air flow. By shortly after TC the flame has filled the bowl and is spreading out over the piston crown. A soot cloud is seen near the top right of the picture at 5° ATC which spreads out around the circumference of the enflamed region. There is always a rim of flame between the soot cloud and the cylinder liner as excess air is mixed into the flame zone (10.5°). The flame is of the carbon-burning diffusion-flame type throughout; little premixed combustion (green in the original photos) is seen, even at the beginning of the burning process.

Figure 10.4 *d* shows the combustion sequence in a swirl chamber IDI engine of the Ricardo Comet V design. The swirl chamber (on the left) is seen in the view of the lower drawing of Fig. 10.3 *d* (with the connecting passageway entering the swirl chamber tangentially at the bottom left to produce clockwise swirl). The main chamber is seen in the plan view of the upper drawing of Fig. 10.3 *d*. Two sprays emerge from the injector nozzle after the SOI at −11°. The smaller, radial auxiliary spray is sharply deflected by the high air swirl. Frame 1 shows how the main spray follows the contour of the chamber; the auxiliary spray has evaporated and can no longer be seen.

Green-colored emission is first apparent at −1° from combustion of the vaporized fuel from the auxiliary spray and indicates premixed burning. Combustion then spreads to the main spray (TC), becoming a yellow-white carbon-particle-burning flame with a green fringe. At 4° ATC the swirl chamber appears full of carbon-burning flame, which is being blown through the throat and into the recesses in the piston crown by the combustion-generated pressure rise in the prechamber. The flame jet impinges on the piston recesses entraining the air in the main chamber, leaving grey patches (green in the original) where all the carbon has burned out (4°, 11°, 15° ATC). A darker grey (brown) soot cloud is emerging from the throat. By 15° ATC this soot cloud has spread around the cylinder, with a bright yellow-white flame at its periphery as this soot finds excess air and burns up. The yellow-white flame becomes yellow and then orange-red as the soot particles and burned gases cool on expansion. By 38° ATC most of the flame is burnt out.

Figure 10.5 shows additional details of the diesel spray development and combustion process. A high-pressure diesel fuel-injection system was used to inject a single fuel spray (similar to one spray exiting a multi-hole nozzle) into a constant-volume combustion vessel containing a quiescent gas (with 21% oxygen) at a temperature and pressure corresponding to diesel engine end-of-compression conditions. [c] Visible light emission from shadowgraph optical images of the jet was recorded with appropriate optical equipment. Figure 10.5 shows a sequence of four spray images from one experiment, as the injected fuel atomizes, penetrates, entrains air, vaporizes, ignites, and burns. The figure caption and labels describe the individual spray and flame images. An enlargement of one image toward the end of injection identifies more clearly the liquid fuel containing (early) region of the spray, the lift-off of the diffusion flame from the nozzle hole exit, the (orange/yellow) turbulent unsteady diffusion flame with fuel-rich partly reacted fuel combustion products on the inside of the flame which is made luminous by the burn up of soot particles within the flame, and air mixed with the hot burned gases diffusing outward from the diffusion flame seen as the turbulent irregularly bounded darker region identified by the shadowgraph optical technique. Overall, the injected liquid's momentum propels the fuel droplets into the combustion chamber like a "shotgun blast," entraining high-temperature air into the developing spray, causing the fuel drops to evaporate (the liquid spray core only penetrates partway). The initial momentum continues to

move the now largely gaseous spray across the combustion chamber, entraining more hot air as it progresses. A flame is first visible fairly close to the injector nozzle and rapidly spreads around the spray. Its color (white-yellow-orange) indicates that the luminosity is emitted from small soot particles formed in the (fuel-rich) partly reacted mixture inside the spray as these particles diffuse into the flame and burn up. The irregularities in the diffusion flame surface (and in the boundary of the hot burned gases diffusing outward into the air surrounding the diffusion flame) clearly indicate that all these processes are turbulent.

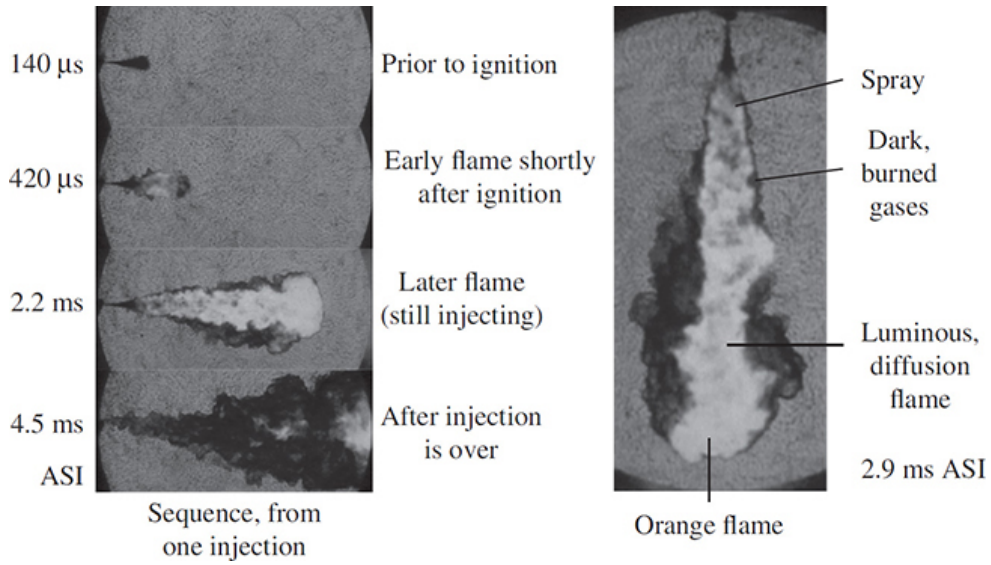


Figure 10.5 Shadowgraph and flame radiation emission images of combustion of a single diesel spray in a constant volume combustion facility at typical diesel engine charge conditions at TC. Sequence of four images: spray prior to ignition, early spray and diffusion flame, later spray and flame, spray and flame after injection has ended. Separate image on right shows hot burned gas (dark) diffusing outward from the diffusion flame. ASI, time after start of fuel injection. (*Courtesy: Dr. Dennis Siebers, Sandia National Laboratories*.)

10.3.2 Combustion in Direct-Injection Multi-Spray Systems

Figure 10.6 shows typical data for cylinder pressure (p), fuel injection rate, and net heat release rate through the compression and expansion strokes of a

DI diesel. The engine had central fuel injection through a multi-hole nozzle into a bowl-in-piston combustion chamber. The rate of fuel injection is obtained from the fuel-line pressure, cylinder pressure, nozzle geometry, and needle-lift profiles by considering the injector as one or more flow restrictions. [5] There is a delay of several degrees between the SOI and SOC [identified by the change in slope of the $p(\theta)$ curve]. The pressure rises rapidly for a few crank-angle degrees, then more slowly to a peak value some 5° after start-of-injection. Injection usually continues well into the combustion process. A rate-of-heat-release diagram, [d] corresponding to this rate of fuel injection and the cylinder pressure data, is also shown in Fig. 10.6. The general shape of the rate-of-heat-release curve is typical of this type of multi-spray DI engine. There is a short *ignition delay* between the SOI at −20° and the SOC at about −15°. The slight loss of "heat" during the delay period is due to heat transfer into the liquid fuel, vaporizing and heating it, and heat transfer to the walls. During the combustion process the burning proceeds in three distinguishable stages. In the first stage, the rate of burning is rapid, lasts for only a few crank-angle degrees, and produces an increasing and then decreasing "spike" in the heat-release rate. It corresponds to the period of rapid cylinder pressure rise. The second stage produces an increasing and then decreasing heat-release rate creating a second more rounded profile, as shown in Fig. 10.6. This is the main heat-release period and usually lasts about 30 to 40°. Up to about 90% of the total fuel energy is released in these first two periods. The "tail" of the heat-release rate diagram during which the remainder of the fuel's chemical energy is released results from the mixing of the excess air that was not involved in the main combustion process with the burned gases, and equilibrating.

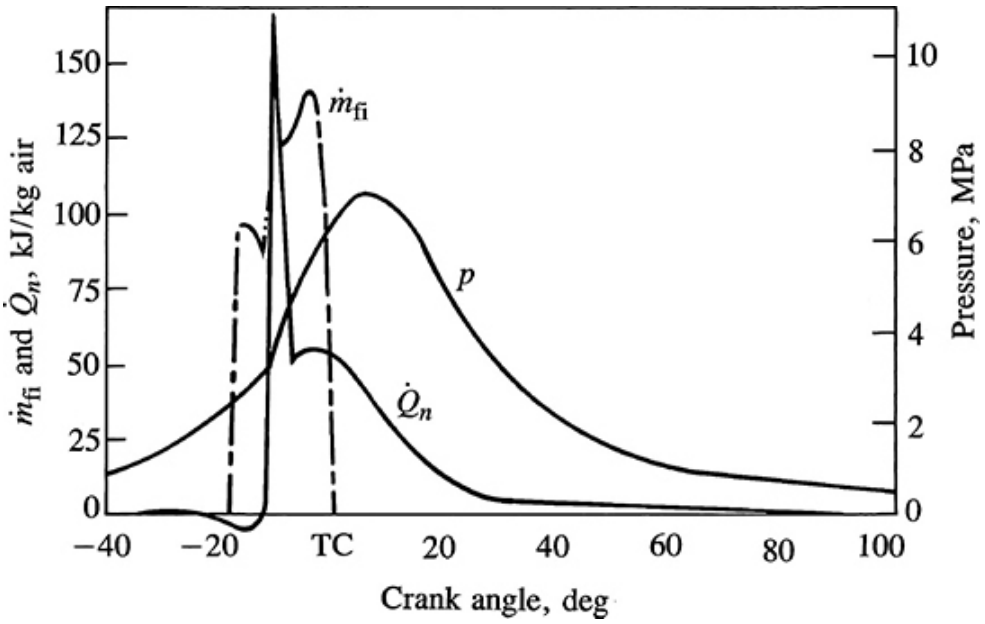


Figure 10.6 Cylinder pressure p, rate of fuel injection and net heat-release rate $\dot{Q}_n$ calculated from p, for a small DI diesel engine, as functions of crank angle: 1000 rev/min, normal injection timing, bmep = 620 kPa.[5]

From studies of rate-of-injection and heat-release diagrams such as those in Fig. 10.6, over a range of engine loads, speeds, and injection timings, Lyn and Austen [5, 6] developed the following summary. First, the total burning period is significantly longer than the fuel injection period. Second, the absolute burning rate increases proportionally with increasing engine speed; thus on a crank-angle basis, the burning interval remains (for a given amount of fuel injected) essentially constant. Third, the magnitude of the initial peak of the burning-rate diagram depends strongly on the duration of the ignition delay period, being higher for longer delays. These considerations, coupled with engine combustion photographic studies, lead to the following overall framework for diesel combustion.

Figure 10.7 shows schematically the rate-of-injection and rate-of-burning diagrams, where the injected fuel as it enters the combustion chamber has been divided into a number of elements. The first element which enters mixes with air and becomes "ready for burning" (i.e., mixes to within combustible limits), as shown conceptually by the lowest triangle along the abscissa in the rate-of-burning figure. While some of this fuel element mixes rapidly with air, the rest will mix more slowly. The second and subsequent elements will

mix with air in a similar manner, and the total "ready-for-burning" diagram, enclosed by the dashed line, is obtained. The total area of this diagram is equal to that of the rate-of-injection diagram. Ignition does not occur until after the delay period is over, however. At the ignition point, some of the fuel already injected has mixed with enough air to be within the combustible limits. That "premixed" fuel-air mixture (the shaded region in Fig. 10.7) is then added to the mixture which becomes ready for burning after the end of the delay period, producing the high initial rate of burning as shown. Such a heat-release profile is generally observed with this type of naturally-aspirated DI diesel engine. When the ignition delay is long, the amount of premixed "ready-to-burn" mixture is larger, leading to a high peak rate of heat release. When the delay is short (e.g., in a warmed-up turbocharged DI diesel) the amount of premixed mixture is, relatively, much less and this premixed rate of heat release spike is much lower. [e]

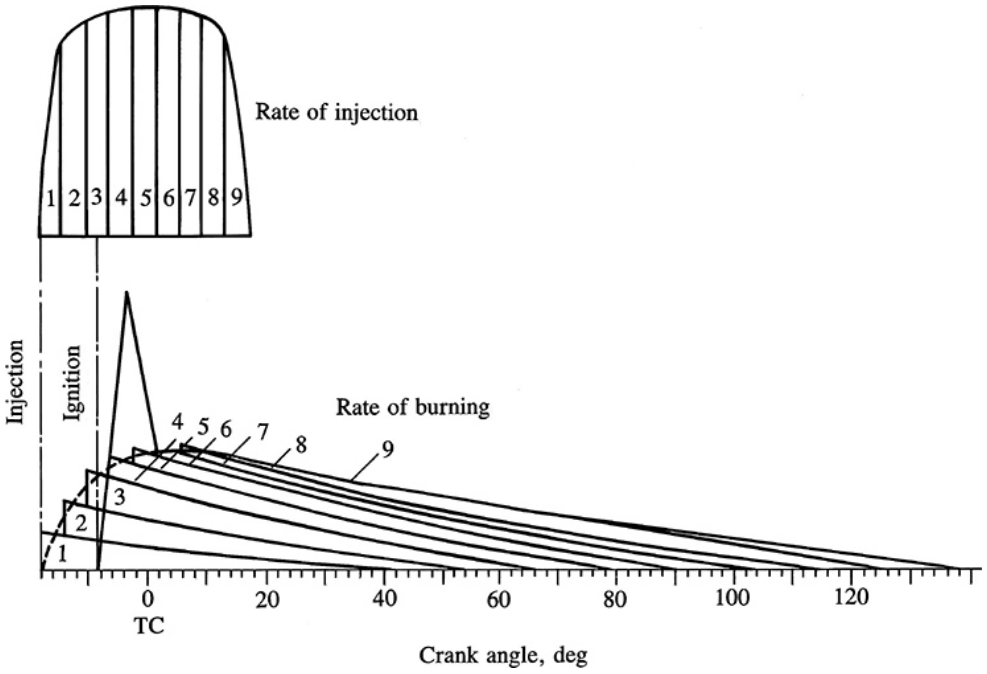


Figure 10.7 Schematic of relationship between rate of fuel injection and rate of fuel burning or chemical energy release.[6]

We therefore identify the following stages of the overall compression-ignition diesel combustion process on a heat-release-rate diagram for a DI engine (Fig. 10.8).

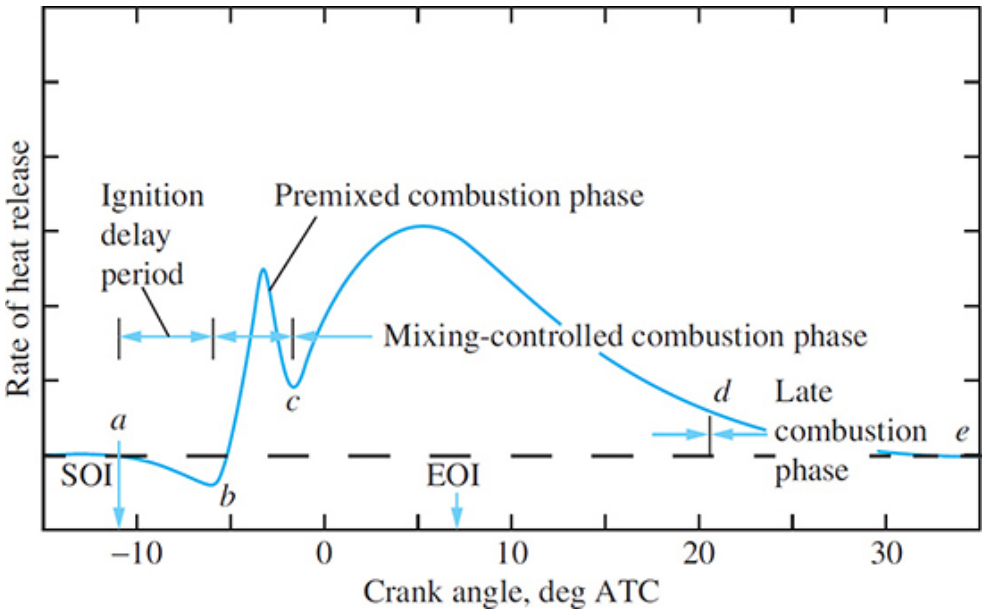


Figure 10.8 Typical DI engine heat-release-rate diagram identifying the different phases of the diesel combustion process.

An *ignition delay* (*ab*). The period between the start of fuel injection into the combustion chamber and the SOC determined from the change in slope on the p-θ diagram, or from a heat-release analysis of the $p(\theta)$ data, or from a luminosity detector.

A *premixed or rapid combustion phase* (*bc*). Here, chemical energy release from the fuel which has mixed with air to within combustible limits during the ignition delay period occurs rapidly in a few crank-angle degrees. When this burning mixture is added to the fuel which becomes ready for burning and burns during this phase, the high heat-release rates characteristics of this phase result. This rapid energy release comes from combustion of the fuel-rich mixture within each spray.

A *mixing-controlled combustion phase* (*cd*). Once the fuel and air which premixed during the ignition delay has released some of its chemical energy, the burning rate (or heat-release rate) is then controlled by the rate at which mixture becomes available for burning. While several processes are involved—liquid fuel atomization, vaporization, mixing of fuel vapor with air, rich-mixture chemical reactions within the spray—the rate of burning in this diffusion-flame phase of diesel combustion is controlled primarily by the fuel vapor-air mixing process.

A *late combustion phase* (*de*). Heat release continues at a lower and decaying rate into the expansion stroke. There are several causes. A small fraction of the fuel may not yet have burned. A fraction of the fuel energy is present in soot and fuel-rich combustion products and can still be released. The cylinder charge is nonuniform and mixing of burned gases with air during this period promotes more complete combustion and less-dissociated product gases. The kinetics of the final burnout processes become slower as the temperature of the cylinder gases fall during expansion.

10.3.3 Heat-Release-Rate Analysis

Cylinder pressure versus crank-angle data over the compression and expansion strokes of the engine operating cycle can be used to obtain quantitative information on the progress of combustion. Suitable methods of analysis which yield the rate of release of the fuel's chemical energy, usually called the *heat release rate* or rate of fuel burning, through the diesel engine combustion process will now be described. The methods of analysis are similar to those described in Sec. 9.2.2 for spark-ignition engines and start with the first law of thermodynamics for an open system which is quasi static (i.e., uniform in pressure and temperature). The first law for such a system (see Fig. 9.11) is

$$\frac{dQ}{dt} - p\frac{dV}{dt} + \sum_i \dot{m}_i h_i = \frac{dU}{dt} \tag{10.1}$$

where dQ/dt is the heat-transfer rate across the system boundary into the system, $p(\ dV/dt)$ is the rate of work transfer done by the system due to system boundary displacement, $\dot{m}\,i$ is the mass flow rate into the system across the system boundary at location i (flow out of the system would be negative), h_i is the enthalpy of flux i entering or leaving the system, and U is the energy of the material contained inside the system boundary.

The following factors make the application of this equation to diesel combustion difficult:

- Fuel is injected into the cylinder. Liquid fuel is added to the cylinder, which vaporizes and mixes with air to produce a fuel/air ratio distribution which is nonuniform and varies with time.

- The composition of the burned gases (also nonuniform) is not known.
- The accuracy of available correlations for predicting heat transfer in diesel engines is not well defined (see Chap. 12).
- Crevice regions (such as the volumes between the piston, rings, and cylinder wall) constitute a few percent of the clearance volume. The gas in these regions, however, is cooled to close to the wall temperature, increasing its density and, therefore, the relative importance of these crevices. Thus, crevices increase heat transfer and contain a non-negligible fraction of the in-cylinder air during combustion at a density significantly higher than in the rest of the combustion chamber.

Due to difficulties in dealing with these problems, methods of heat-release or burn-rate analysis give only approximate answers.

Combustion Efficiency

In both heat-release and fuel mass burned estimations, an important factor is the completeness of combustion. Air utilization in diesels is limited by the onset of black smoke in the exhaust. The smoke is soot particles, which are mainly carbon. Soot and other incomplete combustion products such as unburned hydrocarbons and carbon monoxide represent a combustion inefficiency. Normally, combustion efficiency is calculated from the combustibles in the exhaust. However, torque loss is more appropriately estimated from the inefficiency resulting from soot, CO and HC present in the cylinder at about the end of combustion; only modest additional torque is generated beyond the first one-third of the expansion stroke. Based on exhaust concentrations, the magnitude of that inefficiency is small. At full load conditions, if only 0.5% of the fuel supplied was present in the exhaust as black smoke, the result would be unacceptable. Soot emissions from the cylinder contain less than 0.1% of the fuel's chemical energy. Typically, hydrocarbon emissions are the order of 0.2% of the fuel. The energy corresponding to the exhausted carbon monoxide is about 0.5%. Thus, the combustion inefficiency [Eq. (4.69)] is usually less than 1%: the combustion efficiency is usually greater than about 99% (see Fig. 3.9). While these emissions are important in terms of their air-pollution impact (see Chap. 11), from the point of view of energy conversion it is a good approximation to regard combustion and heat release in diesels as essentially complete.

Heat-Release Analysis: Direct-Injection Engines

For this type of engine, the cylinder contents are a single open system. The only mass flows across the system boundary (while the intake and exhaust valves are closed) are the fuel injection and the crevice flows. An approach, which incorporates the crevice flow, has been described in Sec. 9.2.2 ; crevice flow effects will be omitted here. Equation (10.1) therefore becomes

$$\frac{dQ}{dt} - p\frac{dV}{dt} + m_f h_f = \frac{dU}{dt} \tag{10.2}$$

Two common methods are used to obtain combustion information from pressure data via Eq. (10.2) . In both approaches, the cylinder contents are assumed to be at a uniform temperature at each instant in time during the combustion process. This commonly used method yields the net fuel energy- or heat-release rate; the other method yields a fuel mass burning rate. The term *apparent* is often used to describe these quantities since both are approximations to the real quantities, which cannot be determined exactly.

If U and h_f in Eq. (10.2) are taken to be the sensible internal energy of the cylinder contents and the sensible enthalpy of the injected fuel, respectively,[f] then dQ/dt becomes the difference between the chemical energy or "heat" released by combustion of the fuel (a positive quantity) and the heat transfer from the system (in engines, the heat transfer is from the system, and by thermodynamic convention is a negative quantity). Since $h_{s,f} \approx 0$, Eq. (10.2) becomes

$$\frac{dQ_n}{dt} = \frac{dQ_{ch}}{dt} - \frac{dQ_{ht}}{dt} = p\frac{dV}{dt} + \frac{dU_s}{dt} \tag{10.3}$$

The *net heat-release* rate, dQ_n/dt, which is the difference between the *gross heat-release* rate dQ_{ch}/dt and the heat-transfer rate to the walls dQ_{ht}/dt, equals the rate at which work is done on the piston plus the rate of change of sensible internal energy of the cylinder contents.

If we further assume that the contents of the cylinder can be modeled as an ideal gas, then Eq. (10.3) becomes

$$\frac{dQ_n}{dt} = p\frac{dV}{dt} + mc_v\frac{dT}{dt} \tag{10.4}$$

From the ideal gas law, $pV = mRT$, with R assumed constant, it follows that

$$\frac{dp}{p} + \frac{dV}{V} = \frac{dT}{T} \tag{10.5}$$

Equation (10.5) can be used to eliminate T from Eq. (10.4) to give

$$\frac{dQ_n}{dt} = \left(1 + \frac{c_v}{R}\right)p\frac{dV}{dt} + \frac{c_v}{R}V\frac{dp}{dt}$$

or

$$\frac{dQ_n}{dt} = \frac{\gamma}{\gamma - 1}p\frac{dV}{dt} + \frac{1}{\gamma - 1}V\frac{dp}{dt} \tag{10.6}$$

Here γ is the ratio of specific heats, c_p/c_v. An appropriate range for γ for diesel heat-release analysis is 1.3 to 1.35; Eq. (10.6) is often used with a constant value of γ within this range. More specifically, we would expect γ for diesel engine heat-release analysis to have values appropriate to air temperatures at end-of-compression-stroke prior to combustion (≈1.35) and to burned gases at the overall equivalence ratio following combustion (≈1.26–1.3). The appropriate values for γ during combustion, which would give most accurate heat-release information, are not well defined. [7, 8]

More complete methods of heat-release analysis based on Eq. (10.2) have been proposed and used. These include more sophisticated models for the gas properties before, during, and after combustion, and for heat transfer and crevice effects. [8] However, it is also necessary to deal with the additional issues of: (1) mixture nonuniformity (fuel/air ratio nonuniformity *and* burned and unburned gas nonuniformities); (2) accuracy of any heat-transfer model used (see Chap. 12); and (3) the effects of the crevice regions. These additional phenomena must be dealt with at an equivalent level of accuracy for more complex heat-release models to be worthwhile. For many

engineering applications, Eq. (10.6) is adequate for diesel engine combustion analysis.

Additional insight can be obtained by incorporating a model for the largest of the effects omitted from Eq. (10.6) , the heat transfer rate dQ_{ht}/dt (see Chap. 12): we thereby obtain a close approximation to the *gross* heat-release rate. The integral of the gross heat-release rate over the complete combustion process should then equal (to within a few percent since the analysis is not exact) the mass of fuel injected m_f times the fuel's lower heating value, Q_{LHV}: that is,

$$Q_{ch} = \int_{t_{start}}^{t_{end}} \frac{dQ_{ch}}{dt} dt = m_f Q_{LVH} \tag{10.7}$$

Of course, Eqs. (10.1) to (10.4), (10.6), and (10.7) also hold with crank-angle θ as the independent variable instead of time t.

Figure 10.9 illustrates the relative magnitude of gross and net heat release, heat transfer, crevice effects, and heat of vaporization and heating up of the fuel for a turbocharged DI diesel engine operating in the mid-load, mid-speed range. The net heat release is the gross heat release due to combustion, less the heat transfer to the walls, crevice effects, and the effect of fuel vaporization and heatup. (It was omitted above by neglecting the mass addition term in dU/dt.) This last term is sufficiently small to be neglected. The enthalpy of vaporization of diesel fuel is less than 1% of its heating value: the energy change associated with heating up and vaporizing the fuel from injection temperature to typical end-of-compression temperatures is about 3% of the fuel heating value. The heat transfer integrated over the duration of the combustion period is 10 to 25% of the total heat released.

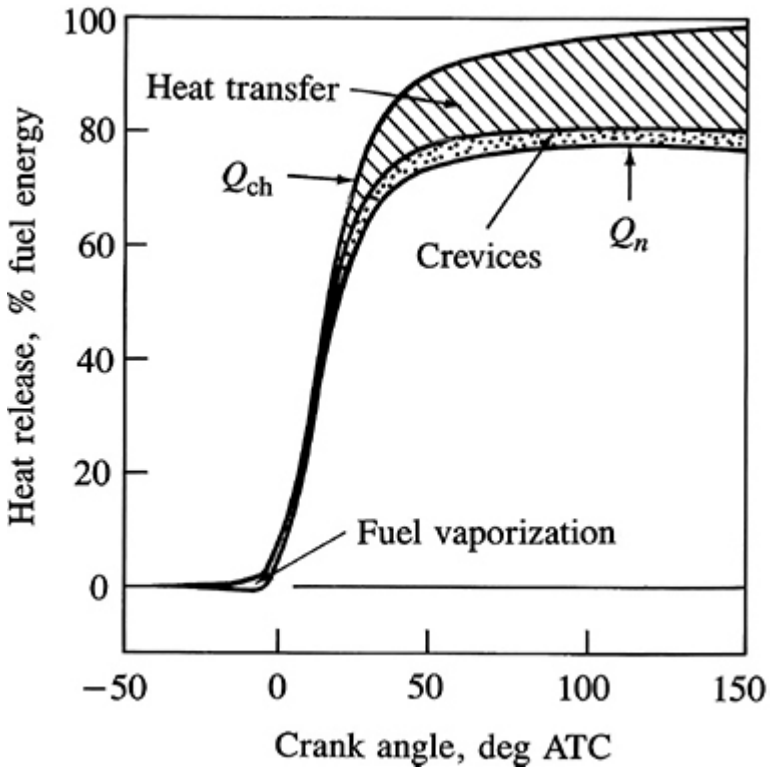


Figure 10.9 Gross and net heat-release profiles during combustion, for a turbocharged DI diesel engine at mid-load, mid-speed, showing relative magnitude of heat transfer, crevice, and fuel vaporization and heat-up effects.

Fuel Mass Burning Rate Analysis

If the internal energies of the fuel, air, and burned gases in Eq. (10.1) are evaluated relative to a consistent datum (such as that described in Sec. 4.5.2), then this equation can be used to obtain an apparent *fuel mass burning rate* from cylinder pressure versus crank-angle data. (With such a species energy datum the "heat release" is accounted for in the internal energy and enthalpy terms.) Following Krieger and Borman, [9] Eq. (10.2) can be written as

$$\frac{d}{dt}(mu) = p\frac{dV}{dt} + \frac{dQ}{dt} + h_f\frac{dm}{dt} \tag{10.8}$$

Here Q is the heat transfer to the gas within the combustion chamber (i.e., $Q = -Q_{ht}$), m is the mass within the combustion chamber, and dm/dt has been substituted for $\dot{m}_f$.

Since the properties of the gas mixture in the cylinder during combustion (assumed to be uniform and in chemical equilibrium at the pressure p and average temperature T) are in general a function of p, T, and the equivalence ratio ϕ:

$$u = u(T, p, \phi) \text{ and } R = R(T, p, \phi)$$

Therefore,

$$\frac{du}{dt} = \frac{\partial u}{\partial T}\frac{dT}{dt} + \frac{\partial u}{\partial p}\frac{dp}{dt} + \frac{\partial u}{\partial \phi}\frac{d\phi}{dt} \tag{10.9a}$$

$$\frac{dR}{dt} = \frac{\partial u}{\partial T}\frac{dT}{dt} + \frac{\partial R}{\partial p}\frac{dp}{dt} + \frac{\partial \mathrm{R}}{\partial \phi}\frac{d\phi}{dt} \tag{10.9b}$$

Also,

$$\phi = \phi_0 + \left(\frac{m}{m_0} - 1\right)\frac{1 + (F/A)_0}{(F/A)_s} \tag{10.10}$$

and,

$$\frac{d\phi}{dt} = \frac{1 + (F/A)_0}{(F/A)_s m_0}\frac{dm}{dt} \tag{10.11}$$

where (F/A) is the fuel/air ratio; the subscript 0 denotes the initial value prior to fuel injection and the subscript s denotes the stoichiometric value. It then follows that

$$\frac{1}{m}\frac{dm}{dt} = \frac{-(RT/V)(dV/dt) - (\partial u/\partial p)(dp/dt) + (1/m)(dQ/dt) - CB}{u - h_f + D(\partial u/\partial \phi) - C[1 + (D/R)(\partial R/\partial \phi)]} \tag{10.12}$$

where

$$B = \frac{1}{p}\frac{dp}{dt} - \frac{1}{R}\frac{\partial R}{\partial p}\frac{dp}{dt} + \frac{1}{V}\frac{dV}{dt}$$

$$C = \frac{T(\partial u/\partial T)}{1 + (T/R)(\partial R/\partial T)}$$

$$D = \frac{[1 + (F/A)_0]m}{(F/A)_s m_0}$$

Equation (10.12) can be solved numerically for $m(t)$ given m_0, ϕ_0, $p(t)$, and appropriate models for the working fluid properties (see Sec. 4.7) and for the heat-transfer term dQ/dt (see Chap. 12).

Figure 10.10 shows cylinder pressure data for an open chamber DI diesel and fuel mass burning rate $dm/d\theta$ calculated from that data using the above method. The heat-transfer model of Annand was used (see Sec. 12.4.3). The result obtained is an *apparent* fuel mass burning rate. It is best interpreted, after multiplying by the heating value of the fuel, as the fuel chemical-energy or heat-release rate. The *actual* fuel-burning rate is unknown because not all the fuel "burns" with sufficient air available locally to produce products of *complete* combustion. About 60% of the fuel has burned in the first one-third of the total combustion period. The integral of the fuel mass burning rate over the combustion process should equal the total fuel mass burned; in this case it is 3% less than the total fuel mass injected. Note that chemical energy continues to be released well into the expansion process. The accuracy of this type of calculation then decreases, however, since errors in estimating heat transfer significantly affect the apparent fuel-burning rate.

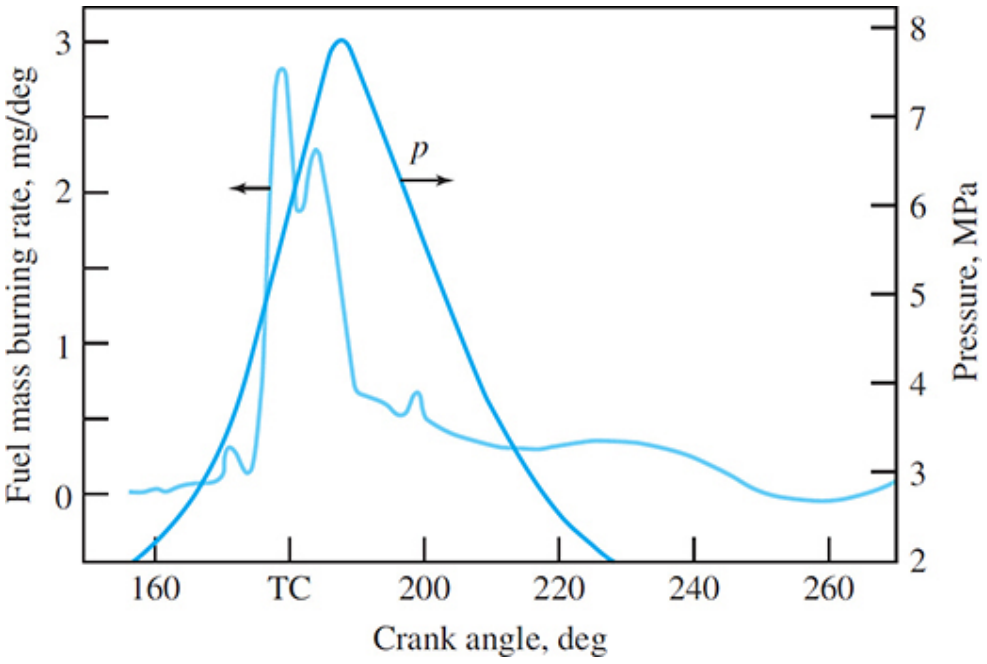


Figure 10.10 Cylinder pressure p and fuel mass burning rate calculated from p, as a function of crank angle, using the Krieger and Borman method[9] for a DI diesel engine at 3200 rev/min and full load.

Krieger and Borman also carried out sensitivity analyses for the critical assumptions and variables. They found that the effect of dissociation of the product gases was negligible. This permits a substantial simplification of Eq. (10.12) . With no dissociation, $u = u(T, \phi)$, and $R = \tilde{R}/M$ can be taken as constant, since the molecular weight M changes little. Then,

$$\frac{dm}{dt} = \frac{[1+(c_v/R)]p(dV/dt)+(c_v/R)V(dp/dt)-(dQ/dt)}{h_f+(c_v/R)(pV/m)-u-D(\partial u/\partial\phi)} \tag{10.13}$$

where D, as before, is $[1 + (F/A)_0]\, m/[(F/A)\, s\, m_0]$. Given the uncertainties inherent in the heat-transfer model and the neglect of nonuniformities and crevices, Eq. (10.13) represents an adequate level of sophistication.

The other sensitivity variations studied by Krieger and Borman were: shifting of the phasing of the pressure data 2° forward and 2° backward; translating the pressure data ±34 kPa; changing the heat transfer ±50%; and increasing the initial mass 5%. The initial mass change had a negligible effect on the fuel burning rate calculations. The heat-transfer changes of

±50% changed the mass of fuel burned by about ±5%. The change in phasing of the pressure data was more significant. It should be stressed that *accurate* (in magnitude and phasing) pressure data are an important requirement for useful heat-release or fuel-mass-burning rate analysis.

Heat Release Rate: Indirect-Injection Engines

In IDI diesel engines, the pressures in the two chambers, main and auxiliary, are not the same during the combustion process. Since combustion starts in the auxiliary or prechamber, the fuel energy release in the prechamber causes the pressure there to rise above the main chamber pressure. Depending on combustion chamber design and operating conditions, the prechamber pressure rises to be 0.5 to 5 bar above that in the main chamber. This pressure difference causes a flow of fuel, air, and burning and burned gases into the main chamber, where additional energy release now occurs. The analysis of the DI diesel was based on uniform pressure throughout the combustion chamber. For IDI engines the effect of the pressure difference between the chambers must be included. The methodology for carrying out a heat release rate analysis for IDI diesel engines can be found in the first edition of this text, [1] and the references quoted. It will not be discussed here since IDI diesels are no longer a widely used technology.

10.3.4 Conceptual Model of DI Diesel Combustion

Over the past 20 or so years, much information has been gained through use of optical laser diagnostics to measure the distributions of various fuel and chemical species that play key roles in the diesel combustion and pollutant formation processes, and of gas temperatures. When coupled with detailed flow and combustion chemistry modeling, this information has enabled a more complete and integrated description of the DI diesel combustion process to be developed. Based on summaries by Dec, [10] and Flynn et al. at Cummins Inc. and U.S. National Laboratories, [11] we will now review that more detailed conceptual model.

Figure 10.11 shows a sequence of illustrative drawings of a developing fuel spray and flame in a multi-spray DI diesel engine without swirl. The crank angles noted on each schematic are after the start of injection (ASI). The needle lift profile for the injector which defines the start and end of fuel

injection, and apparent heat-release-rate diagram for these drawings, are plotted in Fig. 10.12 so they can be keyed to the fuel injection process and fuel chemical energy release. Figure 10.13 shows more details of the spray and flame structure in the "quasi-steady" mixing-controlled phase of combustion. The different shading intensities indicate the regions of the fuel spray where the several different processes involved in this type of diesel combustion occur.

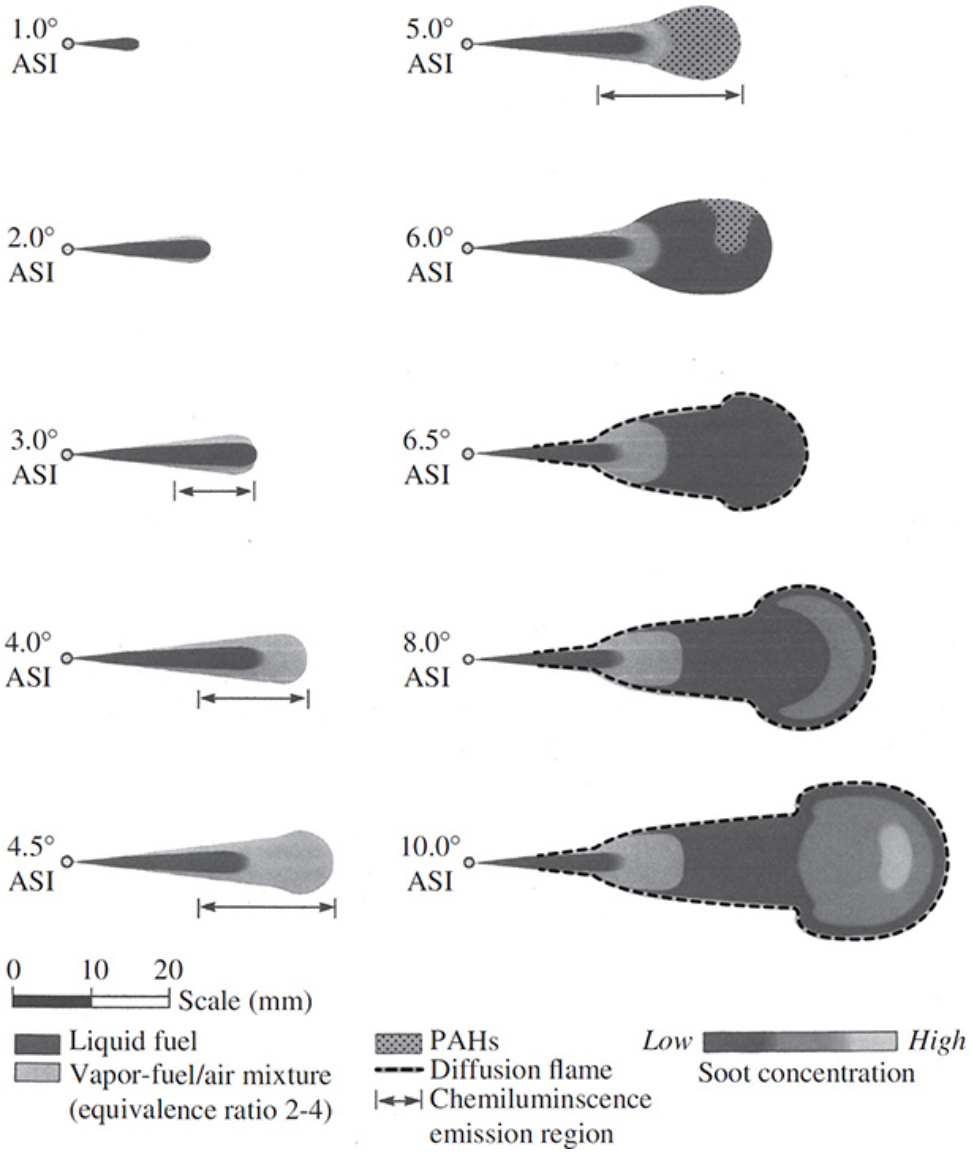


Figure 10.11 Schematic temporal sequence showing how DI diesel combustion evolves from start of liquid fuel injection through the quasi-steady mixing-controlled burning stage. The location in the spray and surrounding flame of the different processes is indicated. The timing (crank-

angle degrees after start of injection, ASI, 1° = 139 μs) and spray penetration (mm) correspond to combustion in a medium-sized diesel engine, at a typical operating condition. The injection duration extends beyond the 10° sequence shown. [10]

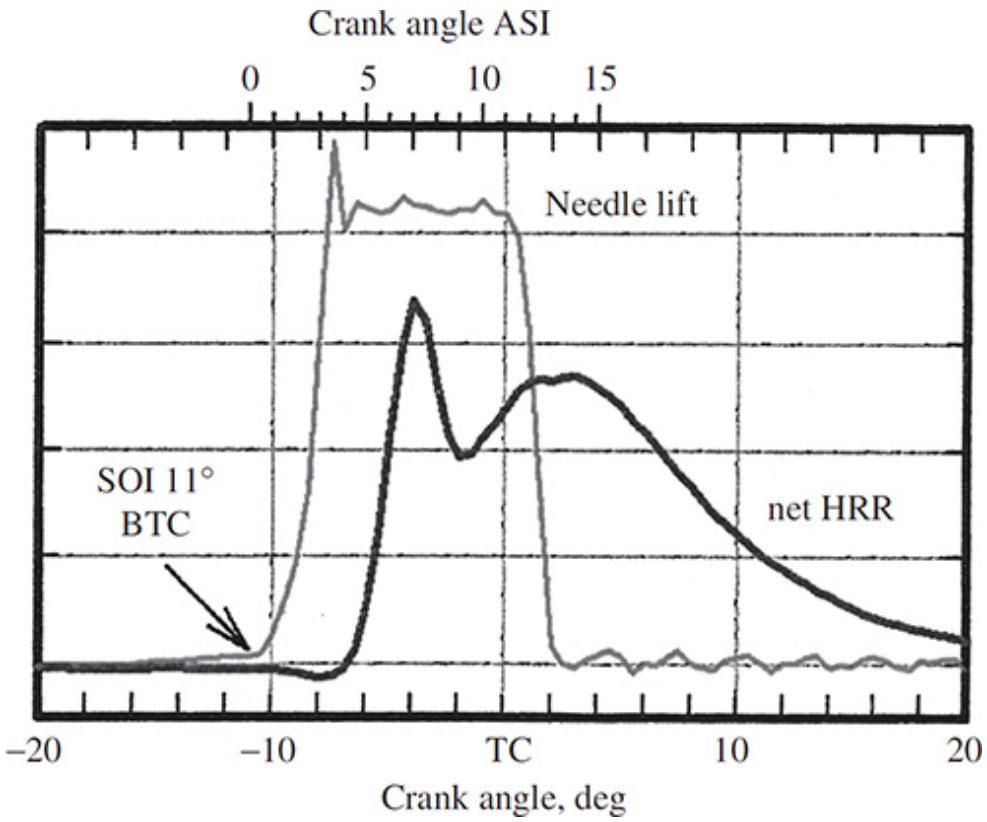


Figure 10.12 Fuel injector needle lift profile which defines the beginning and end of injection, and the net heat-release rate, versus crank angle for the engine conditions of the schematics shown in Fig. 10.11.[11]

As the liquid fuel exits the nozzle, liquid ligaments and larger drops rapidly atomize to fine drops (~10 μ diameter) whose momentum carries them into the combustion chamber as a "jet." This early spray region contains the fuel as drops which heat up and vaporize as their motion entrains hot air into the spray. A fuel-vapor air mixture forms within this droplet-containing region and further expands the spray as the spray continues to penetrate (and slow down), up to 4.5° after start of injection (ASI). The ignition chemistry then spontaneously commences in the leading region of the spray (where the fuel has been within the combustion chamber longest) where mixing of air

entrained into the spray with evaporated fuel has brought the mixture fuel-air equivalence ratio from very rich values to about 2 to 4 (still well-rich of stoichiometric). The diesel fuel molecules then start to break into fragments, which combine to form polycyclic aromatic hydrocarbons (PAHs), which in turn grow to form increasing amounts of soot particles of increasing size.

At about the crank angle of the peak heat-release-rate of the premixed combustion phase (some 6 to 6.5° ASI in this example), the mixing-controlled diffusion flame forms and rapidly surrounds the jet. The "fuel" for this diffusion flame is the fuel-rich mixture of partly reacted diesel-fuel vapor, entrained air, fuel-fragments, PAHs and soot, all inside the diffusion flame. The oxygen for this diffusion flame to complete combustion comes from the air that diffuses into the flame from the outside. Thus, Fig. 10.13 summarizes this description of the diesel combustion process during its quasi-steady phase while fuel injection continues and all these processes are occurring in sequence as indicated (after about 8 to 10° ASI in Fig. 10.11).

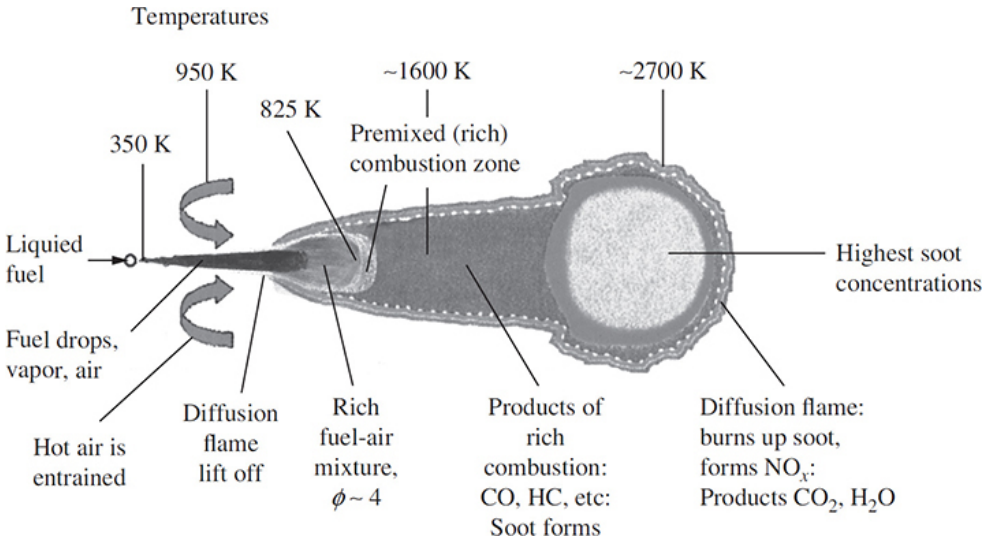


Figure 10.13 Key features, and gas temperature distribution, in a DI diesel fuel spray: injection, atomization, air entrainment, premixed-combustion process, soot formation region, diffusion-flame location, and lift-off distance from nozzle orifice. (*Developed from Ref. 11* .)

Figure 10.13 also shows the temperature distribution within the spray in this quasi-steady stage. The liquid fuel jet entering the chamber atomizes, entrains hot air (plus a modest amount of residual gas) at around 950 K, and

vaporizes, producing cooler rich fuel-air mixture ($\phi \sim 4$) at about 650 K. This mixture enters the spray region within the diffusion flame sheath, entraining recirculated partially oxidized rich combustion products, and raising its temperature to about 825 K. Then, low-temperature oxidation reactions occur releasing enough chemical energy to raise the temperature to about 1150 K. At which point oxidation becomes more rapid, this rich mixture passes through a fuel-rich premixed flame consuming the locally available oxygen, releasing 10 to 15% of the fuel's chemical energy, and attaining temperatures of order 1600 K. As the fuel passes through this rich premixed reaction zone or flame, hydrocarbon fragments form which become the building blocks for soot particles within the evolving rich fuel spray. These build up into particle nuclei which become heated by radiation from the flame, grow into soot particles and agglomerate. The highest levels of soot and largest particles are thus found in the leading region of the spray. These soot-particle-containing gases move into the diffusion flame reaction region around the spray where any remaining fuel vapor, fuel fragments, PAHs, and soot particles become oxidized to CO_2 and H_2O using oxygen that diffuses into this flame sheath from the air outside. The highest temperatures are reached in this diffusion flame where close-to-stoichiometric combustion occurs. It is here, within this high-temperature diffusion flame sheath (where there is a slight excess of oxygen) that the air pollutant nitric oxide (NO) largely forms: see Sec. 11.2.4 . These products of complete combustion then (in a net sense) diffuse outward from the thin diffusion flame reaction zone mixing with excess air, become leaner, with their temperature and degree of dissociation of CO_2 and H_2O decreasing and releasing chemical energy in the process. [11]

Key features of this more complete phenomenological diesel spray combustion model (in a warmed-up engine) are the limited penetration of fuel droplets into the combustion chamber, the two stage nature of the combustion process—a fuel-rich premixed flame within the spray followed by a diffusion flame surrounding the spray where the rich premixed combustion products release most of their chemical energy as they fully oxidize. These flame regions are *lifted off* (separated from) the injector nozzle hole allowing the spray to entrain hot air into the spray core to create the rich mixture within the spray. The substantial amount of additional oxygen required for complete combustion diffuses into the spray-enveloping diffusion flame sheath from the outside.

Two important practical areas to be incorporated into the above conceptual model of diesel spray combustion are air swirl, and multiple fuel injections. Increasing levels of air swirl are used to speed up the fuel-air mixing process as engine size—stroke and bore—are reduced and maximum engine speed rises. This swirling air motion, orthogonal to each initial fuel spray axis, forces additional air into the spray; this decreases the spray's fuel/air equivalence ratio more quickly, and thus achieves a less-rich mixture within the spray for the first "premixed" stage of the autoignition process to occur sooner. The higher the swirl level (Sec. 8.3), the more these effects are augmented. Swirl impacts on diesel combustion are explored more fully in Sec. 10.6 .

Fuel-injection technology has progressed to the point where multiple injection pulses are feasible. Pilot injection—injection of a modest fraction of the total amount, injected just prior to the main injection—has long been used to reduce combustion noise and NO_x emissions. Pilot injection shortens the ignition delay by creating a hotter local environment into which the main fuel spray is injected. Post-main-injection pulses are now used to reduce particulate emissions, raise the exhaust gas temperature to initiate the burn-up of particulate matter collected in the exhaust system particulate trap, or provide hydrocarbons for the diesel oxidation catalyst to combust to increase the exhaust gas temperature, or provide reducing species in the exhaust. Obviously, each of the several injections per cycle occurs in the environment created by the previous injections. The topic of multiple-injection effects on diesel combustion is explored more fully in Sec. 10.7.1 .

10.4 FUEL SPRAY BEHAVIOR

10.4.1 Fuel Injection

The fuel is injected into the cylinder of a diesel engine through a nozzle with a large pressure difference across the nozzle orifice. The cylinder pressure at injection is typically in the range of 50 to 100 bar (atm). Fuel injection pressures in the range 1000 to over 2000 bar (atm) are used depending on the engine size and type of combustion system employed. These large pressure differences across the injector nozzle are required so that the injected liquid

fuel jet will enter the chamber at sufficiently high velocity to (1) atomize into small-sized droplets to enable rapid evaporation, and (2) traverse the combustion chamber in the time available so as to fully utilize the air charge.

Examples of common diesel fuel-injection systems were described briefly in Sec. 1.7 and illustrated in Figs. 1.26 and 1.27. The task of the fuel-injection system is to meter the appropriate quantity of fuel for the given engine speed and load to each cylinder, each cycle, and inject that fuel at the appropriate time in the cycle at the desired rate with the spray configuration required for the particular combustion chamber employed. It is important that injections begin and end cleanly, and avoid any secondary injections. To accomplish this task, fuel is drawn from the fuel tank by a supply pump, and forced through a filter to the injection pump. The injection pump sends fuel under pressure to the nozzle pipes which carry fuel to the injector nozzles located in each cylinder head. Excess fuel goes back to the fuel tank. Individual components can be combined to form different diesel fuel injection systems. In some of these (with axial-and radial-piston distributer pumps, and at times in the common-rail system), the fuel supply pump function is integrated into the high-pressure pump. Figure 10.14 shows the layout of a common-rail fuel injection for a four-cylinder passenger car diesel engine with the set of engine sensors it requires.[12]

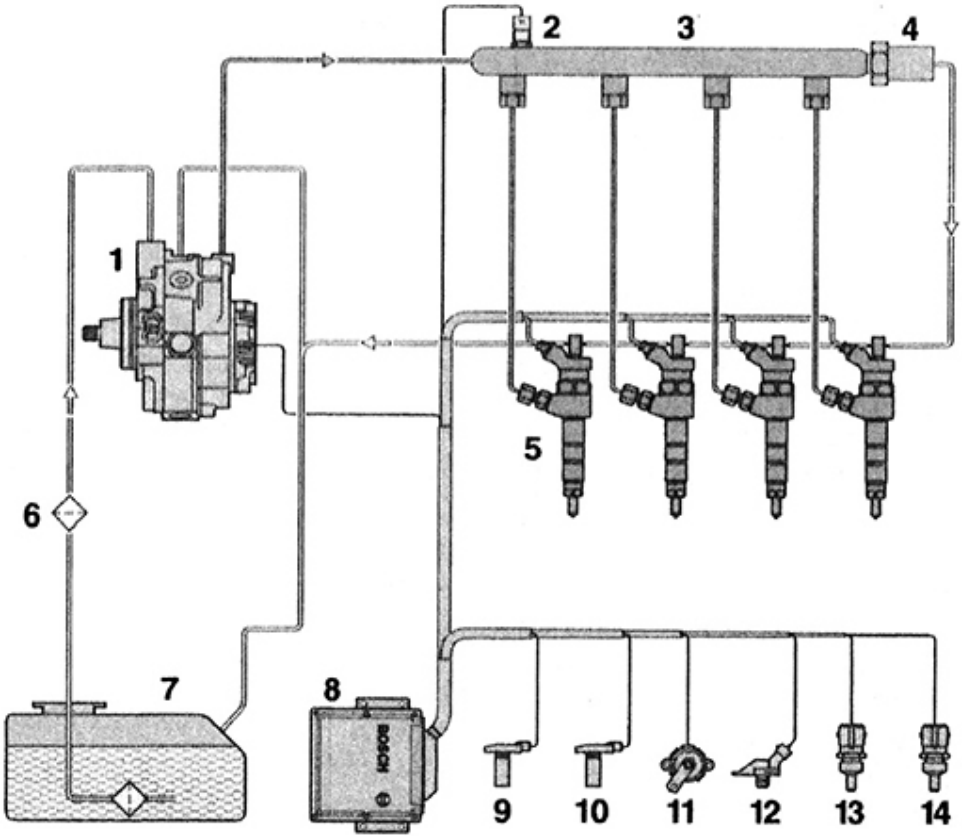


Figure 10.14 Layout of a common-rail fuel-injection system for a passenger car diesel engine with delivery-controlled high-pressure pump: injection system components and sensors for electronic control unit (ECU):[12] (1) high-pressure pump; (2) rail-pressure sensor; (3) "common rail" fuel rail; (4) pressure limiter; (5) injectors; (6) fuel filter; (7) fuel tank with first stage filter; (8) ECU; (9) engine-speed sensor; (10) phase sensor; (11) pedal-travel sensor; (12) boost-pressure sensor; (13) air-temperature sensor; (14) engine-temperature sensor. (*Courtesy Robert Bosch GmbH and SAE.*)

In-line injection pumps are used in engines in the 40- to 100-kW per cylinder maximum power range. They contain a plunger and barrel assembly for each engine cylinder. The fuel metering and injection process is illustrated in Fig. 10.15. An engine-driven cam moves the plunger in the delivery direction, and a spring forces it back to its initial position. The plunger stroke is fixed. The plunger fits sufficiently accurately within the barrel to seal without additional sealing elements, even at high pressures and low speeds. The amount of fuel delivered is altered by varying the *effective* plunger stroke. This is achieved by means of a control rod or rack (see Fig.

10.16) which moves in the pump housing and rotates the plunger with its helical groove via a ring gear or linkage lever on the control sleeve. The high-pressure chamber above the plunger is connected with the chamber below the plunger by a vertical groove in the plunger. Delivery ceases when the plunger helix exposes the intake port (port opening), thus connecting the plunger chamber with the suction gallery. When this takes place depends on the rotational position of the plunger. In the case of a lower helix, delivery always starts (port closing) at the same time, but ends sooner or later depending on the rotational position of the plunger. With a plunger with an upper helix, port closing (start of delivery) not port opening is controlled by the helix and is varied by rotating the plunger. Figure 10.16 illustrates how the plunger helix controls fuel delivery.[12]

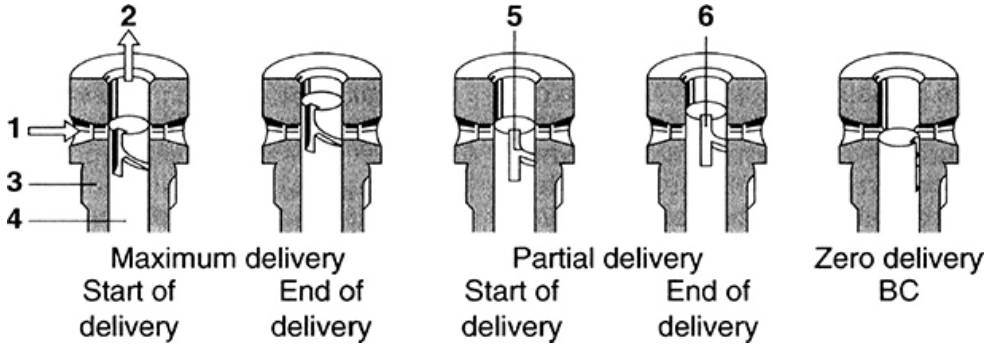


Figure 10.15 Barrel and plunger with helical groove used to control fuel delivery in the in-line diesel fuel injection pump:[12] (1) from fuel gallery; (2) to nozzle; (3) barrel; (4) plunger; (5) lower helix; (6) vertical (stop) groove. (*Courtesy Robert Bosch GmbH and SAE.*)

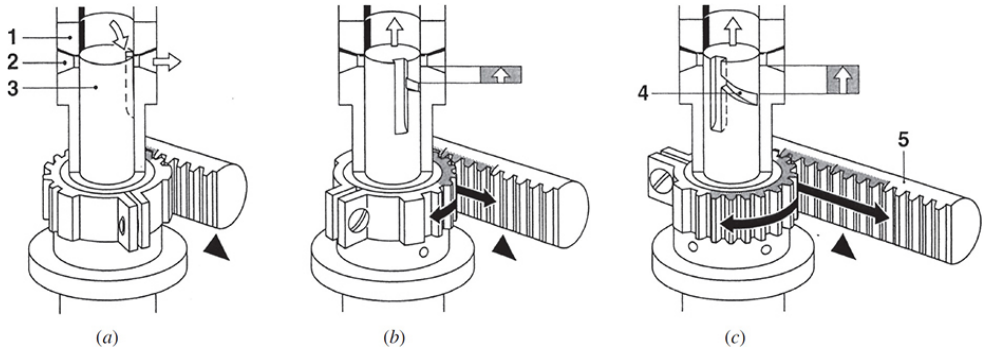


Figure 10.16 Toothed control rack used to rotate plunger (with its helical groove; see Fig. 10.15) to control fuel delivery as a function of engine

load:[12] (*a*) zero delivery; (*b*) partial delivery; (*c*) maximum delivery. (1) Pump barrel; (2) inlet port; (3) pump plunger; (4) helix; (5) control rack. (*Courtesy Robert Bosch GmbH and SAE.*)

Distributor-type fuel-injection pumps are normally used in multi-cylinder engines up to about 50 kW per cylinder maximum power with injection pressures up to 2000 bar (atm), at speeds to about 2500 rev/min. These pumps have only one plunger and barrel. The pump plunger is made to describe a combined rotary and stroke movement by the rotating cam plate. The fuel is accurately metered to each injection nozzle in turn by this plunger which simultaneously acts as the distributor. Such units are more compact and cheaper than in-line pumps but cannot achieve such high injection pressures. The distributor-type fuel-injection pump is combined with the automatic timing device, governor, and supply pump to form a single unit.

Also used extensively on both smaller and larger engines are unit injectors where the pump and injector nozzle are combined into a single unit. An example of a unit injector and its driving mechanism is shown in Fig. 10.17. Fuel, supplied to the injector through a fuel-distributing manifold, enters the cavity or plunger chamber ahead of the plunger through a metering orifice. When fuel is to be injected, the cam via the rocker arm pushes down the plunger, closing the metering orifice and compressing the fuel, causing it to flow through check valves and discharge into the cylinder through the injector nozzle holes.

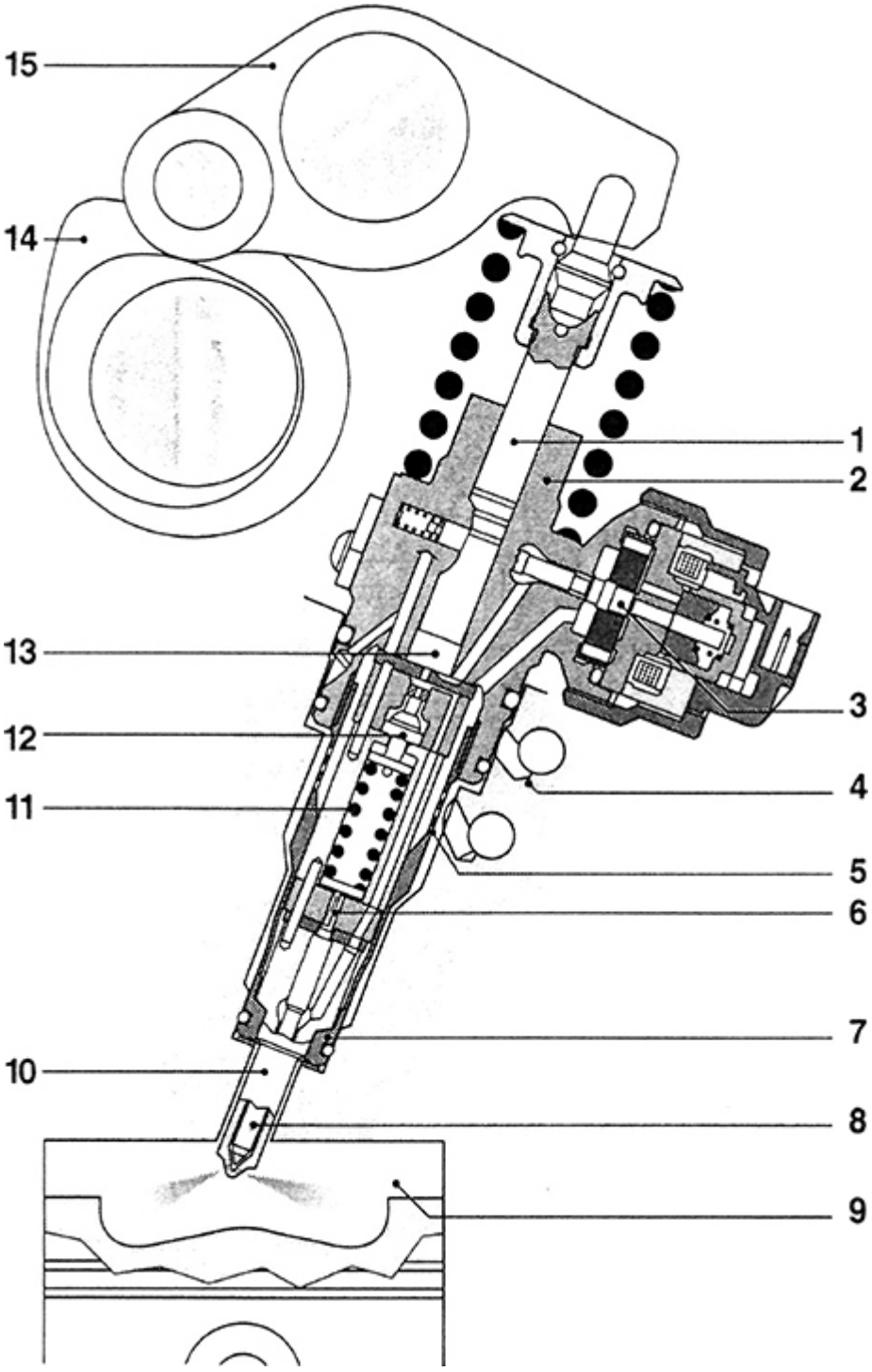


Figure 10.17 Example of a unit injector and its driving mechanism used in passenger car diesel engines.[12] (1) Pump plunger; (2) pump body; (3) solenoid-valve needle; (4) fuel return (low-pressure stage); (5) inlet passage; (6) hydraulic stop (damping unit); (7) retaining nut; (8) nozzle needle; (9) engine combustion chamber; (10) integrated nozzle; (11) nozzle

compression spring; (12) accumulator bypass plunger; (13) plunger chamber; (14) camshaft; (15) roller rocker arm. (*Courtesy Robert Bosch GmbH and SAE*.)

The most important part of the injection system is the nozzle. Examples of different nozzle types and a nozzle holder assembly are shown in Fig. 10.18. The nozzles shown are fluid-controlled needle valves where the needle is forced against the valve seat by a spring. The pressure of the fuel in the pressure chamber above the nozzle aperture opens the nozzle by the axial force it exerts on the conical surface of the nozzle needle. Needle valves are used to prevent dribble from the nozzles when injection is not occurring. It is important to keep the volume of fuel left between the needle and nozzle orifices (the sac volume) as small as possible to prevent any fuel flowing into the cylinder after injection is over, to control hydrocarbon emissions (see Sec. 11.4.4). Multi-hole nozzles are used with most DI diesel engine systems. Pintle nozzles, where the needle projects into and through the nozzle hole, are used in IDI engine systems. The shape of the pin on the end of the nozzle needle controls the spray pattern and fuel-delivery characteristics. Auxiliary nozzle holes are sometimes used to produce an auxiliary smaller spray to aid ignition and starting.

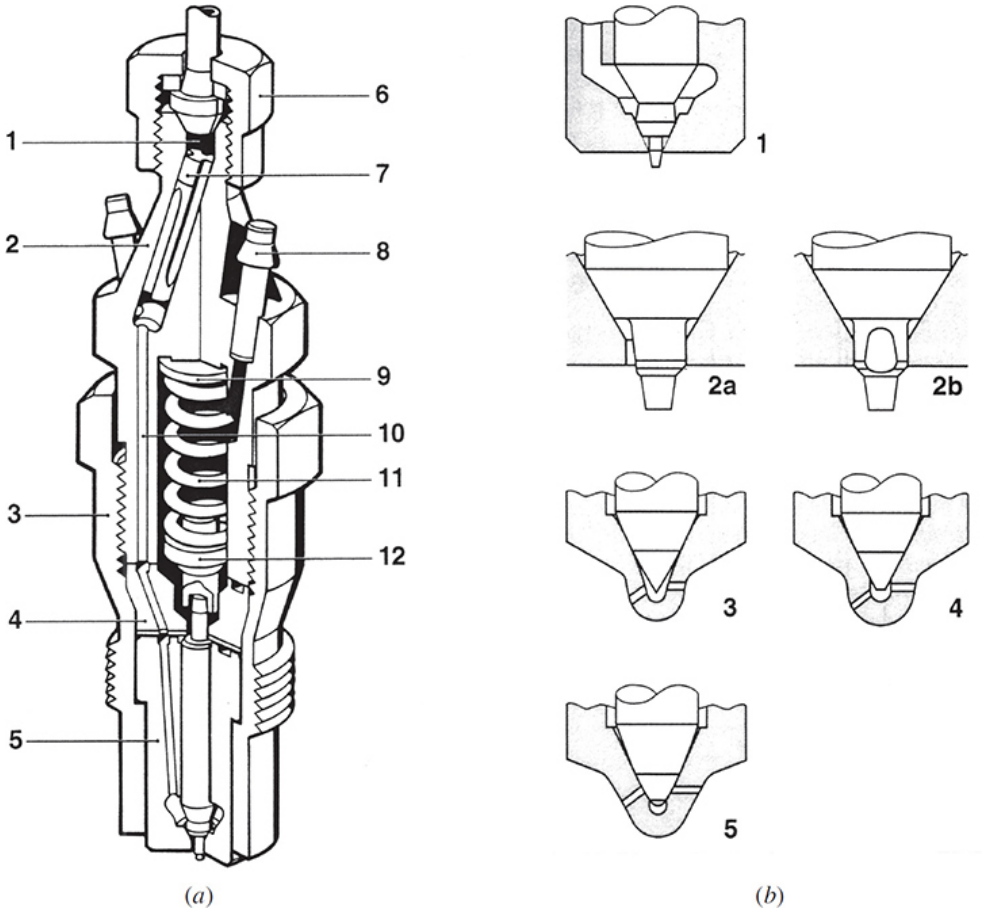


Figure 10.18 (*a*) Layout of injector nozzle and nozzle holder assembly with throttling pintle: (1) inlet; (2) nozzle-holder body; (3) nozzle-retaining nut; (4) shim; (5) nozzle; (6) union nut with high-pressure line; (7) edge filter; (8) leak-fuel port; (9) pressure-adjusting shims; (10) pressure passage; (11) compression spring; (12) pressure pin. (*b*) Examples of different nozzle shapes and nozzle hole arrangements: (1) throttling-pintle nozzle; (2) throttling-pintle nozzle with flat-cut pintle (2 *a* side view, 2 *b* front view); (3) hole-type nozzle with conical blind hole; (4) hole-type nozzle with cylindrical blind hole; (5) sac-less nozzle. [12] (*Courtesy Robert Bosch GmbH and SAE*.)

Electronic control of injection is now widely used. In an electronic injector, such as that shown in Fig. 10.19 *a*, a solenoid-operated control valve performs the injection timing and metering functions in a fashion analogous to the ports and helices of the mechanical injector. Solenoid valve closure initiates pressurization and injection, and opening causes injection

pressure decay and end of injection. Duration of valve closure determines the quantity of fuel injected. The unit shown uses camshaft/rocker arm driven plungers to generate the injection pressure, and employs needle-valve nozzles of conventional design. Increased flexibility in fuel metering and timing and simpler injector mechanical design are important advantages. [1]

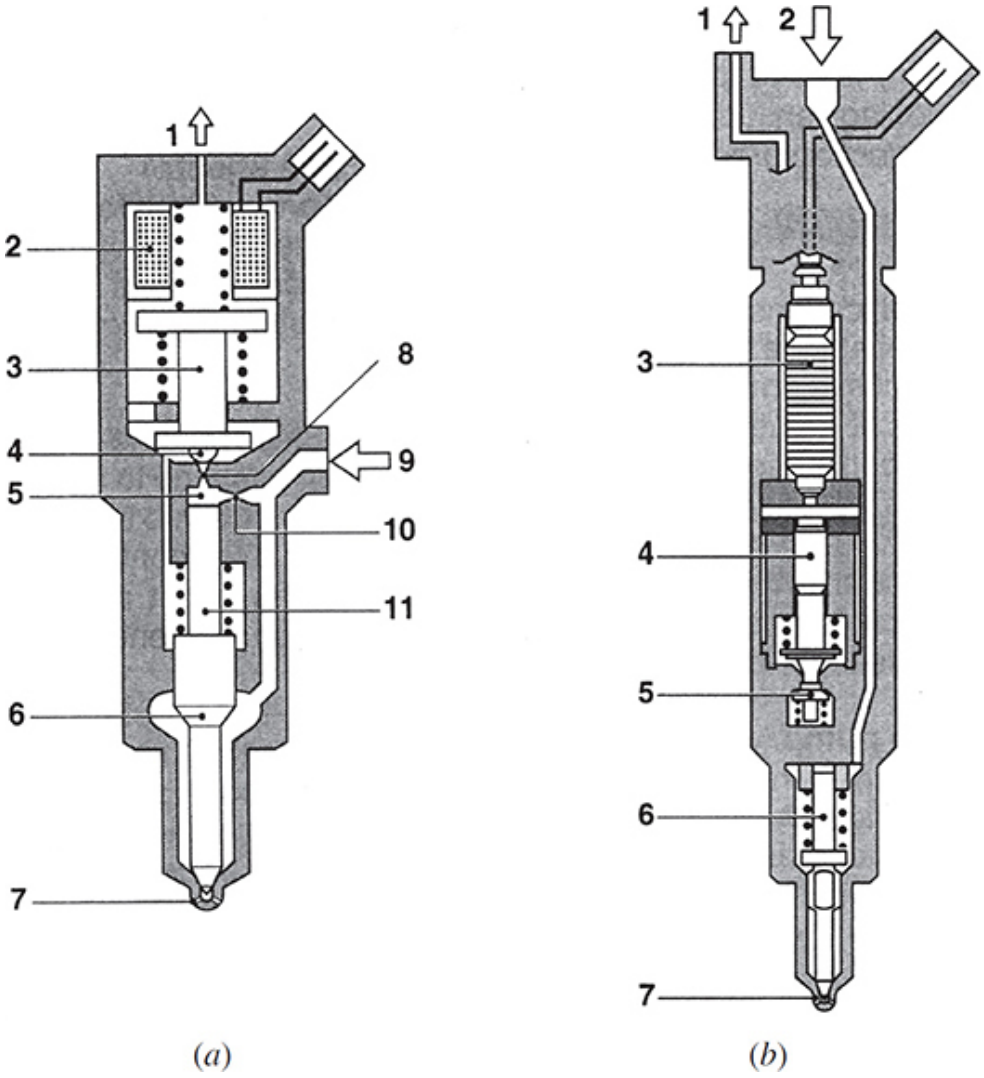


Figure 10.19 Examples of electronically controlled injectors. (*a*) Solenoid-valve injector: (1) fuel return; (2) solenoid coil; (3) solenoid armature; (4) valve ball; (5) valve control chamber; (6) nozzle-needle pressure shoulder; (7) injection orifice; (8) outlet restrictor; (9) high-pressure port; (10) inlet restrictor; (11) valve plunger. (*b*) Piezo injector: (1) fuel return; (2) high-pressure port; (3) piezo-actuator module; (4) hydraulic

booster; (5) valve; (6) nozzle needle; (7) injection orifice. [12] (*Courtesy Robert Bosch GmbH and SAE*.)

A newer option is piezoelectric injectors (Fig. 10.19*b*), where a piezo-actuator drives the nozzle needle rather than solenoid-valve controlled fuel pressure. Here, an electrical pulse through multiple thin quartz crystals creates the force (pressure), which through a hydraulic booster moves the needle. This reduces needle size and weight, and friction within the injector, providing these advantages: more compact and lighter injectors; multiple injections; more precise control of each injection with shorter time intervals between injections. Injectors with pressures of close to 200 MPa produce better fuel atomization, and can provide up to five injections per cylinder per cycle. Use of piezo-injectors in diesel engines reduces engine noise and air-pollutant emissions significantly, and increases power output and efficiency.

Accurate predictions of fuel behavior within the injection system require sophisticated hydraulic models. Approximate estimates of the injection rate through injector nozzles can be made as follows. If the pressure upstream of the injector nozzle can be estimated or measured, and assuming the flow through each nozzle is quasi steady, incompressible, and one dimensional, the mass flow rate of fuel injected through the nozzle $\dot{m}_f$ is given by

$$\dot{m}_f = C_D A_n \sqrt{2\rho_f \Delta p} \tag{10.14}$$

where A_n is the nozzle minimum area, C_D the discharge coefficient, ρ_f the fuel density, and Δp the pressure drop across the nozzle. If the pressure drop across the nozzle and the nozzle open area are essentially constant during the injection period, the mass of fuel injected is then

$$m_f = C_D A_n \sqrt{2\rho_f \Delta p} \frac{\Delta\theta}{360N} \tag{10.15}$$

where $\Delta\theta$ is the nozzle open period in crank-angle degrees and N is engine speed. Equation (10.15) illustrates the dependence of injected amounts of fuel on key injection system and engine parameters.

10.4.2 Overall Spray Structure

The fuel is introduced into the combustion chamber of a DI diesel engine through one or usually several nozzles or orifices with a large pressure differential between the fuel supply line and the cylinder. Different designs of nozzle are used (e.g., single-orifice, multi-orifice, throttle, or pintle; see Fig. 1.27), depending on the needs and size of the combustion system employed. Diesel injectors usually operate with fuel-injection pressures between 1000 atm and 2500 atm (bar). [g] At time of injection, the air in the cylinder has a pressure of 50 to 100 atm (bar), temperature about 1000 K, and density between kg/m^3 15 and 35 kg/m^3. Nozzle hole diameters span the range of 0.1 to 0.3 mm diameter (automotive applications), with length/diameter ratios from 2 to 8. Typical distillate diesel fuel properties are: relative specific gravity of 0.8, viscosity between 3 kg/m · s and 10 kg/m · s and surface tension about 3×10^{-2} N/m (at 300 K). Figure 10.20 illustrates the basic features of a typical DI engine fuel spray. As the liquid jet leaves the nozzle it atomizes into fine drops whose momentum creates a spray, and spreads out as the spray entrains the surrounding air. The initial jet velocity is greater than 100 m/s. The outer surface of the liquid jet breaks up into drops of order 10-μm diameter or less, close to the nozzle exit as the liquid column leaving the nozzle rapidly disintegrates. As injected fuel moves away from the nozzle, the mass of air entrained into the spray increases, the spray diverges, its width increases, and its velocity decreases. The fuel drops evaporate as this hot-air entrainment process proceeds. The tip of the spray penetrates further into the combustion chamber as injection continues, but at a decreasing rate. Figure 10.21 shows photographs of a diesel fuel spray injected into quiescent air in a rapid-compression machine which simulates diesel conditions.[13] Two different photographic techniques, back lighting and shadowgraph,[h] have been used to distinguish the liquid-containing core of the jet and the extent of the fuel vapor region of the spray which surrounds the liquid core. The region of the jet closest to the nozzle (until injection ceases at 3.3 ms) contains liquid drops and ligaments; the major region of the spray consists of a substantial fuel vapor plus air cloud around this narrow core which contains the remaining liquid fuel.

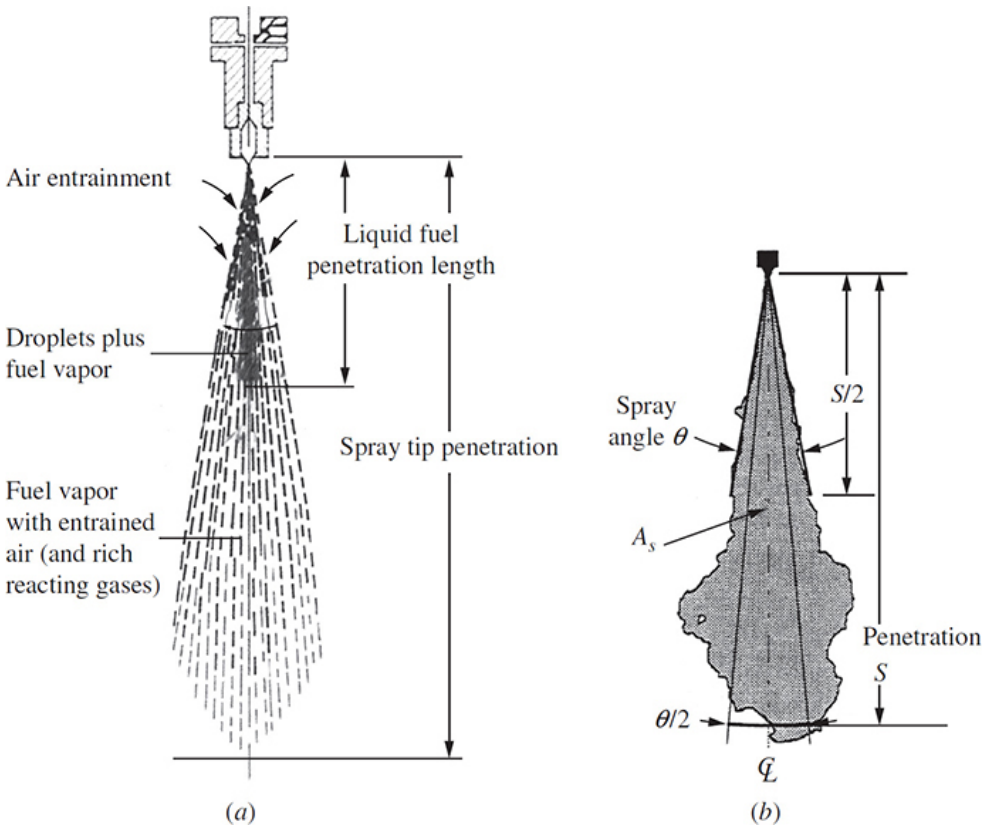


Figure 10.20 Schematic of diesel fuel spray, defining (*a*) its major features and (*b*) its key parameters: spray axis, C_L; penetration distance S; full core angle θ; projected spray area (grey) A_s.

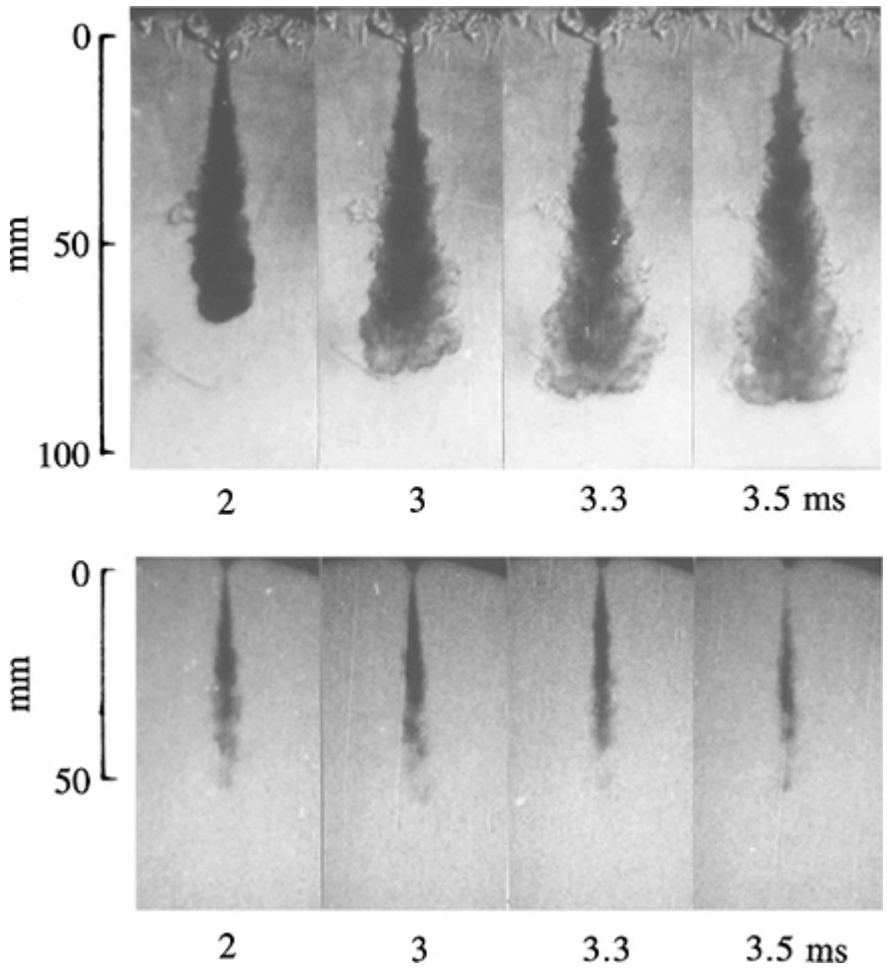


Figure 10.21 Shadowgraph and back-illuminated photographs of evaporating spray injected into nitrogen at 3.4 MPa and 670 K in rapid-compression machine. Times in milliseconds are after start of injection: injection duration 3.3 ms. Top (shadowgraph) photographs show full fuel vapor and liquid region. Bottom (back-illuminated) photographs show only the liquid-containing core. [13]

So far we have discussed a vaporizing fuel spray prior to the onset of combustion. Within this fuel-rich, fuel vapor plus entrained hot air spray, the autoignition chemistry occurs which raises the mixture temperature within the spray in the "premixed rapid-burning" first phase of the combustion process. This then sets up the "mixing-controlled" second phase of the diesel burning process where a diffusion flame (lifted off the injector nozzle exit) surrounds the spray. These spray combustion phases are discussed in more detail in Secs. 10.5 and 10.6.

Different spray configurations are used in the different diesel combustion systems described earlier in this chapter. The simplest configuration involves multiple sprays injected into quiescent air in the largest-size diesels (Fig. 10. 1 *a*). Figures 10.20 and 10.21 illustrate the essential features of each spray under these circumstances until interactions with the wall occur. Each liquid fuel jet atomizes into drops and ligaments as fuel exits the nozzle orifice (or shortly thereafter). The spray immediately starts entraining air, spreads out and slows down as the mass flow in the spray increases. The droplets on the edges of the spray evaporate first, creating a fuel vapor-air mixture sheath around the liquid-containing core. The highest velocities are on the jet axis. The equivalence ratio is highest on the centerline and fuel-rich throughout, and drops to zero (unmixed air) at the spray boundary. Once the sprays have penetrated to the outer regions of the combustion chamber, they interact with the chamber walls. The spray is then forced to split and flow tangentially along the wall. Eventually the sprays from multi-hole nozzles interact with one another. Figure 10.22 shows a diesel fuel spray interacting with the cylindrical outer wall of a disc-shaped combustion chamber in a rapid-compression machine under typical diesel-injection conditions. The cylinder wall causes the spray to split with about half flowing circumferentially in either direction. Adjacent sprays then interact forcing the flow radially inward toward the chamber axis. [14]

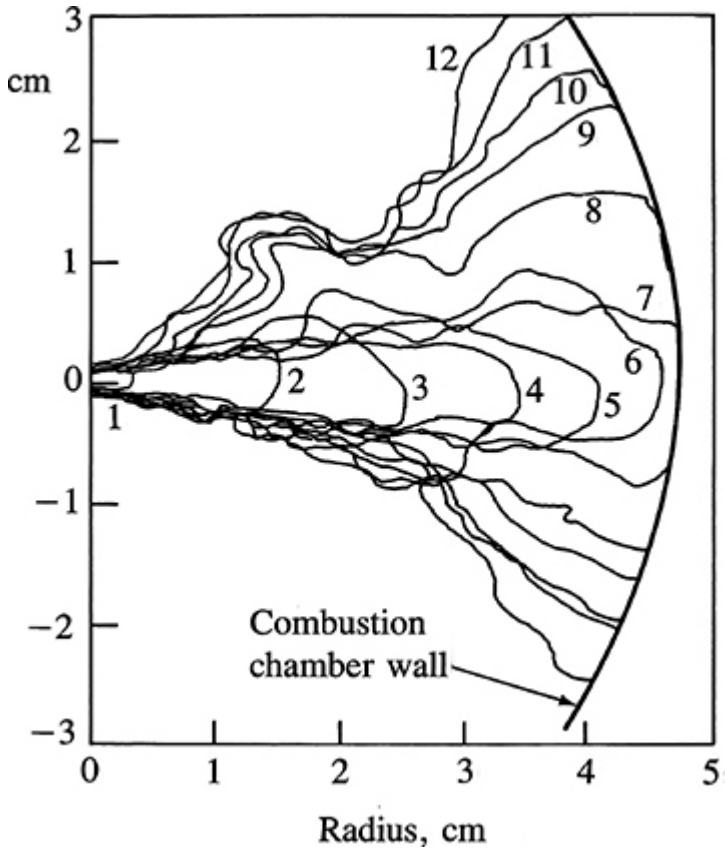


Figure 10.22 Outlines of the outer vapor boundary of a diesel fuel spray from 12 successive frames of a rapid-compression-machine high-speed shadowgraph movie showing interaction of vaporizing spray with the cylindrical wall of the combustion chamber. Injection pressure 60 MPa. Time between frames 0.14 ms. [14]

Most of the other combustion systems in Figs. 10.1 and 10.2 use air swirl to increase fuel-air mixing rates. A schematic of the spray pattern which results when a fuel jet is injected radially outward into a swirling flow is shown in Fig. 10.23. Because there is now relative motion in both radial and tangential directions between the initial jet and the air, the structure of the jet is more complex. As the spray entrains air and slows down it becomes increasingly bent toward the swirl direction; for the same injection conditions it will penetrate less with swirl than without swirl. An important feature of the spray is the large vapor-containing region downstream of the liquid-containing core. Figure 10.24 shows schlieren photographs of four fuel jets injected on the axis of an IDI diesel engine prechamber with high

clockwise swirl. The photograph on the left, with high sensitivity, shows the outer boundary of the fuel vapor region of the spray; the low-sensitivity photograph on the right locates the liquid phase regions of the spray. [15] The interaction between the swirl and both liquid and vapor spray regions is evident, and changes how the the spray interacts with the chamber wall.

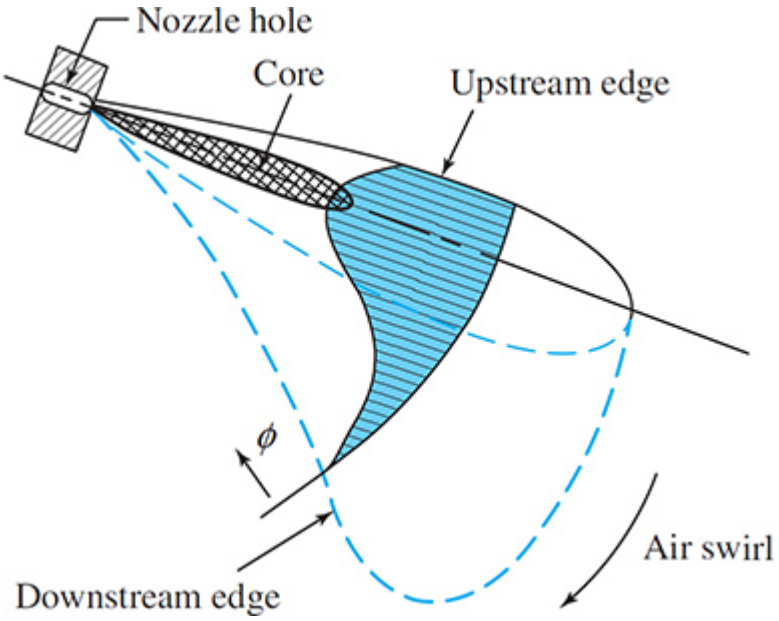


Figure 10.23 Schematic of fuel spray injected radially outward from the chamber axis into a swirling airflow. Distribution of fuel-air equivalence ratio (ϕ) within the spray is indicated.

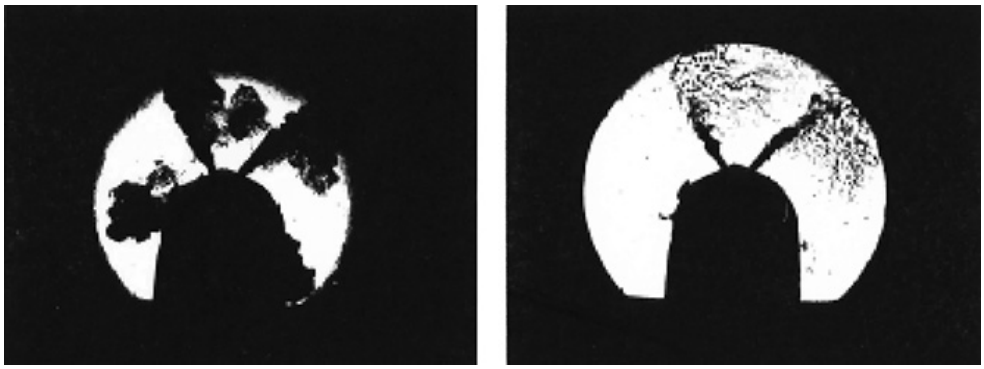


Figure 10.24 Schlieren photographs of vaporizing sprays injected into the swirling airflow in a transparent prechamber of special IDI diesel. Left: high-sensitivity image showing boundaries of the vapor regions of spray.

Right: low-sensitivity image showing liquid-containing core (dark) in relation to vapor regions (mottled). [15]

To couple the spray-development process with the ignition phase of the combustion, it is important to know which regions of the spray contain the fuel injected early in the injection process since it is these regions that are likely to autoignite first. Each spray develops as follows. At the start of injection liquid fuel enters the quiescent air charge, almost immediately atomizes into small drops that move outward from the nozzle, and slow down rapidly as air is entrained into the spray and accelerated. This start-up process forms a vortex or "puff" at the head of the spray. The injected fuel which follows encounters less resistance; thus drops from that fuel overtake the drops from first-injected fuel forcing them outward toward the periphery of the spray. At the tip of this developing spray, the drops with their surrounding fuel vapor and entrained air meet high aerodynamic resistance and slow down, but the spray continues to penetrate the air charge because this retarded mixture at the tip of the spray is continually replaced by new higher-momentum later-injected drops. [16] As Figs. 10.21 and 10.24 indicate, these drops evaporate quickly.

10.4.3 Atomization and Spray Development

Under diesel engine injection conditions, the liquid fuel jet exiting the nozzle breaks up rapidly and forms a cone-shaped spray. This type of behavior falls into the atomization breakup regime, and produces droplets with sizes much less than the nozzle exit diameter. This behavior is different from other modes of liquid jet breakup. At low jet velocity, in the Rayleigh regime, the liquid jet breakup is due to the unstable growth of surface waves caused by surface tension and results in drops larger than the jet diameter. As jet velocity is increased, forces due to the relative motion of the jet and the surrounding air augment the surface tension force, and lead to drop sizes of the order of the jet diameter. This is called the first wind-induced breakup regime. A further increase in jet velocity results in breakup characterized by divergence of the jet spray after an intact or undisturbed length downstream of the nozzle. In this second wind-induced breakup regime, the unstable growth of short-wavelength waves induced by the relative motion between

the liquid and surrounding air produces droplets whose average size is less than the jet diameter. Further increases in jet velocity lead to the atomization regime, where the breakup of the outer surface of the jet occurs at (or before) the nozzle exit plane and results in droplets whose average diameter is much smaller than the nozzle diameter. Aerodynamic interactions at the liquid/gas interface appear to be one major component of the atomization mechanism in this regime. [16–18]

A sequence of very short time exposure photographs of the emergence of a liquid jet from a nozzle of 0.34 mm diameter and $L_n/d_n = 4$, into high-pressure nitrogen at ambient temperature, is shown in Fig. 10.25. The figure shows how the spray tip penetrates and the spray spreads during the early part of its travel. [18] Data such as these have been used to examine the dependence of the spray development on gas and liquid density, liquid viscosity, and nozzle geometry. [18–22] The effects of the most significant variables, gas/liquid density ratio, and nozzle geometry on initial jet spreading angle are shown in Fig. 10.26. For a given geometry (cylindrical hole with sharp corners), the initial jet spreading or spray angle increases with increasing gas/liquid density ratio as shown in Fig. 10.26 *a* for non-vaporizing sprays, typical density ratios for diesel injection conditions are between 0.015 and 0.05. Of several different nozzle geometry parameters examined, the length/diameter ratio proved to be the most significant (see Fig. 10.26 *b*).

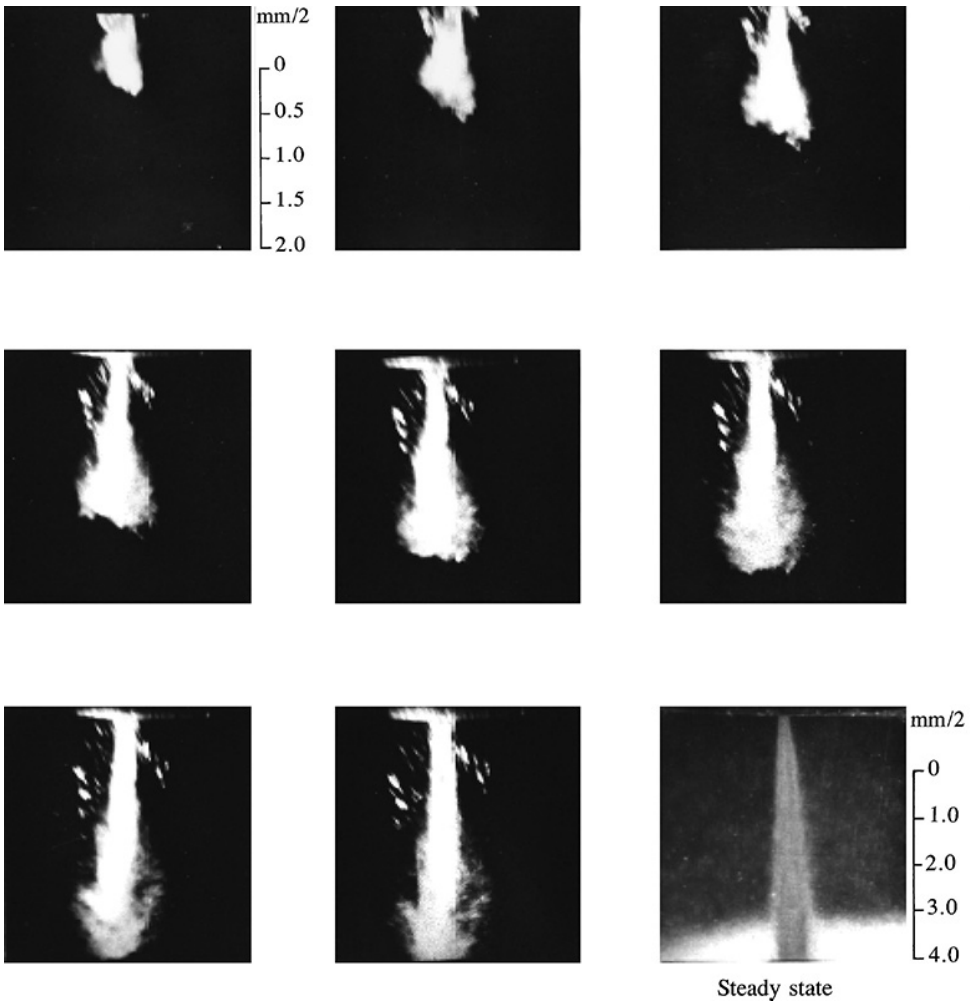


Figure 10.25 Photographs showing the emergence (and steady state image, bottom right) of a high-pressure liquid spray from a nozzle orifice. Time between frames 2.1 μs. Liquid: water; gas: nitrogen at 1380 kPa, Δp across nozzle = 11 MPa, nozzle diameter = 0.34 mm. [19]

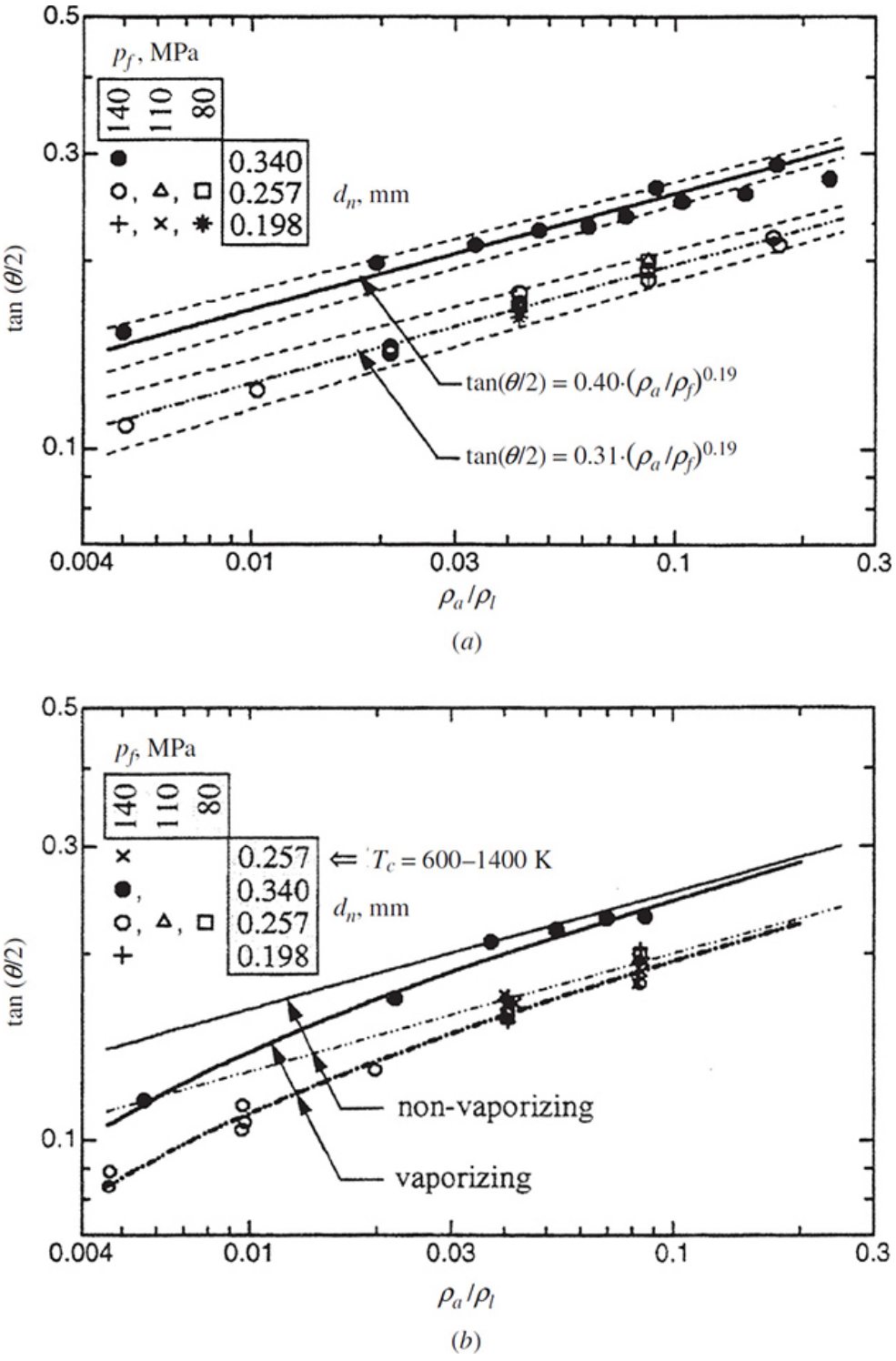


Figure 10.26 (*a*) Measured spray angles of non-vaporizing sprays versus density ratio ($\rho a / \rho l$ = gas density/liquid density) for several nozzle geometries and liquid injection pressures of 80 to 140 MPa. Nozzle diameters d_n = 0.2 to 0.34 mm. (*b*) Measured spray angles versus density

ratio for a range of nozzle geometries. dn = 0.2 to 0.34 mm diameter, same range of injection pressures. Correlating Eq. (10.16) also shown.[21, 23]

The roles of various parameters on the spreading angle of a diesel spray through their effect on the fuel atomization and turbulent air-entrainment processes are not yet well understood. Figure 10.26 compares non-vaporizing spray development (fuel injected into a cold ambient gas) with vaporizing (hot ambient gas—but without combustion), as a function of the gas to liquid density ratio. The spreading angles shown are the spreading angles of the quasi-steady portion of the fuel jet; i.e., its upstream portion. The jet spreading angle (defined by the tangent of $\theta/2$; see Fig. 10.20 *b*) has been correlated empirically by: [22]

$$\tan(\theta/2) = c_1\left[\left(\frac{\rho_a}{\rho_f}\right)^{0.19} - c_2\left(\frac{\rho_f}{\rho_a}\right)^{0.5}\right] \tag{10.16}$$

where ρ_a and ρ_f are ambient gas and liquid fuel densities, c_1 is a constant depending on the nozzle geometry (between 0.26 and 0.4), and c_2 is zero for non-vaporizing sprays and 0.0043 for vaporizing sprays. [21, 23] Equation (10.16) is the basis for the correlating lines in Fig. 10.26: Note that the behavior of *vaporizing* and *non-vaporizing* sprays is different (even at the same ρ_a/ρ_f ratio). Vaporization of the fuel droplets within the spray cools the entrained air through the fuel's heat of vaporization requirement. This cooling results in a higher density mixture in the jet relative to the non-vaporizing situation which, based on jet momentum conservation, results in slower penetration. A competing effect is that air entrainment is reduced by vaporization resulting in slower jet deceleration and reduced spreading angles. [22]

Jet breakup trends can be summarized as follows. The initial jet divergence angles increase with increasing gas density. Divergence begins progressively closer to the nozzle as gas density increases. Jet divergence angles increase with decreasing fuel viscosity; divergence begins at the nozzle exit once the liquid viscosity is below a certain level. Nozzle design affects the onset of the jet atomization regime. Jet divergence angles decrease

with increasing nozzle length. For the same length, rounded inlet nozzles produce less divergent jets than sharp-edged inlet nozzles. The initial jet divergence angle and intact spray length are quasi steady with respect to changes in operating conditions which occur on time scales longer than about 20 μs. [19] Note that while all these results were obtained under conditions where evaporation was not occurring, the atomization and initial spray-development processes are not that significantly affected by evaporation (see Sec. 10.4.6).

10.4.4 Spray Penetration

The speed and extent to which the fuel spray penetrates across the combustion chamber has an important influence on air utilization and fuel-air mixing rates. In some engine designs, where the walls are hot and high air swirl is present, fuel impingement on the walls is desired. However, in multi-spray DI diesel combustion systems, over-penetration results in impingement of liquid fuel on cool surfaces which, especially with little or no air swirl, lowers mixing rates and increases emissions of unburned and partially burned species. Yet under-penetration results in poor air utilization since the air on the periphery of the chamber does not then contact the fuel. Thus, the penetration of liquid fuel sprays under conditions typical of those found in diesel engines has been extensively studied.

Many correlations based on experimental data and turbulent gas jet theory has been proposed for fuel spray penetration. These predict the penetration S of the fuel spray tip across the combustion chamber for injection into quiescent air as occurs in larger DI engines, as a function of time. More recent work [21, 22] has compared vaporizing sprays with non-vaporizing sprays. For short times and penetrations, and long times and penetrations, the differences in jet penetration are small. In between, the differences are more significant.

Studies have examined spray tip penetration as a function of time, following the start of a diesel injection process in high-pressure constant-volume vessels. [22] The data shown in Fig. 10.27 illustrate the sensitivity of the spray tip position as a function of time to the ambient gas state for fuel jets injected into quiescent gas at a temperature of 1000 K. These data show the impact of changes in ambient gas density (ρ_a), the parameter that defines

the ambient resistance to liquid penetration. Injection pressure, especially at short time scales, is also significant: higher injection pressures result in faster penetration due to the higher initial liquid- fuel momentum. See Eq. (10.14) .

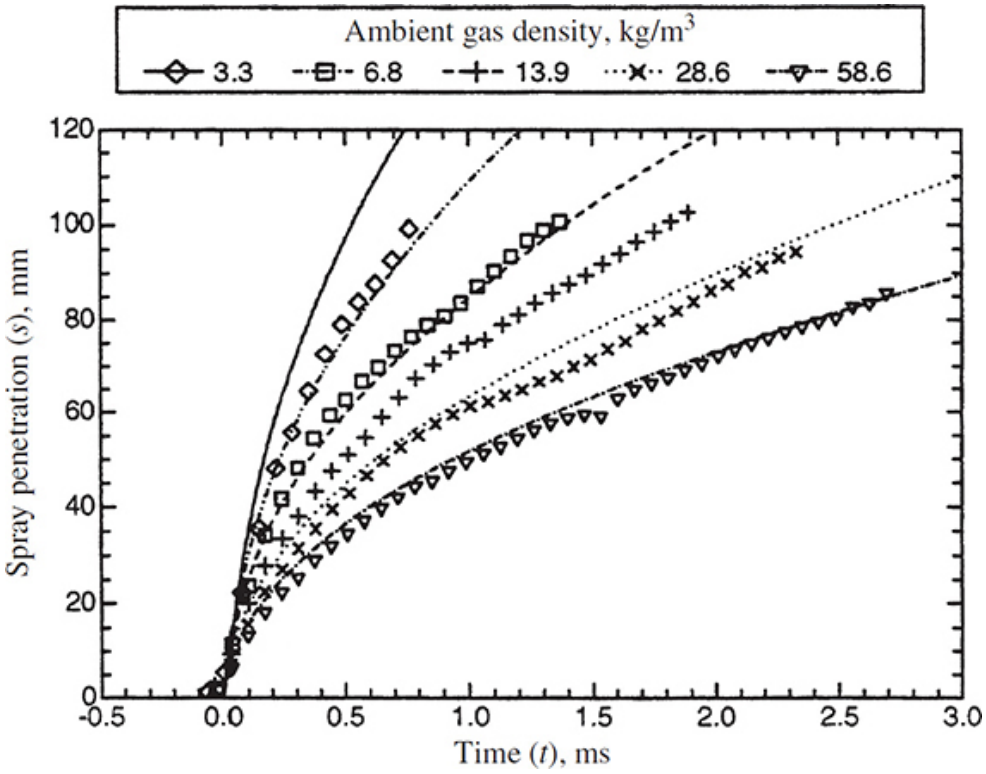


Figure 10.27 Measured penetration data as a function of time for vaporizing fuel sprays (symbols). Lines show predicted penetrations from Eq. (10.17) for non-vaporizing sprays. The nozzle orifice diameter, pressure drop across the orifice, and ambient gas temperature were 241 μm, 137 MPa, and 1000 K. [21]

A scaling law for non-vaporizing (isothermal) spray penetration developed by Naber and Siebers, [21] expressed in dimensionless form is:

$$\tilde{t} = \frac{\tilde{S}}{2} + \frac{\tilde{S}}{4}\sqrt{1+16\tilde{S}^2} + \frac{1}{16}ln\left(4\tilde{S} + \sqrt{1+16\tilde{S}^2}\right) \qquad (10.17)$$

where the dimensionless time and penetration length are given by

$$\tilde{t} = t / t^{+} \quad \text{and} \quad \tilde{S} = S/x^{+} \tag{10.18a,b}$$

t^{+} and x^{+} are the characteristics time and length scales of the fuel spray defined by:

$$t^{+} = \frac{d_n\sqrt{\tilde{\rho}}}{\tan(\alpha/2)U_f} \quad \text{and} \quad x^{+} = \frac{d_n\sqrt{\tilde{\rho}}}{\tan(\alpha/2)} \tag{10.19a,b}$$

The effective orifice diameter d_n, density ratio $\tilde{\rho}$, and liquid fuel velocity at nozzle exit U_f, are given by:

$$d_n = \sqrt{C_a}d, \tilde{\rho} = \rho_f/\rho_a, \tag{10.20a,b}$$

and

$$U_f = C_v\sqrt{2(p_f - p_a)/\rho_f} \tag{10.20c}$$

d is the fuel nozzle orifice diameter, C_a and C_v are the orifice area contraction coefficient and velocity coefficient, respectively, and p_f and p_a are the fuel injector pressure and ambient gas pressure.

Figure 10.28 compares non-vaporizing fuel spray penetration data with the dimensionless correlation [Eq. (10.17)]. Note the short-time and long-time penetration limits have a square-root-of-time, and a linear-time dependence, respectively, as shown. The transition occurs at $\tilde{t} = 1$ and $\tilde{S} = 0.7$ where the spray behavior transitions from being dominated by the injected fuel to being dominated by entrained gas.

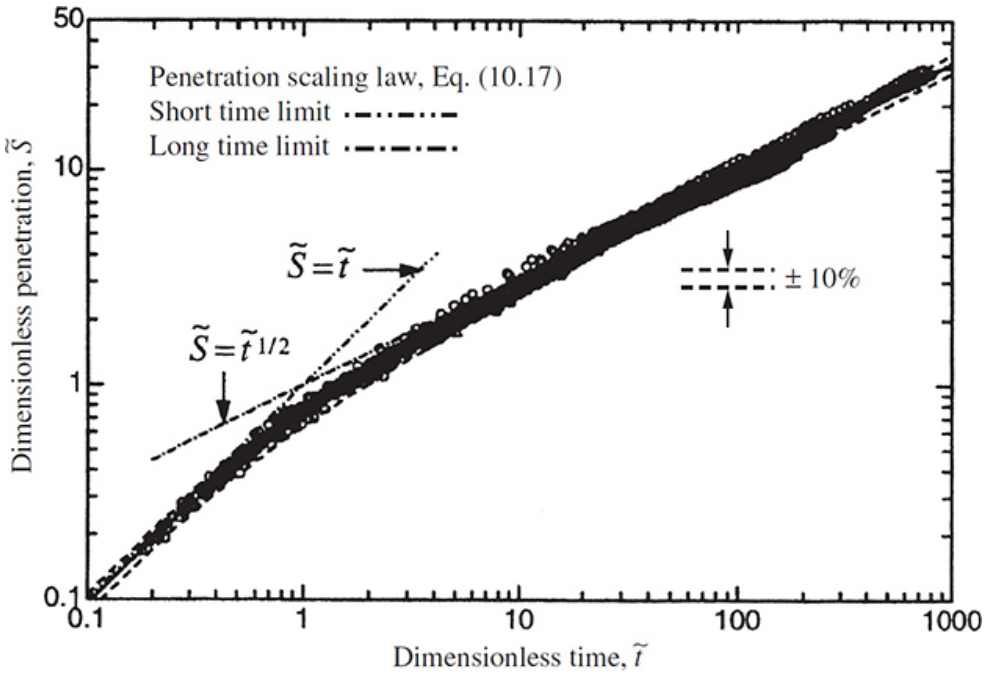


Figure 10.28 Comparison of measured non-vaporizing fuel spray penetration data with the penetration scaling law [Eq. (10.17)]. The data cover a wide range of conditions: the correlation predicts the data well. [21, 22]

An inverse version of Eq. (10.17) , shown below as Eq. (10.21) , gives the penetration distance $\tilde{S}$ as a function of time directly:

$$\tilde{S}=\tilde{t}/(1+\tilde{t}^{n/2})^{1/n} \tag{10.21}$$

where $n = 2.2$. This agrees closely with Eq. (10.17) in the near-field and far-field limits, and is within 5% in the transition region.[22]

Figure 10.29 shows the effect of combustion on spray development. The data were taken in the same constant-volume vessel with a single injected fuel spray. The ambient gas here contains oxygen to enable combustion. The lines are fits to the vaporizing spray data, where the ambient gas was not (1000 K), as with the combusting spray data but contained no oxygen. The combusting sprays (especially at the higher ambient gas densities) spread more quickly due to the chemical energy release within the spray which decreases the spray density (by raising the mixture temperature), thus causing the mixture within the spray to expand and move forward faster.

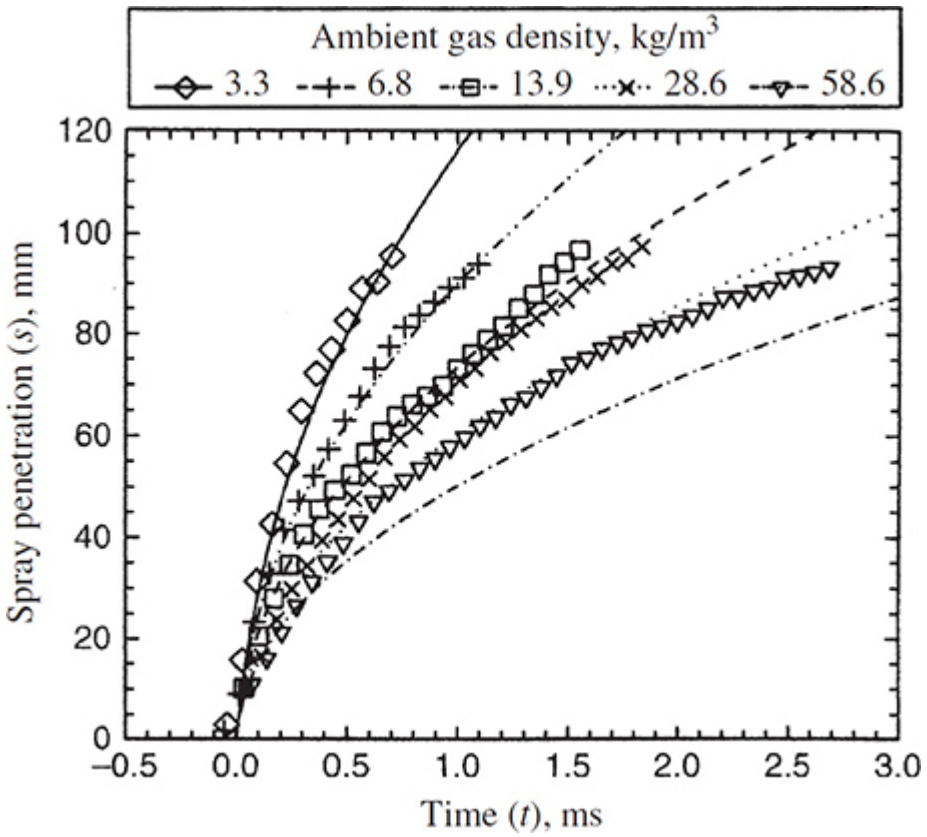


Figure 10.29 Penetration data for combustion fuel sprays as a function of time for a range of ambient gas densities. The orifice diameter, pressure drop across the injector orifice, and ambient gas temperature are the same as in Fig. 10.27. Curves are fits to vaporizing fuel spray penetration data in Fig. 10.27. [21, 22]

Figure 10.30 compares penetration data from vaporizing fuel sprays and combusting (and vaporizing) sprays on dimensionless $\tilde{S}$ versus $\tilde{t}$ plots, with the correlation for non-vaporizing sprays, Eq. (10.17) . Under vaporizing but non-combusting conditions, Fig. 10.30 *a*, the dimensionless spray penetration is slower by as much as 20% under the lower ambient density conditions (see Fig. 10.27). Fuel droplet evaporation cools the mixture within the spray thereby increasing its density and, through conservation of momentum, slowing it down. This is partly offset by reduced entrainment into the spray (as evidenced by the decrease in jet spreading angle; see Fig. 10.26 *b*), which would speed the spray up. The vaporizing and combusting spray behavior, Fig. 10.30 *b*, (which of course is the situation in an actual engine) is similar to that of the vaporizing but non-combusting spray up to

dimensionless time $\tilde{t}$ up to about 10 ($\tilde{S}$ about 2). Combustion has no impact until spontaneous ignition occurs and chemical energy release is initiated. The times of ignition for the higher three ambient gas density conditions in Fig. 10.29 are 0.6, 0.35, and 0.25 ms for the mid to highest density range, respectively. For times beyond these ignition times, combustion has a growing impact. The spray then decelerates increasingly more slowly as evidenced by the data beyond $\tilde{t}$ of about 10 to 20 (when ignition typically occurs). Ultimately, as $\tilde{t}$ increases above about 30, the combusting jet penetrates about 20% further than would an equivalent non-vaporizing jet. [22] This is likely due to the decreasing density of the within-the-spray mixture as hot combustion products produced within the diffusion flame around the spray diffuse into (and outward) from the spray boundary (see photos, Fig. 10.5). Reducing the spray density, since spray momentum is conserved, increases the spray mixture velocity. Flame height measurements of burning fuel sprays indicate that combustion is typically complete by $\tilde{t}$ of about 270 and $\tilde{S}$ of 16. For $\tilde{t}$ greater than this end-of-diffusion-flame time, the combusting spray and the vaporizing spray trends should converge. Figures 10.30 *a* and *b* indicate that is occurring.

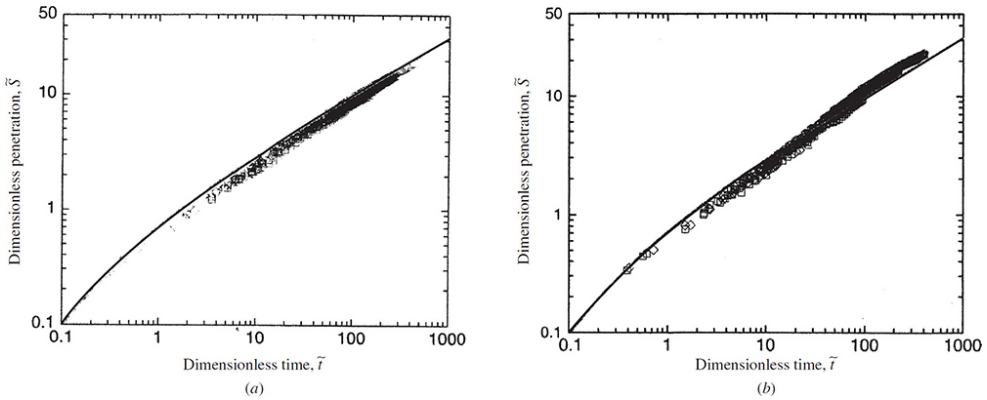


Figure 10.30 (*a*) Comparison between spray penetration measurements for vaporizing sprays under typical diesel conditions and the penetration scaling law for non-vaporizing sprays, Eq. (10.17) , in dimensionless penetration versus dimensionless time coordinates. (*b*) Similar comparison between combusting spray data, and non-vaporizing spray correlation. Conditions for two sets of data are the same except that ambient gas in the combusting experiments had 21% oxygen. [22]

In medium duty, and especially in high-speed light-duty DI diesel engines, air swirl is used to increase the fuel-air maxing rates. Increasingly higher swirl (rotation rate about the cylinder axis), created during intake and enhanced as the in-cylinder air is forced radially inward toward the cylinder axis during compression with reduced bowl diameter/bore ratios and deeper bowls (see Fig. 10.1), is used as cylinder size decreases and maximum engine speed increases. This enhanced air swirl affects the behavior of the four to six fuel sprays injected outward into the piston bowl from the cylinder axis.

The effect of air swirl on spray penetration is important. Figure 10.31 shows how the spray shape and location change as swirl is increased: it also shows how spray tip penetration varies with time and swirl rate.[23] These authors related their data on spray tip penetration with swirl, S_s, through a correlation factor to the equivalent penetration S, without swirl [given by an earlier equivalent to Eq. (10.17)]:

$$\frac{S_s}{S} = \left(1 + \frac{\pi R_s NS}{30 v_j}\right) \tag{10.22}$$

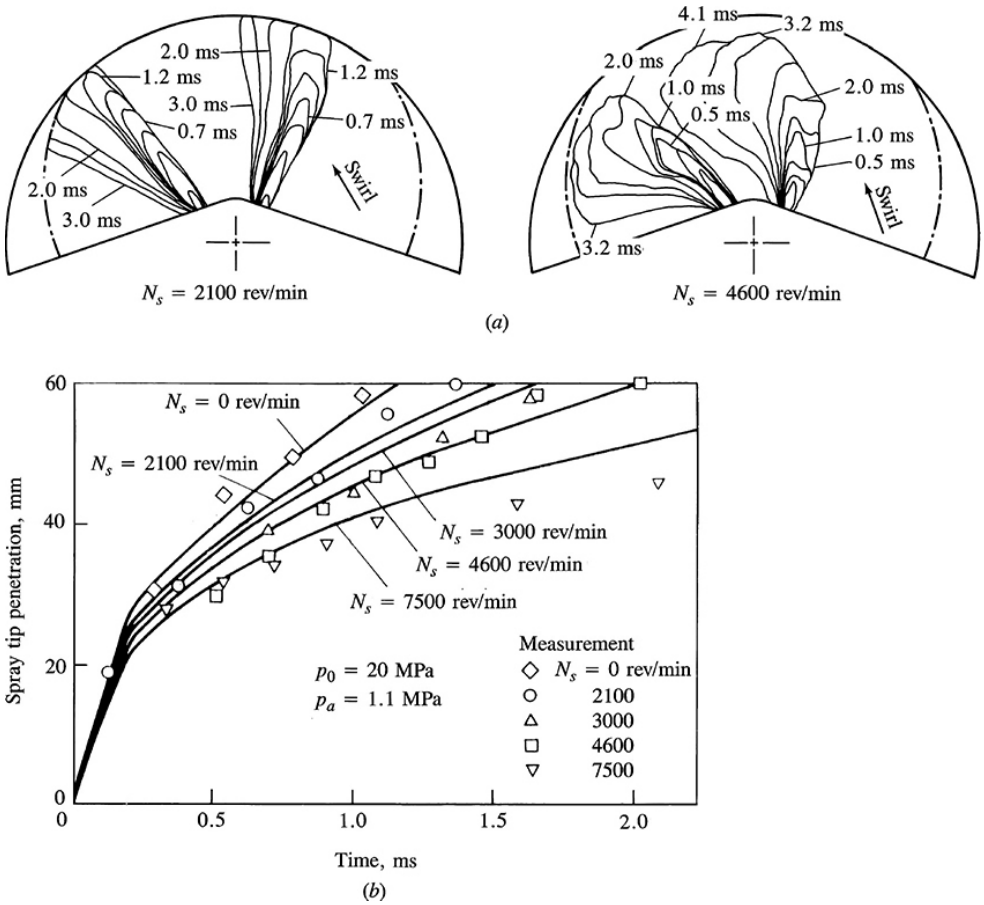


Figure 10.31 (*a*) Measured outer boundary of sprays injected into swirling airflow and (*b*) spray tip penetration as a function of time for different swirl rates. Solid lines show Eq. (10.22) .[23]

Here R_s is the swirl ratio which equals the swirl rate in revolutions per minute N_s divided by the engine speed N (rev/min), and v_j is the initial fuel jet velocity (meters per second). The curves in Fig. 10.31 *b* correspond to Eq. (10.22) . Swirl both reduces the radial penetration of the spray and spreads out the spray in the swirl direction, more rapidly.

10.4.5 Droplet Size Distribution

Previously in Sec. 10.4 , we have discussed the overall characteristics of the diesel fuel spray—its spreading rate and penetration into the combustion chamber. While the distribution of fuel via the spray trajectory throughout the combustion chamber is important, atomization of the liquid fuel into many

small rapidly moving drops is also necessary to create a large surface area across which liquid fuel can evaporate. Here we review how the drop sizes in the spray depend on injection parameters, and the air and fuel properties. Since the measurement of droplet characteristics in an operating diesel engine is extremely difficult, most results have come from studies of fuel injection into constant-volume chambers filled with high-pressure quiescent air, often at room temperature.

During the injection period, the injection conditions such as injection pressure, nozzle orifice area, and injection rate may vary. Consequently, the droplet size distribution at a given location in the spray may also change with time during the injection period. In addition, since the details of the atomization process are different in the spray core and at the spray edge, and the trajectories of individual drops depend on their size, initial velocity, and location within the spray, the drop size distribution is likely to vary with position within the spray. None of these variations has yet been adequately quantified.

The aerodynamic theory of jet breakup in the atomization regime summarized in Sec. 10.4.3 (based on work by G. I. Taylor) leads to the prediction that the initial average drop diameter D_d is proportional to the length of the most unstable surface waves:[16]

$$D_d = C\frac{2\pi\sigma}{\rho_g v_r^2}\lambda^* \tag{10.23}$$

Here σ is the liquid-fuel surface tension, ρ_g is the gas density, v_r is the relative velocity between the liquid and gas (taken as the mean injection velocity v_j), C is a constant of order unity, and λ^* is the dimensionless wavelength of the fastest growing wave. λ^* is a function of the dimensionless number $(\rho_l/\rho_g)(\mathrm{Re}_j/\mathrm{We}_j)^2$ where the jet Reynolds and Weber numbers are given by $\mathrm{Re}_j = \rho_l v_j d_n/\mu_l$ and $\mathrm{We}_j = \rho_l v_j^2 d_n\sigma$ and d_n is the nozzle orifice diameter. λ^* goes to 3/2 as this number increases above unity. Near the edge of the spray close to the nozzle, this equation predicts observed drop size trends with respect to injection velocity, fuel properties, nozzle L/d, and nozzle diameter, though measured mean drop sizes are larger by factors of 2 to 3.[25] However, within the dense early region of the spray,

secondary atomization phenomena—coalescence and breakup—occur which will change the droplet size distribution and mean diameter. The downstream drop size in the conical sprays used in diesel-injection systems is markedly influenced by both drop coalescence and breakup. Eventually a balance is reached as coalescence decreases (due to the expansion of the spray) and breakup ceases (due to the reduced relative velocity between the drops and the entrained gas). [26]

Measurements of droplet size distributions within simulated diesel sprays indicate how droplet size varies with location. Figure 10.32 shows several normalized cumulative drop size distributions superposed, which define a close-to-universal curve.

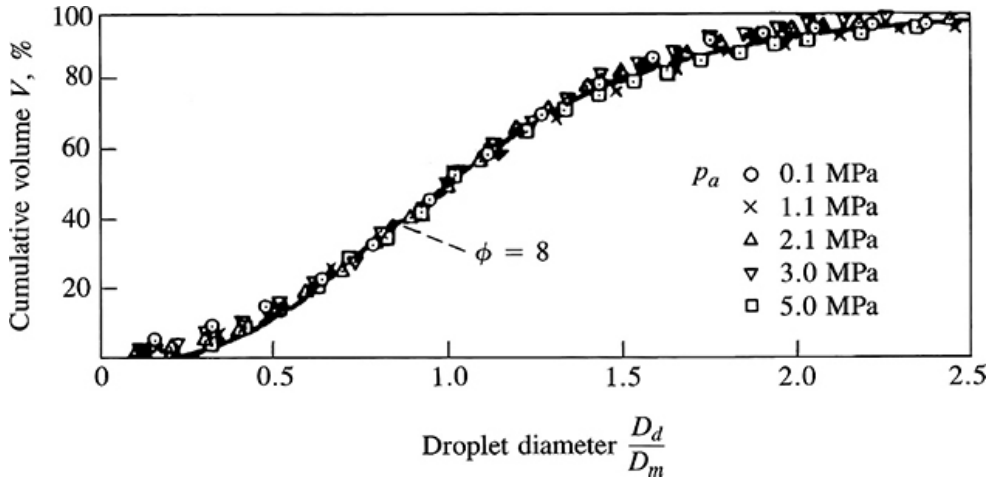


Figure 10.32 Normalized drop-size cumulative frequency distribution in spray injected into ambient-temperature air for air pressures from 0.1 to 5 MPa. Throttling pintle nozzle with nozzle opening pressure of 9.9 MPa. D_d is initial average drop diameter [see Eq. (10.23)]; median drop diameter $D_m = 1.22\, D_{SM}$.[24]

Drop size distributions are usually expressed as frequency distributions of drop volume. If $f(D_d)\, dD_d$ is the probability of finding a drop with diameter between D_d and $D_d + dD_d$, then

$$\int_0^\infty f(D_d) dD_d = 1 \tag{10.24}$$

and the volume distribution is

$$\frac{1}{V}\frac{dV}{dD_d} = \frac{f(D_d)D_d^3}{\int_0^\infty f(D_d)D_d^3\, dD_d} \tag{10.25}$$

With high-pressure DI diesel fuel injection systems [e.g., 1400 bar injection pressure, 200 mm^3 (160 mg) fuel delivered per cycle, 8 nozzle holes per cylinder with 0.2 mm orifice diameter], for a 12.9-liter medium-duty diesel engine rated at 290 to 300 kW at 2000 rev/min, and with fuel exiting the nozzle at a velocity of about 250 m/s, the average initial drop sizes is about 10 μm diameter. As the spray moves away from the injector orifices, the average drop size increases presumably due to some droplet coalescence and vaporization of the smaller drops in the distribution. The average drop size decreases as we move radially outward from the spray axis, presumably due to higher local concentrations of (hotter) entrained air, and higher droplet evaporation rates.

To characterize the spray, expressions for drop size distribution and mean diameter are needed. An appropriate and commonly used mean diameter is the *Sauter mean diameter*:

$$D_{SM} = \left(\int D_d^3\, dn\right) \Big/ \left(\int D_d^2\, dn\right) \tag{10.26}$$

where dn is the number of drops with diameter D_d between $D_d - dD_d/2$ and $D_d + dD_d/2$. The integration is usually carried out by summing over an appropriate number of drop size groups. The Sauter mean diameter is the diameter of the drop that has the same surface/volume ratio as that of the total distribution.

Various expressions for the distribution of drop sizes in liquid sprays have been proposed. One, by Hiroyasu and Kadota, [24] is based on the chi-square statistical distribution fit to the available experimental data. Figure 10.32 shows that the chi-square distribution with a degree of freedom equal to 8 fits well to the measurements. Here, D_m is the median drop diameter which for this chi-square curve is 1.22 D_{SM}. The non-dimensional expression for drop size distribution in sprays injected through hole nozzles,

pintle nozzles, and throttling pintle nozzles given by the chi-square distribution is

$$\frac{dV}{V} = 13.5\left(\frac{D_d}{D_{SM}}\right)^3 \exp\left[-3\left(\frac{D_d}{D_{SM}}\right)\right] d\left(\frac{D_d}{D_{SM}}\right) \tag{10.27}$$

An empirical expression for the Sauter mean diameter D_{SM} (in micrometers) for typical diesel fuel properties given by Hiroyasu and Kadota [24] is

$$D_{SM} = A(\Delta p)^{-0.135} \rho_a^{0.121} V_f^{0.131} \tag{10.28}$$

where Δp is the mean pressure drop across the nozzle in megapascals, ρ_a is the air density in kilograms per cubic meter, and V_f is the amount of fuel delivered per cycle per cylinder in cubic millimeters per stroke. A is a constant which equals 25.1 for pintle nozzles, 23.9 for hole nozzles, and 22.4 for throttling pintle nozzles. Other expressions for predicting D_{SM} can be found in Ref. 27.

The effects of injection pressure, nozzle geometry and size, air conditions, and fuel properties on Sauter mean diameter in sprays obtained with diesel fuel-injection nozzles have been extensively studied. Various immersion, photographic, and optical techniques for making such measurements have been used. Some of the major effects are illustrated in Figs. 10.33 and 10.34, which show average Sauter mean diameters determined optically from studies of steady diesel fuel sprays in a pressurized vessel. Figure 10.33 shows that nozzle size effects the mean drop size in the expected direction. Nozzle length/diameter ratio is also shown to be important: an $L_n/d_n = 4$ gives the minimum mean drop size at low and intermediate injection pressures. This L_n/d_n also corresponds to the minimum value of spray breakup length and to the maximum spray cone angle. Fuel viscosity and surface tension also affect mean drop size as shown in Fig. 10.34, with the effects being most significant at lower injection pressures.

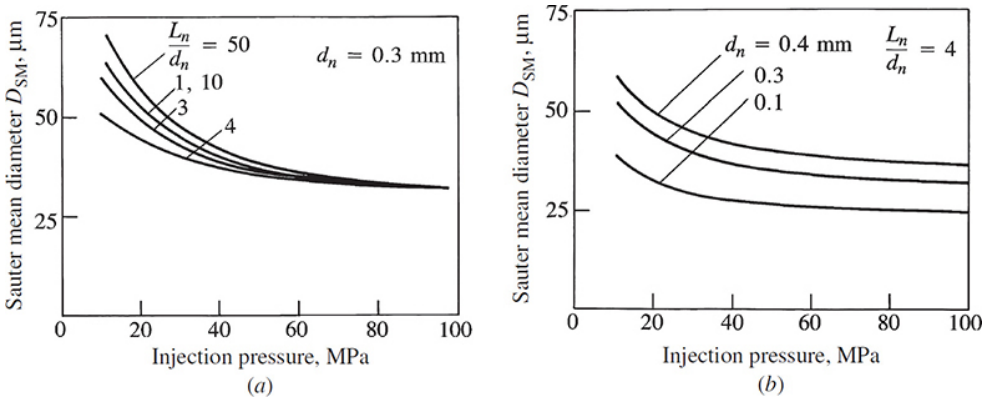


Figure 10.33 Effect of fuel-injection pressure and nozzle geometry and size on Sauter mean drop diameter: (*a*) effect of nozzle length/diameter ratio L_n/dn and injection pressure and (*b*) effect of nozzle diameter d_n and injection pressure.[28]

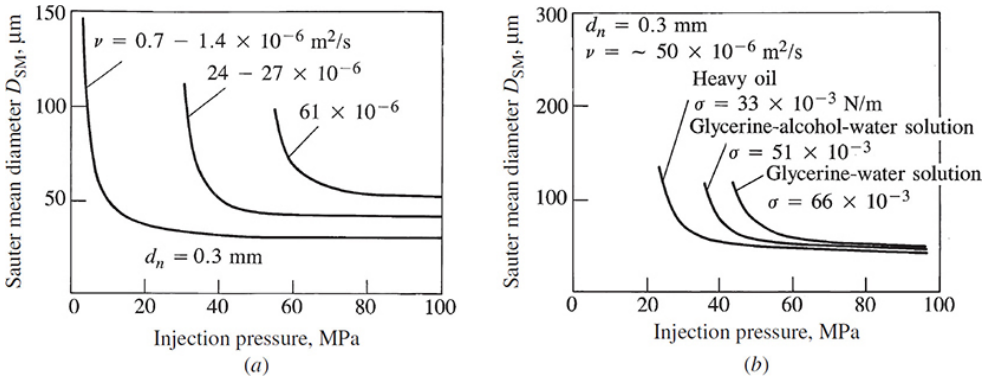


Figure 10.34 Effect of (*a*) liquid viscosity *v* and (*b*) liquid surface tension *s* on Sauter mean drop diameter as a function of injection pressure. Air conditions: 3 MPa and ambient temperature. [27]

10.4.6 Spray Evaporation

The injected liquid fuel, atomized into small drops near the nozzle exit to form a spray, must evaporate before it can mix with air and burn. Figure 10.21 showed the basic structure of an evaporating fuel spray under conditions typical of a large DI diesel engine. Back illumination showed that a core exists along the axis of the spray where the liquid fuel ligaments or drops are sufficiently dense to attenuate the light beam. Once the start-up phase of the injection process is over, the length of this liquid-containing

core remains essentially constant until injection ends. This core is surrounded by a much larger vapor-containing spray region which continues to penetrate deeper into the combustion chamber, the core extends only partway to the spray tip.

Additional insight into the physical structure of evaporating sprays can be obtained from the schlieren photographs taken just after the end of injection in a prechamber engine with air swirl, shown in Fig. 10.35. The lowest magnification picture (A) shows the overall structure of the spray. The only liquid-containing region evident is that part of the core nearest the nozzle which shows dark on the left of the photograph. The spreading vapor region of the spray, carried around the chamber by the swirling airflow, appears mottled due to local turbulent vapor concentration and temperature fluctuations. The dark region within the spray vapor region is due to soot formed where the fuel vapor concentration is sufficiently high. It is probable that, after the breakup length, the dense liquid core of the spray is primarily composed of droplets but the concentration is so high along the optical path that the light beam is fully extinguished. However, the last part of the core close to the nozzle tip (B) disperses sufficiently for individual features to be resolved. The small black dots are liquid fuel drops in the size range of 20 to 100 μm. Fuel drop vapor trails can be observed in the highest magnification photo (C) corresponding to various stages of evaporation. These range from drops showing little surrounding vapor to vapor trails with little liquid remaining at the head. The vapor trails show random orientations relative to the spray axis, presumably due to local air turbulence. The process of droplet evaporation under normal engine operating conditions appears to be rapid relative to the total combustion period. [15]

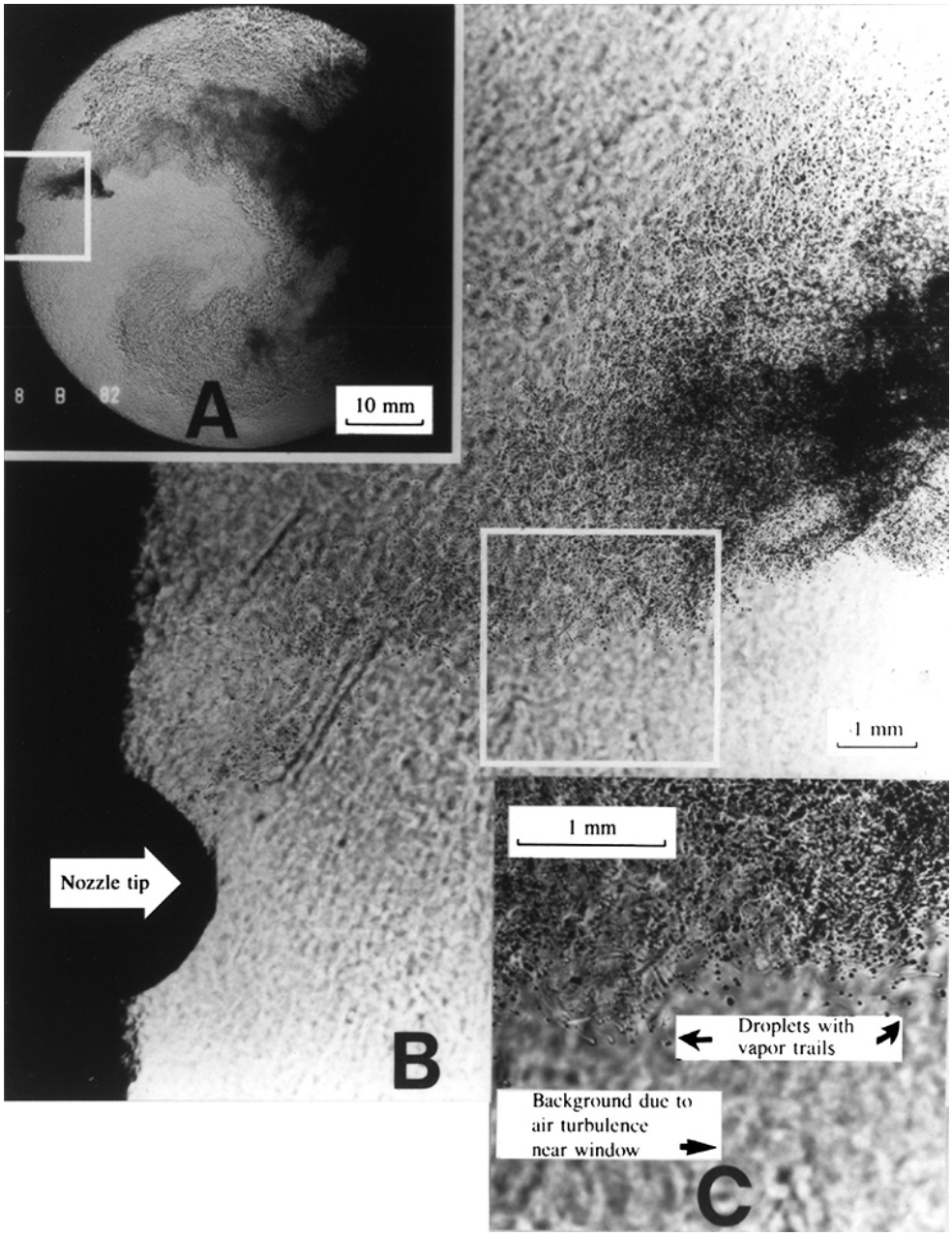


Figure 10.35 Shadowgraph photographs at three magnifications taken just after the end of injection of a diesel-fuel spray into swirling airflow in the prechamber of a diesel engine with optical access. Nozzle hole diameter = 0.25 mm. [15]

Let us now examine the drop evaporation process in more detail. Consider a liquid drop at close to ambient temperature injected into air at

typical end-of-compression engine conditions. Three phenomena will determine the history of the drop under these conditions:

1. Deceleration of the drop due to aerodynamic drag
2. Heat transfer to the drop from the air
3. Mass transfer of vaporized fuel away from the drop

As the droplet temperature increases due to heat transfer from the hot surrounding air, the fuel vapor pressure increases and the evaporation rate increases. As the mass transfer rate of vapor away from the drop increases, so the fraction of the heat transferred to the drop surface which is available to increase further the drop temperature decreases. As the drop velocity decreases, the convective heat-transfer coefficient between the air and the drop decreases. The combination of these factors gives the behavior shown in Fig. 10.36 where drop mass, temperature, velocity, vaporization rate, and heat-transfer rate from the air are shown schematically as a function of time following injection. Analysis of individual fuel drops 25 μm in diameter, injected into air at typical diesel conditions, indicates that evaporation times are usually less than 1 ms. [29, 30]

Such an analysis is relevant to drops that are widely separated (e.g., at the edge of the spray). In the spray core, where drop number densities are high, the evaporation process has a significant effect on the temperature and fuel-vapor concentration in the air within the spray. As fuel vaporizes, the local air temperature will decrease and the local fuel vapor pressure will increase. Eventually thermodynamic equilibrium would pertain: this process is usually called adiabatic saturation. [29] Calculated thermodynamic equilibrium temperatures for diesel spray conditions are plotted in Fig. 10.37 as a function of the fuel/air mass ratio for *n*-dodecane and *n*-heptane. The initial liquid fuel temperature was 300 K. The ratio of fuel vapor to air mass at these equilibrium conditions is also shown. To the left of the peaks in the m_{fv}/m_a curves, only fuel vapor is present. To the right of these peaks, liquid fuel is also present because the vapor phase is saturated. [30] Liquid fuel vaporization causes substantial reductions in gas temperature. While this equilibrium situation may not be reached within the spray, these results are useful for understanding the temperature distribution within an evaporating spray.

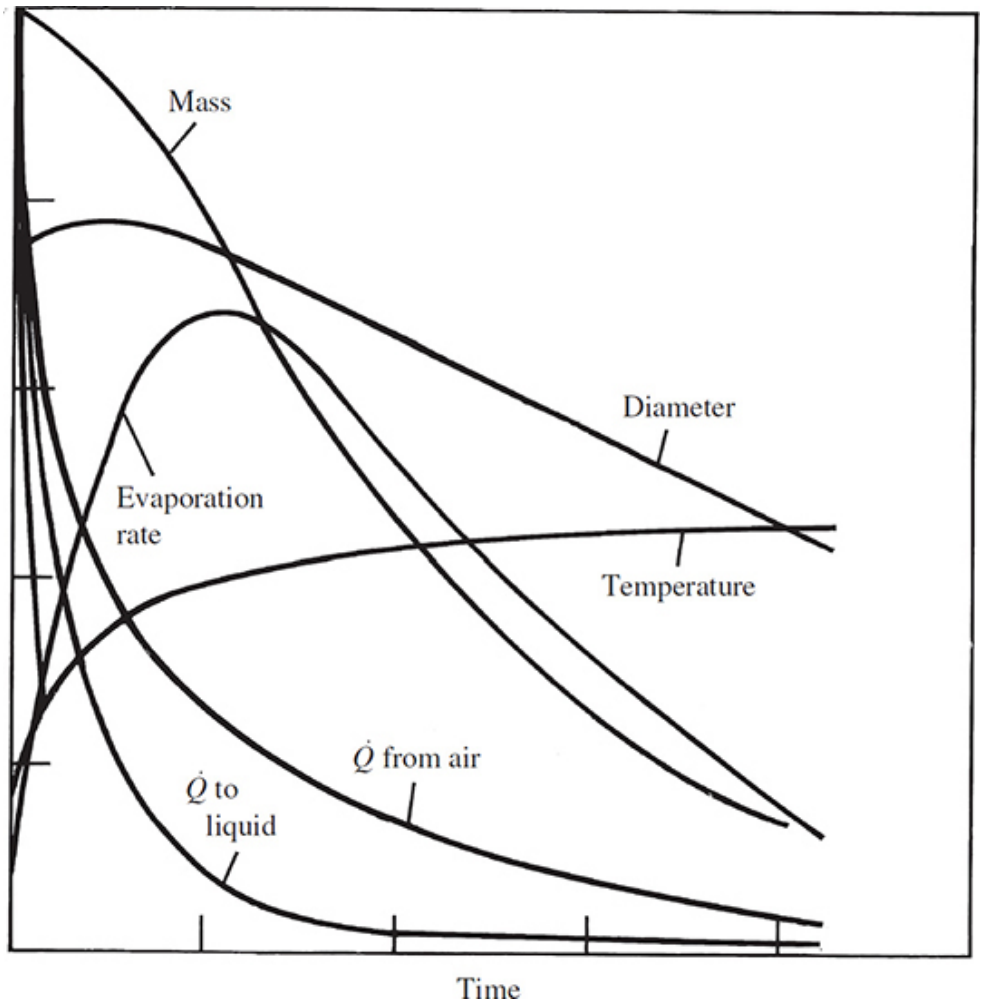


Figure 10.36 Schematic of variation of mass, diameter, temperature, evaporation rate, heat-transfer rate from air, and heat-transfer rate to liquid drop core as function of time during evaporation process of individual drop in diesel environment at the time of injection.[29]

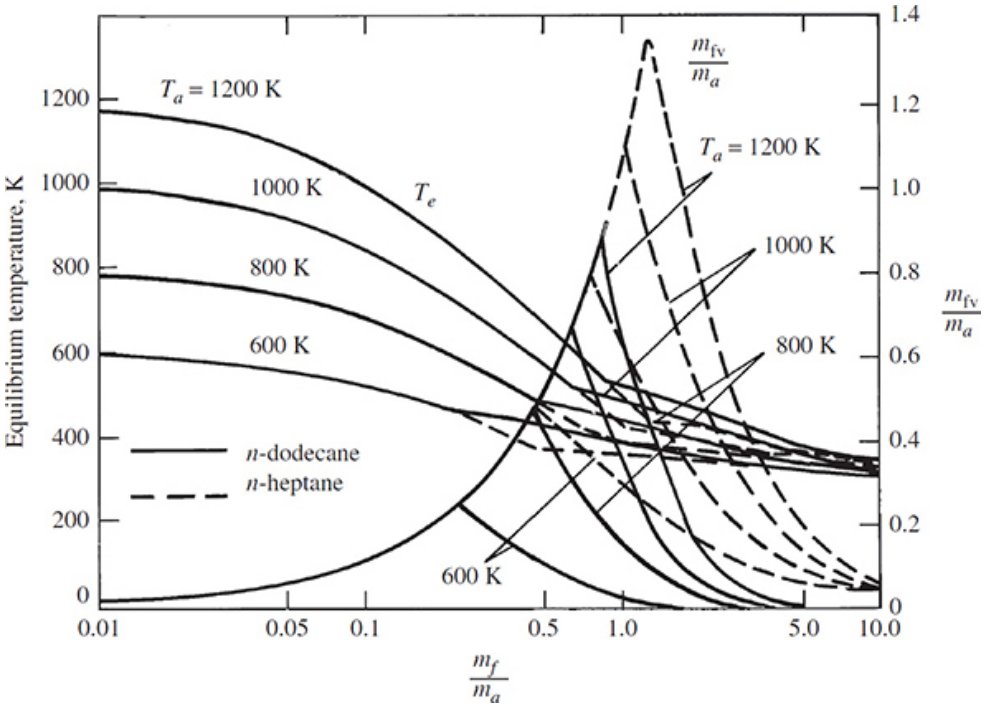


Figure 10.37 Adiabatic-saturation conditions for equilibrium mixtures formed by injecting *n*-dodecane and *n*-heptane, initially liquid at 300 K, into air at initial temperature T_a between 600 K and 1200 K and initial density 6.5 kg/m^3. Equilibrium mixture temperature (T_e) and ratio of fuel vapor mass (m_{fv}) to air mass (m_a) shown as function of ratio of total fuel mass m_f to m_a. Fuel vapor only present to left of peaks in m_{fv}/ma curves: liquid fuel also present to right of peaks. [31]

To quantify accurately the fuel vaporization rate within a diesel fuel spray requires the solution of the coupled conservation equations for the liquid droplets and the air within the combustion chamber. Various phenomenological models and computational fluid dynamic models have been developed for this purpose (see Secs. 14.4.3 and 14.5.5). In the more sophisticated of these, the spray is assumed to be composed of discrete computational particles each of which represents a group of droplets of similar size, temperature, etc. The distribution functions in droplet size, velocity, temperature, etc. produced by the fuel injector are statistically sampled and the resulting discrete particles are followed along lagrangian trajectories as they interact and exchange mass, momentum, and energy with the surrounding gas. Drops interact directly with each other via collisions

and indirectly via evaporation by modifying the ambient vapor concentration and gas temperature. Studies with such models indicate that, under normal diesel engine conditions, 70 to 95% of the injected fuel is in the vapor phase at the start of combustion. Evaporation is more than 90% complete after about 1 ms. However, only 10 to 35% of the vaporized fuel has mixed to within flammability limits in a typical medium-speed DI diesel engine. Thus combustion is largely mixing-limited, rather than evaporation-limited. [32] Of course, under cold-starting conditions, evaporation becomes a major constraint.

The extent of the liquid-containing core of the spray is an important factor. Initially, the fuel drops move rapidly into the engine's combustion chamber thus promoting entrainment of air into the spray, and fuel-air mixing. Figure 10.21 shows that this liquid core initially penetrates increasingly into the combustion chamber within the overall spray. It then ceases to grow in length, while the spray continues to penetrate due to its momentum, because the liquid fuel drops that constitute this liquid core have now evaporated. This maximum core penetration distance, relative to the distance along the spray axis from nozzle orifice to combustion chamber wall, is an important fuel injection system parameter. Overpenetration of liquid fuel results in wall wetting leading to slower fuel vaporization and to higher hydrocarbon and particulate emissions. The maximum liquid fuel penetration distance—the so-called liquid length within the fuel spray—thus needs to be characterized. It depends on the basic spray-initiating parameters (drop size and velocity, and ambient gas density) as well as the fuel's vaporization parameters (volatility characteristics, fuel and ambient gas temperatures). Optical techniques are usually used to determine liquid (core) lengths for sprays injected into closed vessels under diesel-like conditions.

Figure 10.38 shows the effects of orifice diameter and orifice pressure drop for heptamethylnonane (HMN), cetane, and No. 2 diesel fuel, for a range of conditions relevant to diesel operation. The liquid length rises linearly with orifice size, but is essentially independent of orifice pressure drop. Both ambient air density which impacts the deceleration of the droplets and the spray, and ambient air temperature which impacts the droplet evaporation process, have significant effects.

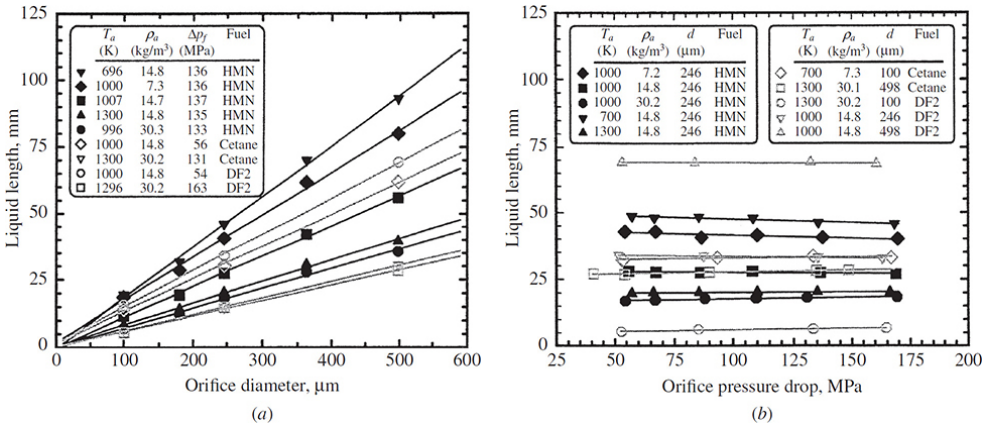


Figure 10.38 Measured liquid lengths—the maximum liquid fuel penetration distances—as a function of (*a*) nozzle orifice diameter and (*b*) orifice pressure drop, for different ambient gas temperatures and densities. Fuels are HMN, cetane, and No. 2 diesel fuel. Lines are least square fits to the data. [32]

A mixing-limited fuel vaporization process has been shown to explain these results. [22] The fuel injection rate is proportional to the injection velocity, which varies with the square root of the orifice pressure and the orifice diameter squared (through Eq. [10.15]). The total air entrained into the spray up to any axial location is proportional to that axial distance, the orifice diameter, and the injection velocity. [23] These dependences indicate that to entrain enough hot air to fully vaporize the fuel, the axial extent of the liquid core will increase in proportion to the increase in orifice diameter, as the data indicates. With respect to the fuel injection velocity, these dependencies mean that the amount of air entrained up to any given axial liquid-core tip location, and the fuel injection rate, scale with injection velocity. Thus the *liquid length* is independent of the orifice pressure drop, as Fig. 10.38 *b* indicates. [22]

A scaling law for liquid-phase fuel penetration can be derived from an energy balance for a one-dimensional model of an evaporating spray, in addition to spray mass and momentum balances. [32] At the maximum liquid core penetration (the liquid length), just enough energy has been transferred from the hot entrained air to the liquid drops to fully vaporize the fuel. It is also assumed that at the liquid length, saturated fuel vapor conditions pertain, and that air entrainment rates for the vaporizing spray differ little from those

of a non-vaporizing spray. That is, the spray penetration length scaling law, Eq. (10.17) and its inverse Eq. (10.21) defines the overall spray behavior.[32] That liquid-length scaling law, in dimensionless form is,

$$\tilde{L} = 0.47\sqrt{\left(\frac{2}{B(T_a, p_a, T_f)} + 1\right)^2 - 1} \tag{10.29}$$

where $\tilde{L} = L_t / x^+$ with x^+ defined by Eq. (10.19 *b*) , and

$$B(T_a, p_a, T_f) = \frac{Z_a(T_s, p_a - p_s) p_s M_f}{Z_f(T_s, p_s)(p_a - p_s) M_a} = \frac{h_a(T_a, p_a) - h_a(T_s, p_a - p_s)}{h_f(T_s) - h_f(T_f, p_a)} \tag{10.30}$$

The term B results from the energy balance on the fuel spray, and an equation of state. The expressions in Eq. (10.30) are for single component fuels. Equation (10.30) is an implicit equation for the saturated fuel/ambient-gas mixture temperature T_s, at the liquid length, in terms of the fuel and ambient gas properties, and the initial fuel and ambient gas conditions. The temperature T_a is the ambient gas temperature. The pressures p_a and p_s are the ambient gas pressure and the saturation (i.e., partial) pressure of the fuel at the liquid length. Thus the partial pressure of the entrained ambient gas in the fuel jet is $p_a - p_s$. The (saturated) fuel vapor pressure p_s is related to the saturation temperature T_s through the saturation pressure-temperature relationship of the fuel. The variables h_f and h_a , and M_f and M_a, are the specific enthalpies and molecular weights of the fuel and ambient gas, while Z_f and Z_a are the compressibilities of fuel vapor and the ambient gas at saturated conditions. (Note that at such conditions, gases deviate from the ideal gas behavior—see App. B—and the ideal gas law is then written as $p = \rho\, \tilde{Z}\, RT/M$.) The fuel at the nozzle orifice exit is assumed to be at the injector tip temperature T_f and ambient gas pressure p_a. The enthalpy different in the numerator of the right-hand side of Eq. (10.30) is the specific enthalpy transferred *from* the entrained gas to the fuel: the enthalpy difference in the denominator is the specific enthalpy required to heat up and vaporize the liquid fuel.

Figure 10.39 shows a comparison of the liquid length scaling correlation, Eq. (10.30) , with data, as a function of B. The conditions are representative of diesel in-cylinder ambient conditions at TC, given in Ref. 21, using HMN and cetane fuel. B varies from about 0.2 to 1.7 as the dimensionless liquid length (L_l/x^+) decreases from about 5 to 1. The correlation fits the data well.

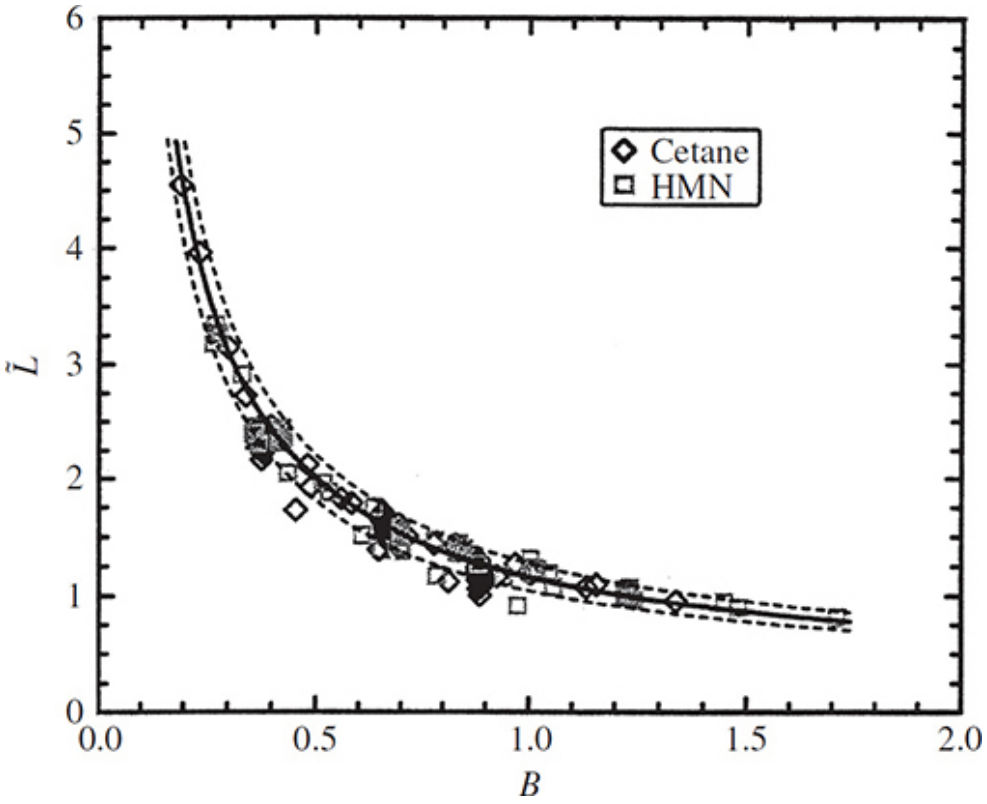
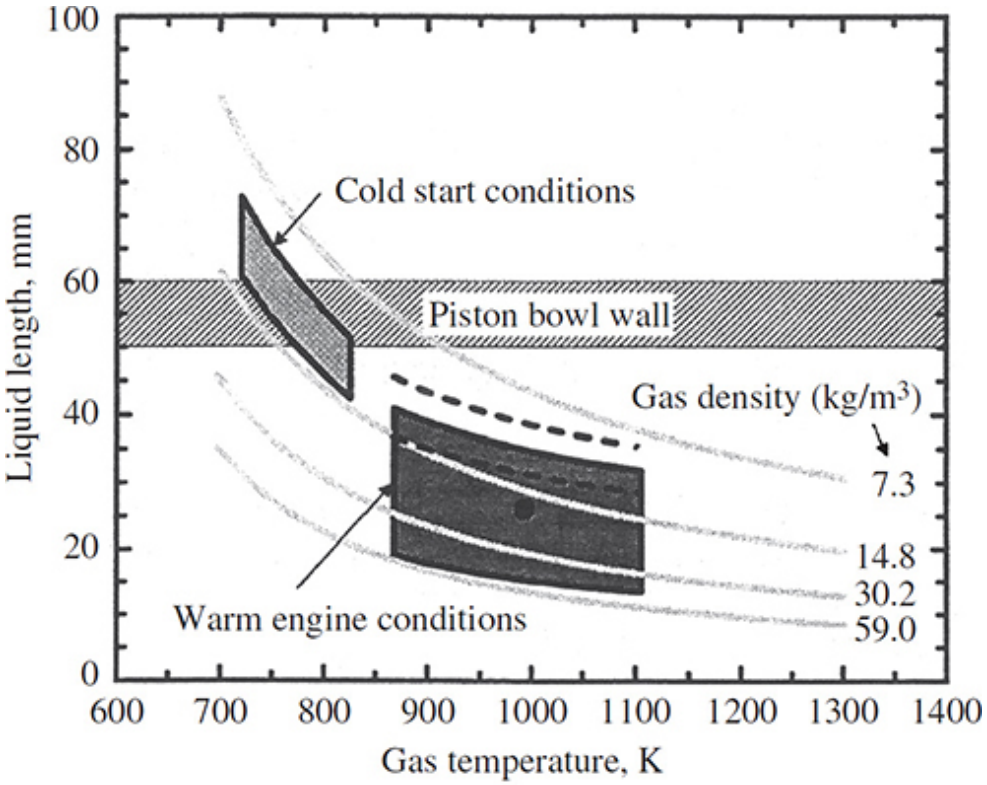


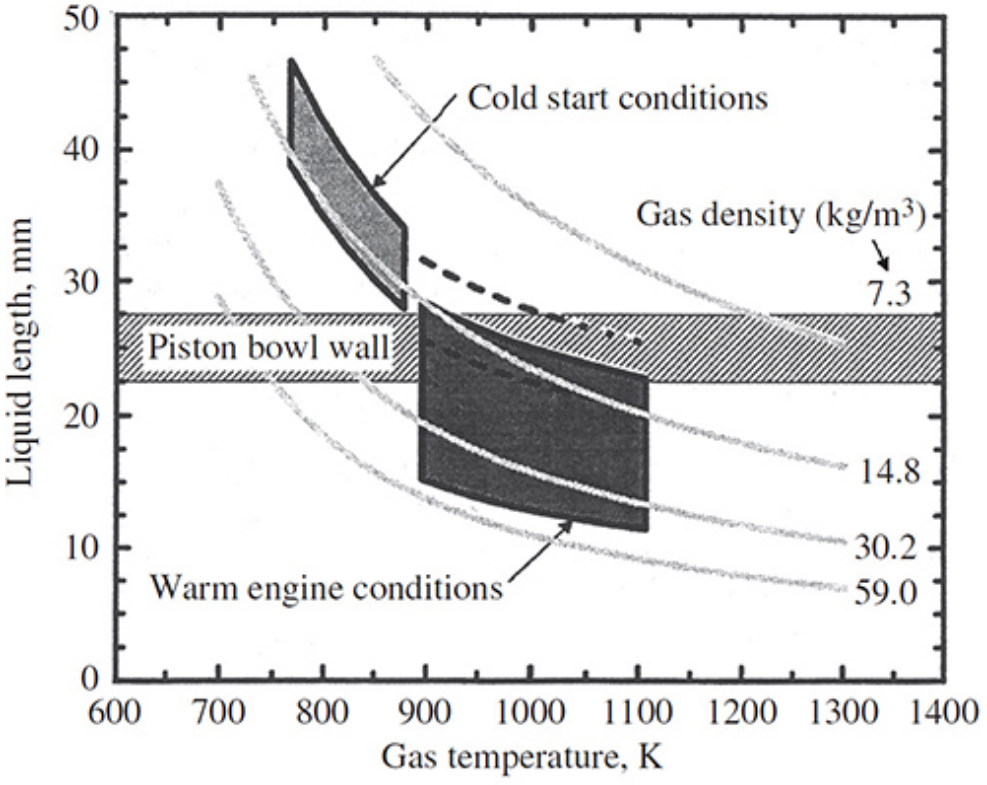
Figure 10.39 Dimensionless liquid length [$\tilde{L} = L_l/x^+$; see Eq. (10.29)] as a function of B [see Eq. (10.30)], an expression that results from an energy balance of the fuel spray and its entrained air, and equations of state for fuel vapor and air.[32]

The relevance of this liquid fuel core penetration length for (larger) heavy-duty DI diesels and (smaller) high-speed light-duty DI diesels has been examined and provides useful insights into the behavior of evaporating sprays under both warmed-up and cold-start engine conditions. Table 10.3 gives the assumed geometry and operating details of the two diesel engines considered: a 12-liter six-cylinder large truck engine and a 2-liter four-cylinder high-speed passenger car engine. Their bowl-in-piston combustion

chambers correspond to those shown in Figs. 10.1*a* and *c*. Figure 10.40 shows liquid lengths for these two engines, predicted with the correlation Eq. (10.29) as a function of TC in-cylinder ambient gas temperature (typically 1000 K). The upper and lower edges of the darker shaded regions correspond to boost pressures of 0.7 and 3 bar, respectively. The lighter shaded regions correspond to cold start conditions with ambient gas temperatures at TC of about 800 K. With the smaller bore engine, liquid impingement in the bowl wall is more likely to occur. Note, however, that the smaller high-speed engine will have high swirl, which would reduce the actual liquid fuel penetration somewhat below that given by the no-swirl correlation.


100
80
60
40
20
0
Liquid length, mm
Cold start conditions
Piston bowl wall
Gas density (kg/m3)
7.3
14.8
30.2
59.0
Warm engine conditions
600
700
800
900
1000
1100
1200
1300
1400
Gas temperature, K

(*a*)


50
40
30
20
10
0
Liquid length, mm
Cold start conditions
Gas density (kg/m3)
7.3
Piston bowl wall
14.8
30.2
59.0
Warm engine conditions
600
700
800
900
1000
1100
1200
1300
1400
Gas temperature, K

(*b*)

Figure 10.40 Liquid lengths predicted for (*a*) heavy-duty DI diesel and (*b*) (smaller) high-speed DI diesel engines defined in Table 10.3. The dark gray regions are for warmed-up engine operating conditions; the light gray regions denote cold start conditions. Typical piston bowl wall distances from the fuel injector nozzle orifice are indicated. The light gray lines are for constant in-cylinder gas densities, at TC of compression. [32]

TABLE 10.3 Parameters used in liquid-length scaling law predictions for heavy- and light-duty DI diesel engines [32]

	Heavy-duty	Light-duty	Units
Bore	135–140	80–85	mm
Bowl diameter/bore	0.75	0.5	—
Compression ratio	15	19	—
Orifice diameter	194	160	μm
Orifice C_a	0.84	0.84	—
Warm operation:			
Intake temperature	60–160	45–120	°C
Intake pressure	0.7–3.0	0.7–2.3	bar
Fuel temperature	100	100	°C
Cold start:			
Intake temperature	0–40	0–40	°C
Intake pressure	0.7–1	0.7–1	bar
Fuel temperature	0–40	0–40	°C

10.5 IGNITION DELAY

10.5.1 Definition and Discussion

The ignition delay in a diesel engine is defined as the time (or crank angle) interval between the start of injection and the start of combustion. The SOI is usually taken as the time when the injector needle lifts off its seat. The SOC is more difficult to determine precisely. It is often identified from the change in slope of the heat-release rate (from negative to positive) calculated from the cylinder pressure data using the techniques described in Sec. 10.3.3 . This occurs shortly after detectible heat (chemical energy) release begins, which offsets the slight decline in cylinder pressure due to fuel evaporation

within the developing spray. However, hydrocarbon reaction chemistry is apparent within the spray before this point, evidenced by the onset of luminosity (chemiluminescence). This luminosity effectively defines the onset of the breakup of the fuel molecules and the initiation of partial fuel oxidation, and precedes the point of pressure rise by some 100 μs, about 1 crank-angle degree.

Both physical and chemical processes must take place before a noticeable fraction of the chemical energy of the injected liquid fuel can be released. The physical processes are: the atomization of the liquid fuel into droplets close to the injection nozzle exit; the vaporization of some of these fuel droplets; the mixing of fuel vapor with air that has been entrained into the spray. The chemical processes are the pre-major-heat-releasing combustion reactions of the fuel, air, residual gas mixture, that lead to autoignition. These processes are affected by engine design and operating variables, and fuel characteristics, as this section (10.5) examines.

Good atomization requires high fuel-injection pressure, small injector hole diameters, appropriate fuel viscosity, and high cylinder air pressure at the time of injection (see Sec. 10.4.3). The rate of vaporization of the fuel droplets depends on the size of the droplets, their distribution, and their velocity, the pressure and temperature inside the combustion chamber, and the volatility of the fuel. The rate of fuel-air mixing is controlled largely by injector and combustion chamber design. Some cylinder-head and piston-crown shapes are designed to amplify swirl and create turbulence in the air charge during compression. IDI engine designs use a prechamber or swirl chamber to create the vigorous air motion necessary for rapid fuel-air mixing. In DI engines, injector design features such as the number and spatial arrangement of the injector holes determine the injected fuel distribution. The details of each nozzle hole affect the spray cone angle. The penetration of the spray depends on the size of the fuel droplets, the injection pressure, the air density, and the air-flow characteristics. The arrangement of the sprays, the spray cone angle, the extent of spray penetration, and the airflow all affect the rate of air entrainment into the spray. These physical aspects of fuel-injection and fuel-spray behavior are reviewed in Sec. 10.4 .

The chemical component of the ignition delay is controlled by the precombustion reactions of the fuel. A fundamental discussion of autoignition or spontaneous hydrocarbon oxidation in premixed fuel-air mixtures is given in Sec. 9.6.2 . Since the diesel engine combustion process is heterogeneous,

its spontaneous ignition process is especially complex. Ignition occurs in regions where fuel vapor has mixed with some air, through the cracking of large hydrocarbon molecules to smaller molecules (partial oxidation reactions). These chemical processes depend on the composition of the fuel and the cylinder charge temperature and pressure, as well as the physical processes described above which govern the distribution of fuel within each spray.

Since the spontaneous ignition characteristics of the fuel affect the ignition delay, this property of a fuel is very important in determining diesel engine operating characteristics such as fuel conversion efficiency, smoothness of operation, misfire, smoke emissions, noise, and ease of starting. The ignition quality of a fuel is defined by its *cetane number*. Cetane number is determined by comparing the ignition delay of the fuel with that of primary reference fuel mixtures in a standardized engine test (see Sec. 10.5.2). For low cetane fuels with too long an ignition delay, most of the fuel will be injected before ignition occurs, which results in very rapid burning rates once combustion starts with high rates of pressure rise and high peak pressures. Under extreme conditions, when the autoignition of much of the injected fuel occurs after a long delay (e.g., under cold-starting conditions), this produces an audible "knocking sound," often referred to as diesel knock. For higher cetane number fuels, with shorter ignition delays, ignition occurs before most of the fuel has been injected. The rates of heat release and pressure rise are then controlled primarily by the rate of injection and fuel-air mixing, and smoother engine operation results.

The timing relationships between the various measures of autoignition onset and the fuel injection process, as a function of the in-cylinder air temperature at SOI (close to TC), are shown in Fig. 10.41. (Note that 1 ms at 1800 rev/min corresponds to about 10 crank-angle degrees.) Typical end-of-compression temperatures in a warmed-up engine are close to 1000 K. The luminosity onset precedes the pressure recovery point at 1000 K by about 0.2 ms (about 2 degrees CA). Under cold-start conditions (750 to 800 K air temperature), the pressure rise lag is significantly longer. The mixture temperature values where the autoignition chemistry transitions from low- to intermediate- to high-temperature reaction schemes are indicated: [33] Section 10.5.3 will further discuss these details.

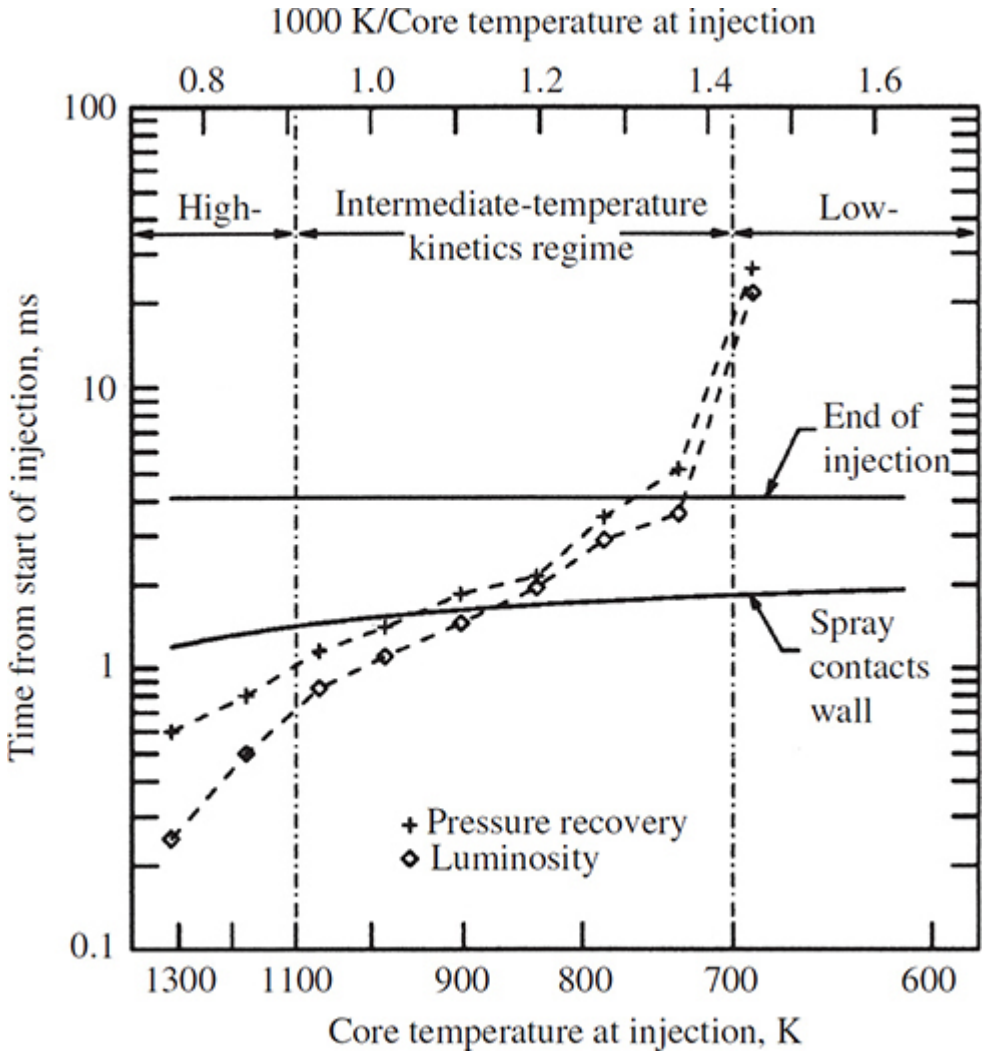


Figure 10.41 Timing of several fundamental features of the diesel spray autoignition process as a function of in-cylinder air temperature at time of injection. Symbols show measured times (in a constant volume combustion vessel) of luminosity onset and pressure recovery (rise) occurrence. Solid lines show approximate times for the spray to impinge on the chamber walls, and of end of injection. Ignition did not occur for gas temperatures below about 750 K. [33]

Pilot injection (the injection of 5 to 10% of the fuel some 10 to 20 degrees before the start of the main injection) is a strategy enabled by the development of multi-pulse fuel injection technology (see Sec. 10.7.1) that is used to shorten the ignition delay. The spontaneous ignition and combustion of the pilot fuel creates a hotter local gas environment around the

fuel injector into which the main injection sprays penetrate. The higher temperature of the ambient gas (mainly air), which is then entrained into the main-injection sprays, raises the within-the-spray mixture temperature thus speeding up the premixed-mixture ignition chemistry.

10.5.2 Fuel Ignition Quality

The ignition quality of a diesel fuel is defined by its *cetane number*. The method used to determine the ignition quality in terms of cetane number is analogous to that used for determining the antiknock quality of gasoline in terms of octane number. The cetane number scale is defined by blends of two pure hydrocarbon reference fuels. Cetane (*n*-hexadecane, $C_{16}H_{34}$), a hydrocarbon with high ignition quality, represents the top of the scale with a cetane number of 100. An iso-cetane, 2-2-4-4-6-8-8 heptamethyl-nonane (HMN), also $C_{16}H_{34}$, which has a very low ignition quality, represents the bottom of the scale with a cetane number of 15. [i] Thus, cetane number (CN) is given by

$$\text{CN} = \text{percent } n\text{-cetane} + 0.15 \times \text{percent HMN} \tag{10.31}$$

The engine used in cetane number determination is a standardized single-cylinder, variable compression ratio, engine with special loading and accessory equipment and instrumentation. The engine, the operating conditions, and the test procedure are specified by ASTM Method D 613. [34] The operating requirements include: engine speed, 900 rev/min; coolant temperature, 100°C; intake air temperature, 65.6°C (150°F); injection timing, 13° BTC; injection pressure, 10.3 MPa (1500 lb/in. [2]). With the engine operating under these conditions, on the fuel whose cetane number is to be determined, the compression ratio is varied until combustion starts at TC: that is, an ignition delay period of 13° (2.4 ms at 900 rev/min) is produced. The above procedure is then repeated using reference fuel blends. Each time a reference fuel is tried, the compression ratio is adjusted to give the same 13° ignition delay. When the compression ratio required by the actual fuel is bracketed by the values required by two reference fuel blends differing by less than five cetane numbers, the cetane number of the fuel is determined by

interpolation between the compression ratios required by the two reference blends.

Because of the expense of the cetane number test, many correlations which predict ignition quality based on the physical properties of diesel fuels have been developed. A calculated *cetane index* (CCI) is often used to estimate ignition quality of diesel fuels (ASTM D976 [35]). It is based on API gravity and the mid-boiling point (temperature 50% evaporated). It is applicable to straight-run fuels, catalytically cracked stocks, and blends of the two. Its use is suitable for most diesel fuels and gives numbers that correspond quite closely to cetane number. A *diesel index* is also used. It is based on the fact that ignition quality is linked to hydrocarbon composition: n-paraffins have high ignition quality, and aromatic and napthenic compounds have low ignition quality. The aniline point (ASTM D611 [36]), the lowest temperature at which equal volumes of the fuel and aniline become just miscible, is used together with the API gravity to give the diesel index:

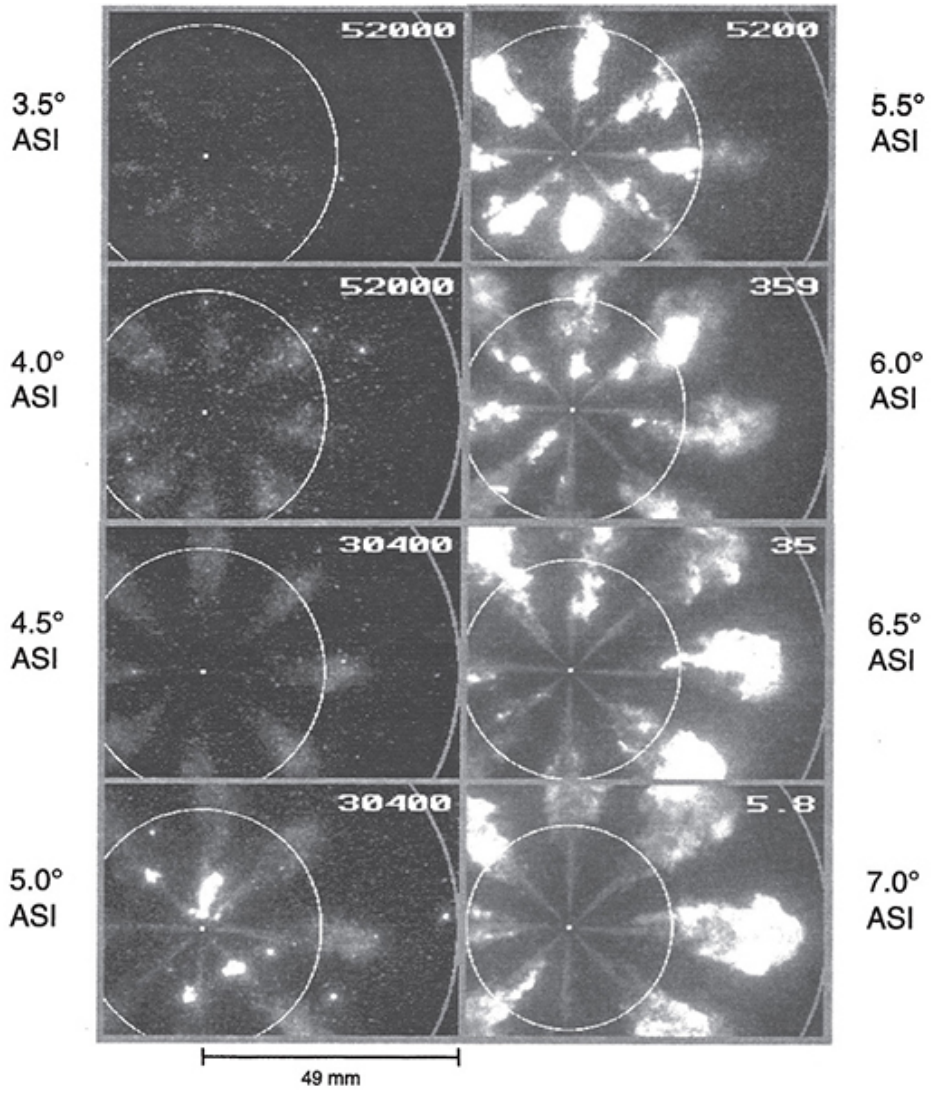
$$\text{Diesel index} = \text{aniline point } (^\circ\text{F}) \times \text{API gravity}^{j}/100 \tag{10.32}$$

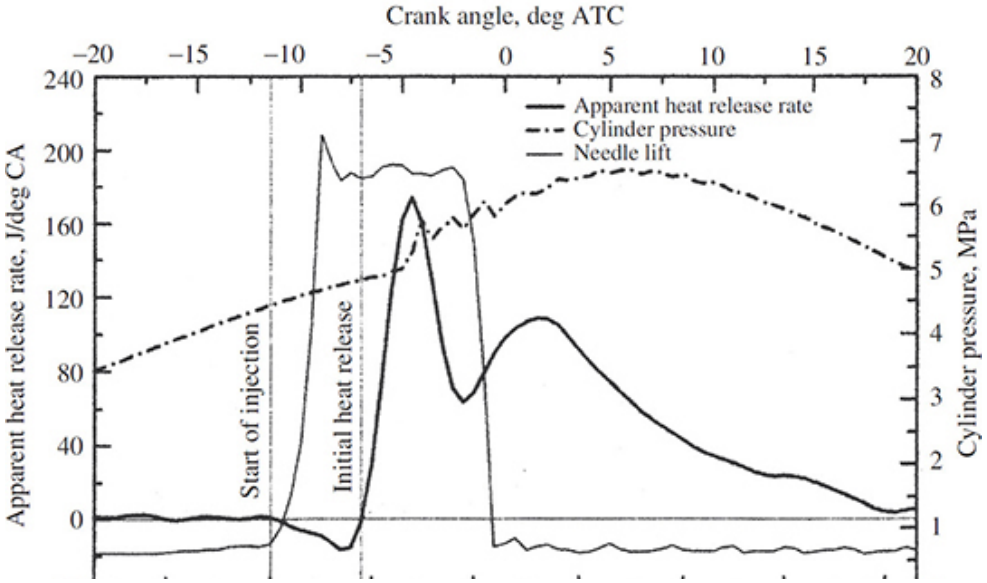
The diesel index depends on the fact that aromatic hydrocarbons mix completely with aniline at comparatively low temperatures, whereas paraffins require considerably higher temperatures before they are completely miscible. Similarly, a high API gravity denotes low specific gravity and high paraffinicity and, again, good ignition quality. The diesel index usually gives values slightly above the cetane number. It provides a reasonable indication of ignition quality in many (but not all) cases.

10.5.3 Autoignition and Premixed Burn

Basic studies in constant-volume combustion vessels, in steady-flow reactors, and in rapid-compression machines have been used to examine the various stages by which autoignition of fuel-air mixtures under conditions relevant to diesel engines occurs. In some of these studies the fuel and air were premixed; in many, fuel injection was used. Studies with fuel injection into constant-temperature and pressure environments, similar to conditions in a diesel engine at TC of compression, have identified the key steps in the process. The start of the diesel combustion process—spontaneous autoignition, followed by a short and rapid "premixed-burning" process—is

illustrated by the natural flame emissions images in Fig. 10.42 with the diagram showing the fuel injection profile (injector needle lift), apparent heat release rate and the cylinder pressure versus crank angle. [37] Faint (low intensity) chemiluminescence emission is apparent at 3.5° after start of injection (ASI), from the early injected fuel vapor/entrained air mixture in the leading region of these eight fuel sprays. These regions are beyond the "liquid length," the maximum penetration of the liquid fuel containing spray core (see Sec. 10.4.6). This chemiluminescence (emitted because fuel molecule breakdown and partial oxidation is occurring) becomes much stronger (4.0° ASI and 4.5° ASI) and clearly visible throughout the downstream region of all the sprays. Note that the apparent heat release rate is now rising sharply: the premixed burn is getting underway.


52000
5200
3.5°
ASI
5.5°
ASI
52000
359
4.0°
ASI
6.0°
ASI
30400
35
4.5°
ASI
6.5°
ASI
30400
5.8
5.0°
ASI
7.0°
ASI
49 mm


Crank angle, deg ATC
−20
−15
−10
−5
0
5
10
15
20
240
200
160
120
80
40
0
8
7
6
5
4
3
2
1
Apparent heat release rate
Cylinder pressure
Needle lift
Start of injection
Initial heat release
Apparent heat release rate, J/deg CA
Cylinder pressure, MPa

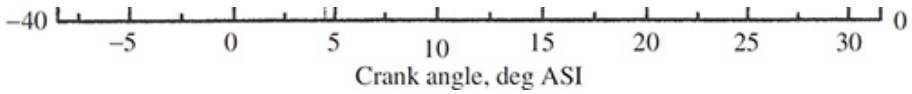


Figure 10.42 Sequence of natural flame emissions images taken through a window in the piston crown of an optical-access DI diesel engine with eight fuel sprays and 140-mm bore, operating at naturally-aspirated engine conditions and 1200 rev/min. Timing of each image, after start of injection (ASI), and relative light intensity gain (upper right of each image) are indicated. Bottom of figure: apparent heat-release rate, cylinder pressure, injector needle lift profiles versus crank angle. [37]

The early stages of the (weaker) chemiluminescence are primarily the result of contributions from formaldehyde (CH_2O) and CH emissions [see reactions in Eq. (9.63) , an outline of the hydrocarbon autoignition chemistry], corresponding to the "cool-flame" and "preignition glow" phenomena in rich mixtures. And the leading regions of these diesel sprays contain relatively well-mixed rich fuel-vapor/entrained air mixture in the fuel-air equivalence ratio range of 2 to 4. [37, 38]

The hydrocarbons autoignition initiating mechanism is similar to the chemistry that initiates spark-ignition engine knock (see Sec. 9.6.2). At sufficiently high temperatures, a hydrocarbon air mixture undergoes a sequence or chain of reactions that produces an ever increasing pool of radicals (highly reactive species) through radical chain propagating and branching reactions. As the bonds in the basic hydrocarbon molecule are successively broken, chemical energy is released at a first slowly then rapidly increasing rate. Radical removing (chain-terminating) reactions also occur, offsetting the radical build-up to varying extent as chemistry and energy release proceed. See the reaction set in Eq. (9.63) which further delineates this process.

The much brighter emission images in Fig. 10.42 (from about 5° to 6° ASI) result from soot particle radiation. Fuel molecule breakdown in the rich fuel-vapor/air mixture within the spray, in the early chemiluminescence phase, leads to the formation of PAHs which are precursors in the soot formation process (see Secs. 11.5.3 and 11.5.4 on diesel particulate formation). These PAHs form rapidly, grow into nuclei, and grow and

agglomerate into soot (primarily carbon) particles. Significant chemical energy (heat) release is now occurring (6° ASI, Fig. 10.42).

The final images, 6.5° and 7° ASI, in Fig. 10.42 show the development of diffusion flames around the individual sprays. These diffusion flames, where oxidation of the fuel is completed, are located at the spray boundaries with "fuel" (very rich mixture consisting of fuel vapor, partially reacted fuel molecules, PAHs, soot particles, CO, and H_2) on the inside of the flame and air on the outside, constitute the second mixing-controlled phase of diesel combustion where the majority of the heat release occurs. By extrapolating the rounded portion of the heat release rate curve (which is this second mixing-controlled phase) at its left-hand end down to the horizontal axis, one sees that these diffusion flames should become apparent at 6° to 6.5° ASI. Note that these developing burning-spray images correspond to the illustrative schematic already shown in Fig. 10.11. Autoignition usually commences a few crank-angle degrees after start of fuel injection, close to TC, in the rich fuel-vapor/entrained-air mixture in the leading region of each developing spray, begins with the breakdown of fuel molecules through high-enough velocity molecular collisions to initiate this bond-breaking process. It then continues and transitions into the (rapid) premixed burning phase as the fuel hydrocarbon fragments build-up into PAHs, which continue to grow in molecular size into soot particles releasing significant chemical energy in the process and increasing the temperature within each spray (see Fig. 10.13). [37]

A final point to note relates to the magnitude of the peak premixed-burn heat-release rate and its relative timing (see Fig. 10.42 for an illustrative example). As in-cylinder gas conditions such as ρ_g and T_g change, and as a consequence the ignition delay shortens or lengthens, so the premixed-burn peak heat-release rate decreases or increases. With shorter delays, the amount of rich fuel-vapor/entrained-air mixture within the sprays ready for spontaneous autoignition to occur decreases so its chemical energy available for rapid release decreases. The mixing-controlled burning heat-release-rate profile then absorbs the additional fuel to be burned in this second mode without major change. However, as the ignition delay lengthens significantly (and the premixed-burn fuel amount increases substantially) then the mixing-controlled burning phase is more noticeably affected. [37]

Ignition delay data from actual and simulated engine experiments have usually been correlated by equations of the form:

$$\tau_{id} = Ap^{-n} \exp\left(\frac{E_A}{\tilde{R}T}\right) \tag{10.33}$$

In general, τ_{id} is a function of mixture temperature, pressure, equivalence ratio, and fuel properties (though no accepted form for the variation with equivalence ratio is yet established). In the studies summarized and referenced in the first edition of this text, [1] the fuel was injected into a uniform air environment where the pressure and temperature only changed due to the cooling effect of the fuel-vaporization and fuel-heating processes. In an engine, pressure and temperature change during the delay period due to the compression resulting from piston motion. To account for the effect of changing conditions on the delay the following empirical integral relation has been used:

$$\int_{t_{si}}^{t_{si}+\tau_{id}} \left(\frac{1}{\tau}\right) dt = 1 \tag{10.34}$$

where t_{si} is the time of start of injection, τ_{id} is the ignition delay period, and τ is the ignition delay at the conditions pertaining at time t. Whether the variation in conditions is significant depends on the amount of injection advance before TC that is used and the length of the delay.

The first edition of this text [1] compared several published Arrhenius equations for ignition delay of the form of Eq. (10.33) . See also Ref. 39. These earlier correlations no longer seem useful for several reasons. In-cylinder pressures and temperatures at TC continue to change as turbocharging dominates and boost levels increase. Such conditions are difficult to realize in out-of-engine experiments. Increasing injection pressures and high swirl in smaller high-speed DI engines are reducing the contribution to the delay that these enabling physical processes contribute. Also, using a single Arrhenius expression to represent the overall chemistry of the autoignition process is too simplistic given the complex chemistry actually involved.

An illustration of that chemistry is shown in Fig. 10.43 where a simpler hydrocarbon molecule *n*-heptane (C_7H_{16}), which has a cetane number of 60 (close to that of diesel fuel which is about 50), was used diesel fuel

surrogate. The detailed chemical kinetic model (using 550 species and 2454 reactions) is briefly described in the reference. [11] The chemistry calculation is at a starting temperature of 900 K, pressure of 83 bar, and fuel-air equivalence ratio of 4, appropriate conditions for an autoignition, and premixed-burn simulation in a fuel spray in a turbocharged diesel engine. Autoignition commences at about 0.03 msec (about 3° ASI) and the low-temperature first stage (and modest) energy release occurs (temperature rise of some 150 K). Just after 0.07 msec (7° ASI), the second stage of the ignition process (high-temperature) chemistry begins which rapidly raises the temperature in the leading half of the spray plume to rise to 1650 K (see also Fig. 10.14), through a larger energy release (raising the temperature by some 400 K or more). After 0.1 to 0.15 msec (1 to 1.5°) ASI, the rich mixture within the spray contains H_2O, CO_2, CO, CH_4, soot precursors (acetylene and polyacetylenes), and lower carbon atom hydrocarbons (C_3 and C_4) along with some 60% nitrogen. By energy fraction (mass fraction × heating value), the combustibles H_2, CO, CH_4, soot precursors, and lower carbon number hydrocarbons are 11, 21, 16, 40, and 12%, respectively, of the *unreleased* rich-mixture chemical energy. Some 25% of the fuel's full heating value has been released at this point. This is the "fuel" on the fuel side of the diffusion flame, during the mixing-controlled combustion phase. [11]

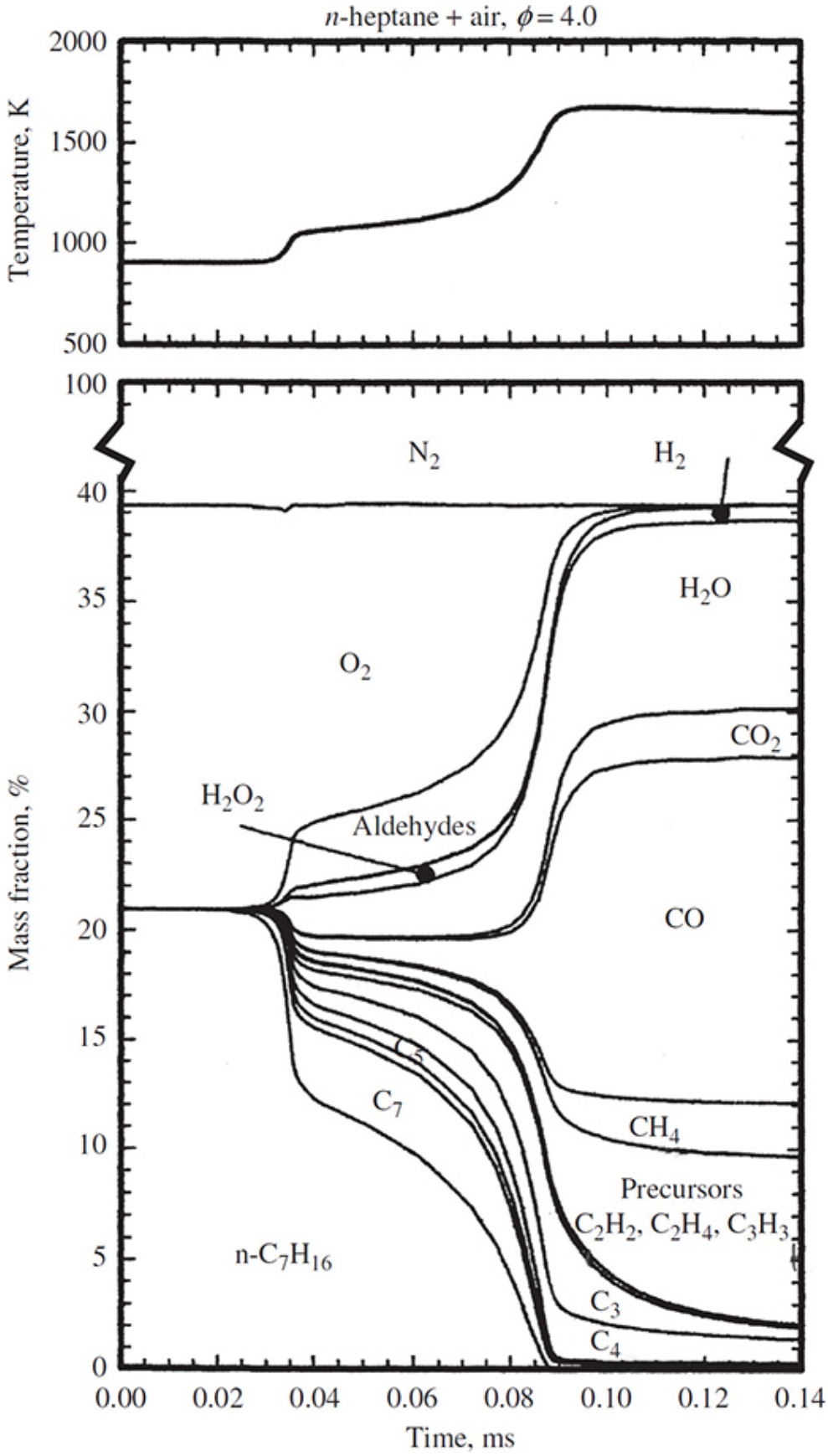

n-heptane + air, $\phi = 4.0$
Temperature, K
2000
1500
1000
500
100
40
35
30
25
20
15
10
5
0
Mass fraction, %
N_2
H_2
H_2O
O_2
CO_2
H_2O_2
Aldehydes
CO
C_5
C_7
CH_4
Precursors
C_2H_2, C_2H_4, C_3H_3
$n\text{-}C_7H_{16}$
C_3
C_4
0.00
0.02
0.04
0.06
0.08
0.10
0.12
0.14
Time, ms

Figure 10.43 Results of chemical kinetic analysis of *n*-heptane vapor reaction with air, at $f = 4$ and initial mixture temperature of 900 K and pressure 83 bar (typical TC diesel engine conditions). The evolving species concentrations and gas temperature in this rich mixture within each diesel spray are shown over time (0 to 0.14 ms). [11]

Such chemical-kinetic autoignition modeling studies indicate a strong dependence of reaction time scales on ambient temperatures. For example, an increase from 750 K to 1000 K reduces reaction times from about 1 to 0.1 ms. Ignition delays of some few degrees crank angle, 0.2 to 0.3 ms, are needed for normal diesel engine operation. [11]

10.5.4 Physical Factors Affecting Ignition Delay

The physical variables that control the development of the fuel spray, and the air-charge state (its pressure, temperature, and velocity), will influence the ignition delay. These quantities depend on the design of the fuel-injection system and combustion chamber, and the engine operating conditions. The injection system variables affecting the fuel-spray development are injection timing, quantity, velocity, rate, drop size, and spray form or type. The relevant charge conditions depend on the combustion system employed, the details of the combustion chamber design, inlet air pressure and temperature, compression ratio, injection timing, residual gas amount and state, coolant and oil temperature, and engine speed. Data on these interactions are available for different types of diesel engines. The trends observed with the different diesel combustion systems are generally similar, though details may differ. In this section the ignition delay trends during fully warmed-up engine operation are discussed. In modern diesel engines, ignition delays are short: a few crank-angle degrees. Thus, changes in delay in warmed-up engines are now less significant. However, the dependence of the ignition delay on engine design and operating variables during engine starting and warm-up is important, and may be different from fully warmed-up behavior due to lower air temperatures and pressures.

Injection Timing

At normal engine conditions (low to medium speed, fully warmed engine), the minimum delay occurs with the start of injection at about 10° BTC. The increase in the delay with earlier or later injection timing occurs because the air temperature and pressure change. If injection starts earlier, the initial air temperature and pressure are lower so the delay will increase. If injection starts later (closer to TC) the temperature and pressure are initially slightly higher but then decrease as the delay proceeds. The most favorable conditions for ignition lie in between.

Injection Quantity or Load

Figure 10.44 shows the effect of increasing injection quantity or engine load on ignition delay. The delay decreases approximately linearly with increasing load for this DI engine. As load is increased, the residual gas temperature and the wall temperature increase, as does heat transfer to the air charge during the first half of compression. This results in higher charge temperature (and, to a lesser extent, charge pressure) at injection, thus shortening the ignition delay. When adjustment is made for this increasing temperature, it is found that increasing the quantity of fuel injected has little significant effect on the delay period under normal operating conditions. Under engine starting conditions, however, the delay increases significantly due to the larger drop in mixture temperature: engine components are cold, there is no in-cylinder air heating, and significantly more air cooling during compression. This latter result should be expected since evaporation of the injected fuel further reduces mixture temperature.

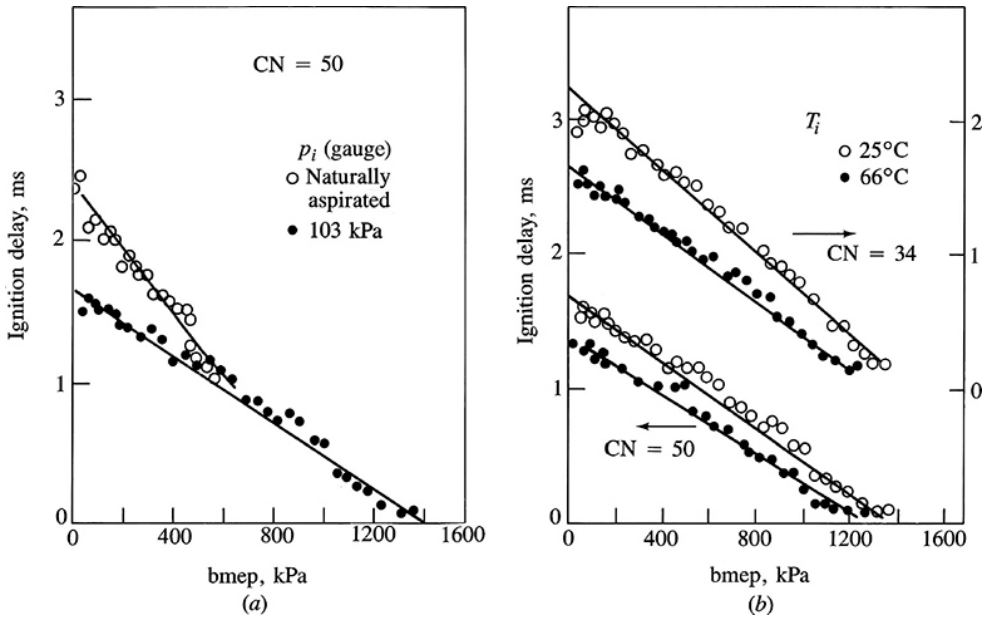


Figure 10.44 Effect of inlet air pressure and temperature on ignition delay versus load in a small DI diesel at 1980 rev/min. (*a*) Engine naturally-aspirated, and turbocharged with 1 atm boost (2 bar absolute); inlet air temperature $T_i = 25°C$; 50 cetane number fuel. (*b*) Engine naturally-aspirated; $T_i = 25$ and 66°C; 34 and 50 cetane number fuel. [40]

Drop Size, Injection Velocity, and Rate

These quantities are determined by injection pressure, injector nozzle hole size, nozzle type, and geometry, but they have little impact on the ignition delay. At normal operating conditions, increasing injection pressure produces only modest decreases in the delay. Doubling the nozzle hole diameter at constant injection pressure to increase the fuel flow rate (by a factor of about 4) and increase the drop size (by about 30%) had no significant effect on the delay. Studies of different nozzle hole geometries showed that the length/diameter ratio of the nozzle was not significant; nor did changes in nozzle type (multi-hole, pintle, pintaux) cause any substantial variation in delay at normal engine conditions. [38]

Intake Air Temperature and Pressure

Figure 10.44 also shows values of ignition delay for diesel fuels for different charge temperatures and pressures at the time of injection. The intake air temperature and pressure will affect the delay via their effect on charge

conditions during the delay period. The figure shows the effects of inlet air pressure and temperature as a function of engine load. The fundamental ignition data available show a strong dependence of ignition delay on charge temperatures below about 1000 K at the time of injection. Above about 1000 K, the data suggest that the charge temperature is no longer as significant. Through this temperature range there is an effect of pressure at the time of injection on ignition delay: the higher the pressure the shorter the delay, with the effect decreasing as charge temperatures increase and delay decreases. Since air temperature and pressure during the delay period are such important variables, other engine variables that affect the relation between the intake air state and the charge state at the time of injection will influence the delay. Thus, an increase in the compression ratio will decrease the ignition delay, and injection timing will affect the delay (as discussed above), largely due to the changes in charge temperature and pressure at the time of injection.

Engine Speed

Increases in engine speed at constant load result in a slight decrease in ignition delay when measured in milliseconds; in terms of crank-angle degrees, the delay increases almost linearly. A change in engine speed changes the temperature/time and pressure/time relationships. Also, as speed increases, injection pressure increases. The peak compression air temperature increases with increasing speed due to smaller heat loss during the compression stroke. [40]

Combustion Chamber Wall Effects

The impingement of the spray on the combustion chamber wall obviously affects the fuel evaporation and mixing processes. Impingement of the fuel jet on the wall occurs, to some extent, in almost all of the smaller, higher speed engines (see Fig. 10.40). Engine and combustion bomb experiments have been carried out to explore the effect of wall impingement on the ignition delay. There does not seem to be a consistent trend. When the evolving fuel spray impinges on the combustion bowl wall, there will be some cooling of the spray tip. However, with short delays (in warmed-up engines) autoignition usually occurs before any significant spray-wall interaction.

Swirl Rate

Changes in swirl rate change the fuel evaporation and fuel-air mixing processes (see Figs. 10.23 and 10.24). They also increase wall heat transfer during compression and, hence, reduce the charge temperature at injection. While engine studies of the effect of swirl rate on ignition delay have been made, only limited data are available. At normal operating engine conditions, the effects of swirl rate changes on the delay are small. Under engine starting conditions (low engine speeds and compression temperatures), the effect is more important, presumably due to the higher rates of evaporation and mixing obtained with swirl.

Oxygen Concentration

The oxygen concentration in the charge into which the fuel is injected would be expected to influence the delay. The oxygen concentration is changed, for example, when exhaust gas is recycled to the intake for the control of oxides of nitrogen emissions (see Sec. 11.2.4). Results of a study carried out in a single-cylinder DI engine operated at a constant air/fuel ratio (30:1), manifold temperature, injection timing, and speed (1800 rev/min), where the oxygen concentration was varied by recirculating known amounts of cooled exhaust, are shown in Fig. 10.45. [41] Oxygen density is normalized by the naturally-aspirated no-recirculation test value. As oxygen concentration is decreased, ignition delay increases.

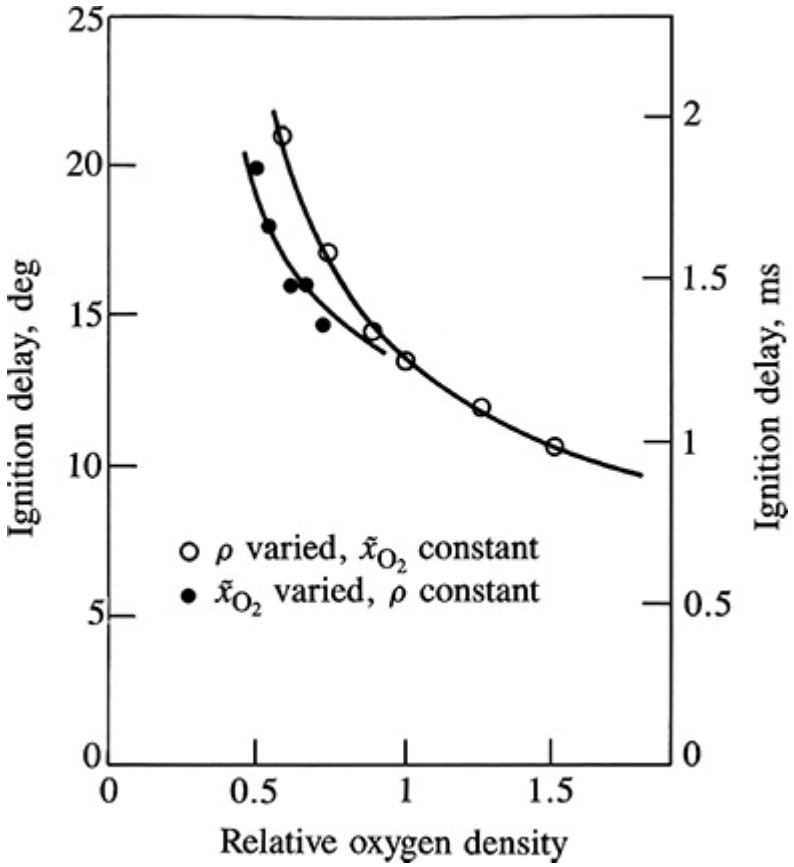


Figure 10.45 Effect of oxygen density in gas on ignition delay in single-cylinder DI engine of 1.3-dm^3 displacement with r_c = 15 at 1800 rev/min. Oxygen density changed by recycling exhaust gas at constant inlet density and by varying inlet pressure from 0.5 to 3 atm with air. [41]

10.5.5 Effect of Fuel Properties

Diesel Fuel

Since both physical and chemical processes take place during the ignition delay, the effects of changes in the physical and chemical properties of fuels on the delay period have been studied. The chemical characteristics of the fuel are much the more important. The ignition quality of the fuel, defined by its cetane number, will obviously affect the delay. The dependence of cetane number on fuel molecular structure is as follows. Straight-chain paraffinic compounds (normal alkanes) have the highest ignition quality (autoignite more quickly), a trend which improves as the chain length increases. Aromatic compounds have poor ignition quality as do the alcohols (hence,

the difficulties associated with using methanol and ethanol, possible alternative fuels, in compression-ignition engines). Figure 10.46 illustrates these effects. A base fuel was blended with pure paraffinic (normal, iso-, and cycloalkanes), aromatic, and olefinic hydrocarbons of various carbon numbers, by up to 20% by volume. The base fuel, a blend of 25% *n*-hexadecane and 75% isooctane, had a cetane number of 38.3. The figure shows that the resulting ignition delays correlate well as a function of cetane number at constant compression ratio and engine operating conditions. Addition of normal alkanes (excluding *n*-pentane and lower carbon number alkanes) improves the ignition quality. As the chain length of the added paraffin gets longer (higher carbon number) the cetane number improvement increases. Isoalkanes, depending on the degree of branching, degrade ignition quality (unless the branching is concentrated at one end of the molecule, when these types of isoalkanes improve ignition quality). Cycloalkanes and aromatics generally reduce the cetane number, unless they have a long *n*-alkane chain attached to the ring. The cetane number of a fuel (a measure of its ability to autoignite) generally varies inversely with its octane number (a measure of its ability to resist autoignition; see Fig. 9.65 for the effect of hydrocarbon structure on knock). The cetane number of commercial diesel fuel is normally in the range of 40 to 55.

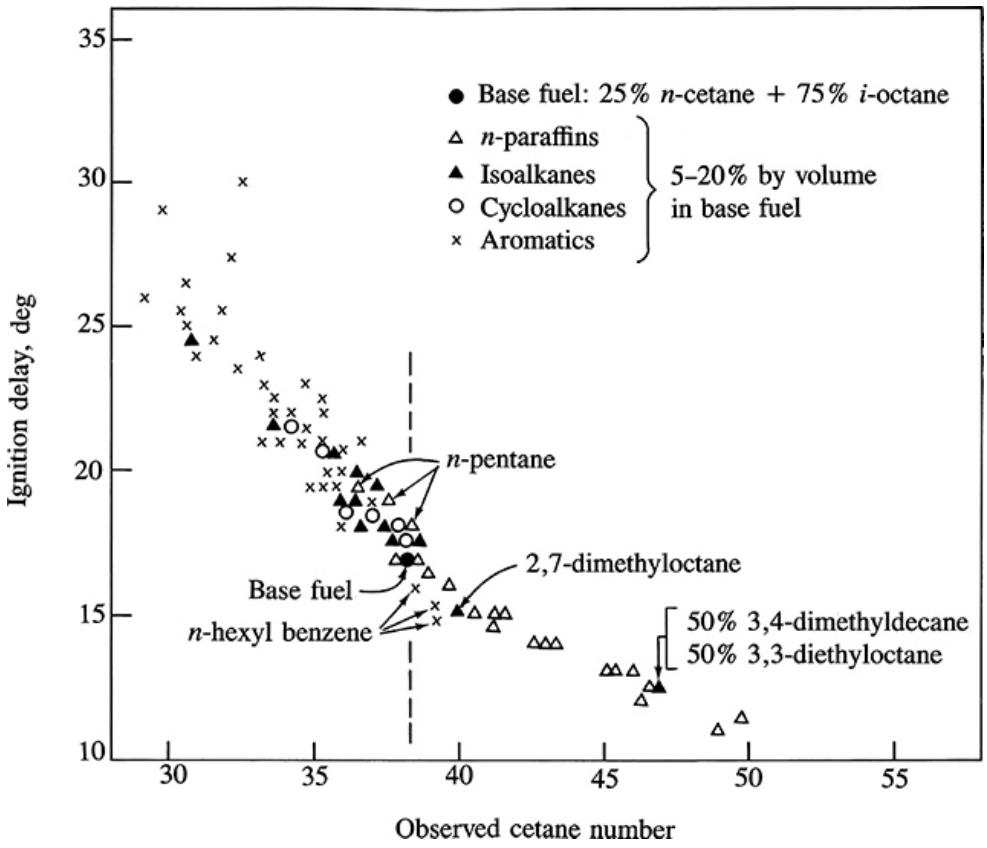


Figure 10.46 Effect of type of hydrocarbon structure on ignition quality of fuels in DI diesel combustion process at constant compression ratio and engine operating conditions.[42]

Cetane number is controlled by the source of crude oil, by the refining process, and by additives or ignition accelerators. Just as it is possible to reduce the tendency to knock or autoignite in spark-ignition engine fuels by adding antiknock agents, so there are additives that improve the ignition quality of compression-ignition engine fuels. Generally, substances that increase the tendency to knock enhance ignition, and vice versa. Ignition-accelerating additives include organic peroxides, nitrates, and nitrites. The most important of these commercially are the alkyl nitrates (isopropyl nitrate, primary amyl nitrates, primary hexyl nitrates, octyl nitrate). Typically, about 0.5% of these additives by volume in a distillate fuel gives about a 10 cetane number increase in a fuel's ignition quality, though their effectiveness may depend on the composition of the base fuel. The incremental effect of increasing amounts of ignition-accelerating additives on cetane number decreases. [40] Usually, the ignition delay obtained with cetane improved

blends are found to be equivalent to those obtained with natural diesel fuels of the same cetane number. Two potential practical uses for ignition accelerators are in upgrading the ignition characteristics of poorer quality diesel fuel and (in much larger amounts) making possible the use of alcohols in compression-ignition engines.

The physical characteristics of diesel fuel do not significantly affect the ignition delay in fully or partially warmed-up engines. Tests with fuels of different front-end volatility (over the cetane number range 38 to 53), and with substantially different front-end ignition quality for the same average cetane number, showed no discernible differences. Fuel viscosity variations over a factor of 2.5 also showed no significant effect. [40] Thus, in a warmed-up engine, variations in fuel atomization, spray penetration, and vaporization rate over reasonable ranges do not appear to influence the duration of the delay period significantly (see also Sec. 10.4.6 on fuel spray evaporation).

Biodiesel

An alternative to petroleum-derived diesel fuel is *biodiesel*. The term refers to a vegetable oil or animal-fat based fuel consisting of long-chain alkyl (methyl or ethyl) esters. [k] These are made by chemically reacting lipids (e.g., vegetable oils, animal fats) with methanol or ethanol. While similar in behavior to diesel fuel, these are differences. The vegetable oils used in biodiesel production differ in the composition of their fatty acids, and differences in the pretreatment of these oils and the biodiesel production process also cause variations in properties.

Biomass sources used to produce vegetable oils for biodiesel production include rapeseed, soybeans, sunflower seed, palm and coconut oils. Highest yields come from palm oil (4500–5000 liters/hectare) with coconut about half that, and the others listed above in the 500 to 1000 l/ha range. Present expectations are that an upper bound on biodiesel productions is about 10% of petroleum diesel.

Biodiesel is currently mostly blended with conventional diesel to minimize the impact of its somewhat different characteristics. Five to ten percent blends (B5 and B10) are commonly used. B100 can be used in diesels but engine modifications would be required.

The characteristics of biodiesel that could affect combustion in diesel engines are its physical and chemical properties. Its molecular size (C_{16}–C

$_{18}$) is in the diesel fuel range (C_{10}–C_{20}), but its boiling point is higher. Its viscosity is higher than that of diesel fuel. It has about 10% lower energy content per unit mass or volume (lower heating value varies with composition but is about 38 MJ/kg). Its cetane number (46 to 60) is higher than that of standard diesel fuel (40 to 55). It has better lubricating properties than diesel due to its higher viscosity. It is, however, a more aggressive solvent, and is less resistant to oxidation so its "shelf life" is significantly more limited. Its use blended in modest amounts with petroleum-based diesel fuel is thus a pragmatic approach. [43, 44]

10.5.6 Correlations for Ignition Delay in Engines

Many correlations have been proposed for predicting ignition delay as a function of engine and air charge variables. These usually have the form of Eq. (10.33) and have been based on data from more fundamental experiments in combustion vessels and flow reactors. An important factor in assessing the appropriateness of any correlation is how it is to be used to predict the magnitude of the delay. If an equation for predicting the complete delay process (including all the physical and chemical processes from injection to combustion) is required, then the data show that such a simple form for the equation is unlikely to be adequate for the full range of engine condition. Much more detailed chemical-kinetic models for the autoignition process of a premixed fuel-air mixture during the delay period are being used in conjunction with models for the physical processes of fuel evaporation and fuel-air mixing to predict the evolving spray processes.

An empirical formula, developed by Hardenberg and Hase [45] for predicting the duration of the ignition delay period in DI engines, has been shown to give reasonable agreement with experimental data over a wide range of engine conditions (see Fig. 10.47). [46] This formula gives the ignition delay (in crank-angle degrees) in terms of charge temperature T (kelvins) and pressure p (bar) during the delay (taken as TC conditions) as

$$\tau_{id}(\mathrm{CA}) = (0.36 + 0.22\overline{S}_p)\exp\left[E_A\left(\frac{1}{\tilde{R}T} - \frac{1}{17{,}190}\right) + \left(\frac{21.2}{p - 12.4}\right)^{0.63}\right] \qquad (10.35)$$

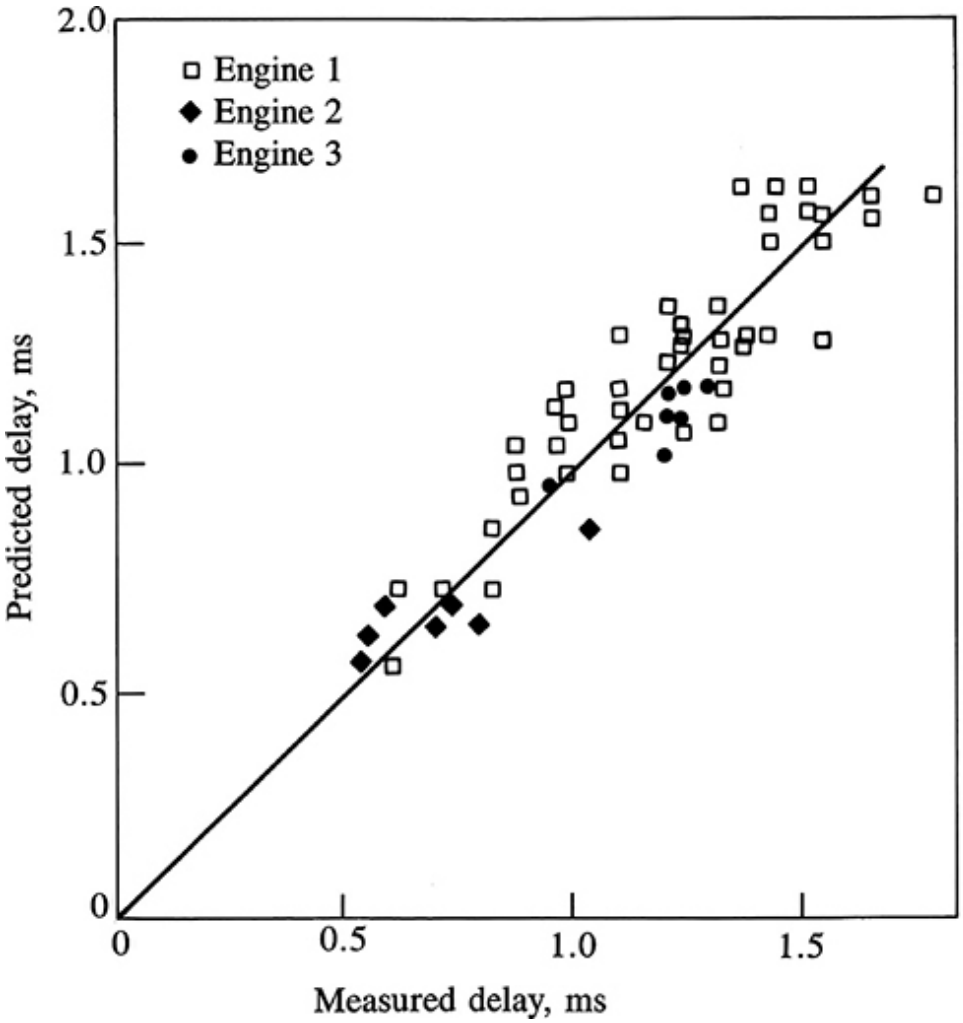


Figure 10.47 Comparison of engine ignition delays predicted with Eq. (10.35) with corresponding measured values.[46]

where $\overline{S}_p$ is the mean piston speed (meters per second) and $\tilde{R}$ is the universal gas constant (8.3143 J/mol · K). E_A (joules per mole) is the apparent activation energy, and is given by

$$E_A = \frac{618{,}840}{\mathrm{CN} + 25} \tag{10.36}$$

where CN is the fuel cetane number. The apparent activation energy decreases with increasing cetane number. The delay in milliseconds is given by

$$\tau_{id}(\text{ms}) = \frac{\tau_{id}(\text{CA})}{0.006N}$$

where N, engine speed, is in revolutions per minute. Values for T and p can be estimated using a polytropic model for the compression process:

$$T_{TC} = T_i r_c^{n-1}, \; p_{TC} = p_i r_c^n \qquad (10.37a,b)$$

where n is the polytropic exponent, r_c is the compression ratio, and the subscript i denotes intake manifold conditions. Values of the polytropic exponent are given in Fig. 10.48 for a direct-injection diesel under warm and cold engine operating conditions. [41, 46]

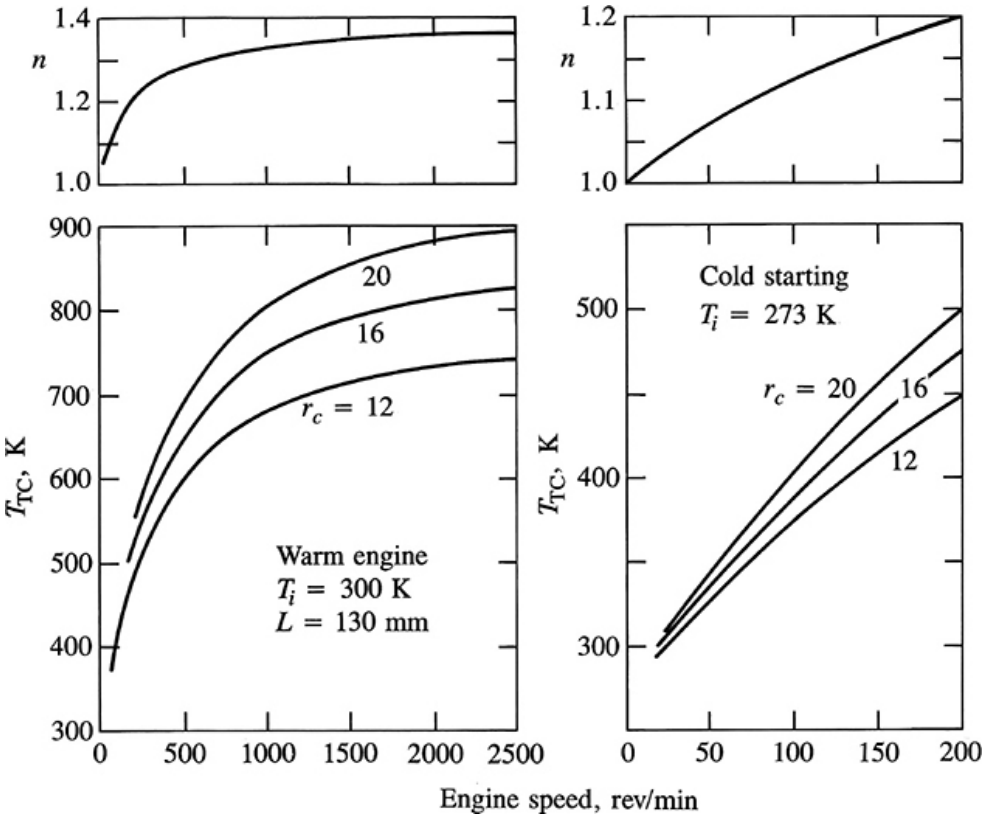


Figure 10.48 Exponent n for polytropic model of compression process in Eq. (10.37) and corresponding end-of-compression air temperature at TC. Warm and cold DI diesel engine with 130-mm stroke. [45]

10.6 MIXING-CONTROLLED COMBUSTION

10.6.1 Background

Earlier sections of this chapter have described our current understanding of the individual processes, which together make up the total fuel injection-mixing-burning sequence—atomization, vaporization, spray development, air entrainment, ignition, and combustion. The photos in Fig. 10.5 illustrate details of the diesel combustion process, and the schematics in Fig. 10.11 identify the several steps involved. The interaction mixing-controlled phase of combustion is the area of focus of this section. The photographs of the compression-ignition combustion process in different types of diesel engines in Figs. 10.4 and 10.5 show how the diesel diffusion flame forms, following ignition and start of the premixed-burned phase, spreads rapidly, and envelops the spray. Depending on the spray configuration, the visible flame may then fill almost the entire combustion chamber. The diffusion flame and spray geometries are closely related. Mixing processes are also critical during the ignition delay period: while the duration of the delay period is not influenced in a major way by the rates of the spray processes which together control "mixing," the amount of fuel mixed with air to within combustible limits during the delay (which affects the rate of pressure rise once ignition has occurred) is obviously directly related to mixing rates.

However, while it is well accepted that diesel combustion is normally controlled by the fuel-air mixing rate, the mechanisms by which the diffusion flame forms, spreads rapidly, and then stabilizes are not well understood. The difficulties are twofold. First, the spray geometry in real diesel combustion systems is extremely complex. Second, the phenomena, which must be described (especially the development of the unsteady turbulent diffusion flame) are inadequately understood.

Current capabilities for modeling these phenomena are reviewed in Chap. 14 . Thermodynamic-based models of the diesel combustion process with atomization, vaporization, and spray development described by empirical or phenomenological turbulent-jet-based submodels have been developed and used to predict burning rates. These are described in Secs. 14.4.3 and 14.4.4 and show reasonable but not precise agreement with data. Fluid-mechanic-based models of airflow, fuel spray behavior, and combustion have been and

are still being developed (see Sec. 14.5). While realistic air-flow predictions are now feasible, predictions of spray behavior are less well developed and combustion-rate predictions are still exploratory.

In the following sections, the evidence linking spray characteristics to flame structure and fuel burning rates is summarized.

10.6.2 Spray and Flame Structure

The structure of each fuel spray is that of a narrow liquid-containing core (densely filled with drops of order 20 μm in diameter) surrounded by a much larger gaseous-jet region containing fuel vapor (see Fig. 10.21). The fuel concentration in the core is extremely high: local fuel/air equivalence ratios near the nozzle of order 10 have been measured during the injection period. Fuel concentrations within the spray decrease with increasing radial and axial position at any given time, and with time at a fixed location once injection has ended. The fuel distribution within the spray is controlled largely by turbulent-jet mixing processes. Fuel vapor concentration contours determined from interferometric studies of unsteady vaporizing diesel-like sprays, confirm this gaseous turbulent-jet-like structure of the spray, with its central liquid-containing core which evaporates relatively quickly once fuel injection ends. [47]

Figure 10.42 shows how the autoigniting of each spray and the subsequent development of the diffusion flame around the edge of the penetrating spray occur. Figure 10.49 shows how this process continues under conditions typical of DI quiescent-chamber diesel engines. It shows tracings of the liquid spray core and the initiation and spread of the diffusion flame boundaries, taken from high-speed movies of the injection and combustion processes with central injection of five fuel jets into a disc-shaped chamber in a rapid-compression machine. [48] Such studies show that diffusion flame first forms around the edge of the spray (which may by then have interacted with the combustion chamber walls). This diffusion flame then moves rapidly back along the spray boundary toward the injector nozzle. As time progresses, combustion products diffuse outward (and inward), from the diffusion flame enveloping the spray, as indicated in Fig. 10.49. This outward diffusion of (hot) burned gas is apparent in the combined natural

light emission and shadowgraph image of a single diesel fuel spray in Fig. 10.5.

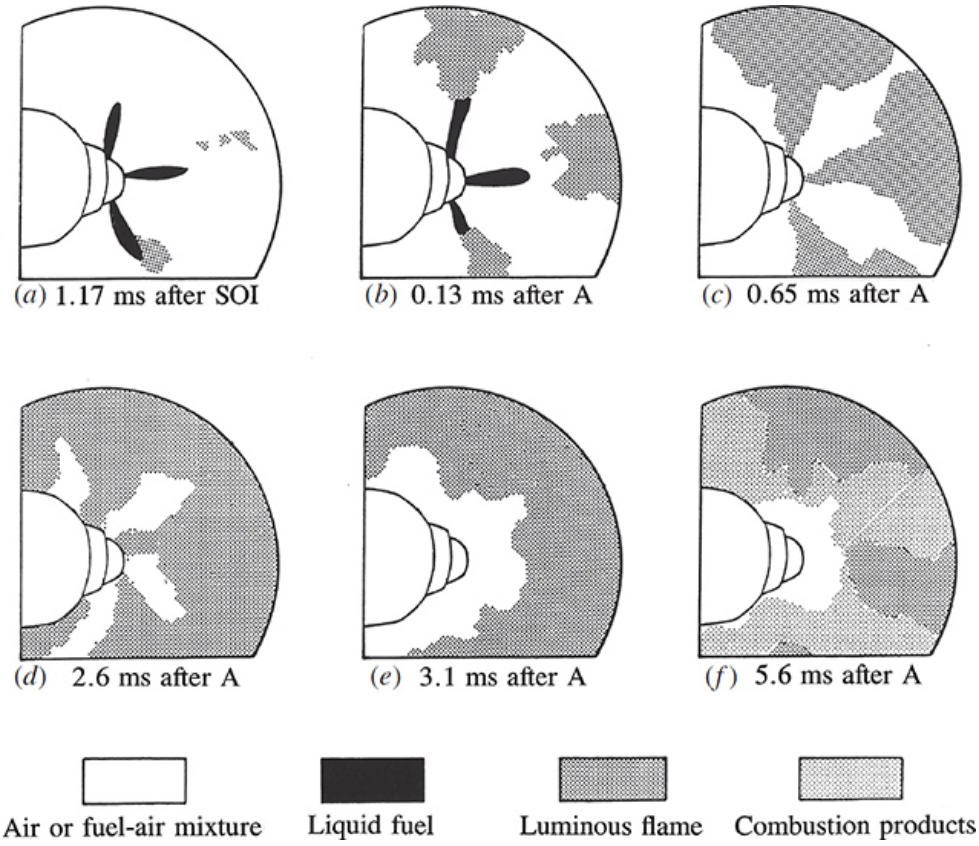


Figure 10.49 Tracings of the boundary of liquid fuel core in the spray and flame, from high-speed movies of diesel combustion taken in a rapid-compression machine, looking down on piston through transparent head. First occurrence of luminous flame at A, 1.17 ms after start of injection. End of injection at D. [48]

With modern high-injection-pressure (1000 bar plus) fuel systems, which produce high-velocity (of order 500 m/s) fuel drops, the flame does not move all the way back to the injector. It stabilizes during the quasi-steady phase of the spray's development some distance away from the nozzle exit. This phenomenon is called *flame lift-off*: it is illustrated in Fig. 10.13. As shown, its importance is that substantial entrainment of hot air into the spray occurs along this lift-off length. [1] This significantly affects the fuel-air equivalence ratio within the spray, which impacts the mixture temperature and its autoignition chemistry. For moderate loads, in a quiescent DI diesel engine,

some 20% of the air needed for complete combustion of the fuel is entrained into the spray over this lift-off distance.

Thus, the initiation of the diffusion flame and its spread around the developing spray, the essential component of the mixing-controlled combustion phase in diesel engines, occurs as follows. As the premixed-burn combustion phase progresses, the fuel-rich mixture within the leading portion of the spray becomes hotter through its chemical energy release, and due to the rising pressure within the cylinder (which also increases the air temperature). Within one or a few crank-angle degrees, one or more (localized) diffusion flames form with their energy release further increasing gas temperatures. These initial diffusion flames spread rapidly around the outer boundaries of the turbulent sprays where the fuel-air mixture is slightly rich of stoichiometric where reaction rates are fastest. The flame moves back toward the injector nozzle orifice to the location where the local jet velocity is high enough that its convective resistance to flame spreading stabilizes the diffusion flame, a short distance (the flame lift-off distance, typically some 20 mm) from the jet or spray's origin.

Figure 10.50 shows a time-averaged image of a burning diesel fuel spray in an ambient environment at 1000 K with gas density of 14.8 k/m^3, injection pressure of 138 MPa, and orifice diameter of 180 μm. The flame lift-off length, indicated by the white bar, was 22 mm. The experiments were done in a closed vessel. [22] The lower part of the figure shows how this lift-off length varies with ambient-gas temperature and density at the conditions listed above (with 21% oxygen in the ambient gas).

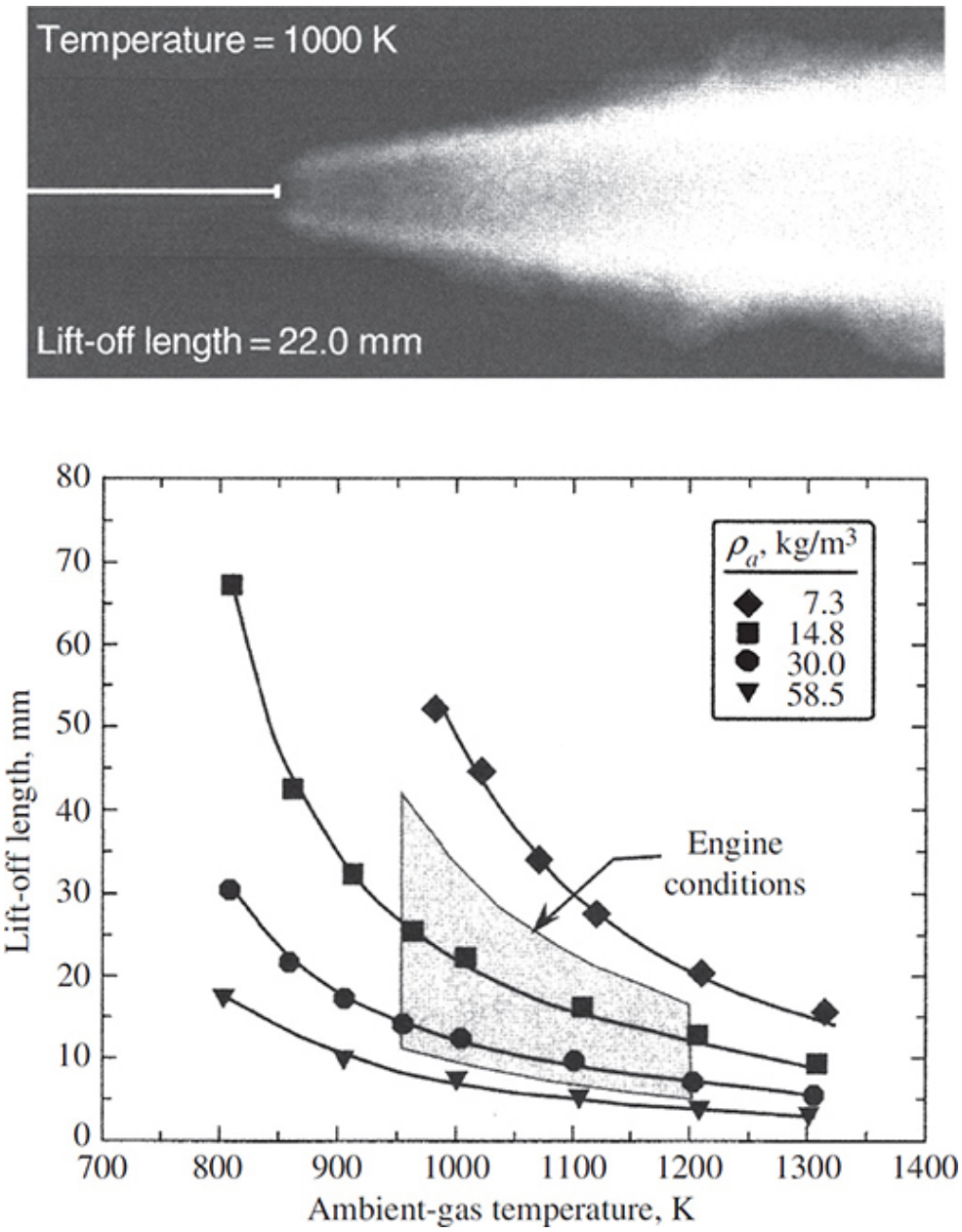


Figure 10.50 Example of time-averaged image of a burning diesel fuel spray under diesel TC-of-compression conditions showing diffusion flame lift-off-distance (22 mm). See text for details of the conditions, in quiescent combustion vessel (upper figure). Lower figure, lift-off lengths as a function of ambient gas conditions (temperature and density). Oxygen concentration in ambient gas 21%, pressure drop across orifice 138 MPa, orifice diameter 180 μm. Grey region indicates typical conditions for a quiescent DI diesel. 22, 49

The changes in lift-off length with respect to these and other parameters cause changes in the air entrained into each spray. The quantity of entrained air prior to lift-off can be estimated. This amount can be expressed as a percentage of the total air required to fully burn the fuel being injected: the stoichiometric air requirement. This amount defines how rich the mixture is that undergoes the autoignition and premixed-burn first phase of diesel combustion. That region is shown in the annotated schematic of an individual spray in Fig. 10.13: it lies downstream of the rich unburned-mixture region, which is just downstream of flame lift-off, and ahead of the products of rich combustion region, which is further downstream. This percentage, ξ_{st} is given by:

$$\xi_{st}(\%) = \frac{100}{\phi} = 100\left[\frac{(1+16\tilde{L}_{LO}^{2})^{1/2}-1}{2(F/A)_{s}}\right] \tag{10.38}$$

where $\tilde{L}_{LO}$ is the lift-off length/ x^{+}, and x^{+} is the characteristic length scale of the fuel spray defined by Eq. (10.19b) .[22]

The variables that affect the diffusion flame lift-off length are: ambient gas temperature and density, oxygen concentration in the ambient gas (air or air plus EGR), orifice diameter, orifice pressure drop, local fuel-air equivalence ratio. The stable lift-off location downstream of the nozzle orifice is a balance between the downstream (turbulent) flow in the outer regions of the spray, and the upstream diffusion-flame area expansion speed. This balances the downstream spray momentum convecting the flame away from the nozzle against the upstream flame propagation, which depends on the local rate of chemical energy release. Effectively, the spreading quenches at the lift-off distance. With an easier-to-quench mixture state, flame lift-off is farther from the injector nozzle: with a harder-to-quench mixture state, the flame stabilizes closer to the nozzle. [22, 49]

This flame lift-off length discussion when connected with the earlier discussion of the liquid containing core of the spray (quantified by its liquid length, Sec. 10.4.6), and the premixed- and diffusion-phases of combustion, suggests that changes in the relative spatial relationship between air entrainment, fuel vaporization, and combustion zones can occur and substantially change conditions within the spray. Figure 10.51 shows two extremes: on the left the liquid length is substantially longer than the (short)

flame lift-off length; and on the right the reverse occurs with a short liquid length and long lift-off distance. On the right, the amount of (hot) air entrained is almost one-third of the stoichiometric air requirement, the premixed rapid combustion process will be less rich (a ϕ of order 3), and the amount of soot formed subsequently will be less (see Fig. 10.56). On the left, the entrained air is less than 10%: ϕ at the lift-off point will be very rich (ϕ ~ 10), and would be expected to become less-rich moving downstream through the elongated premixed-combustion region as additional air is entrained. Here the increased richness would likely increase the soot yield significantly.

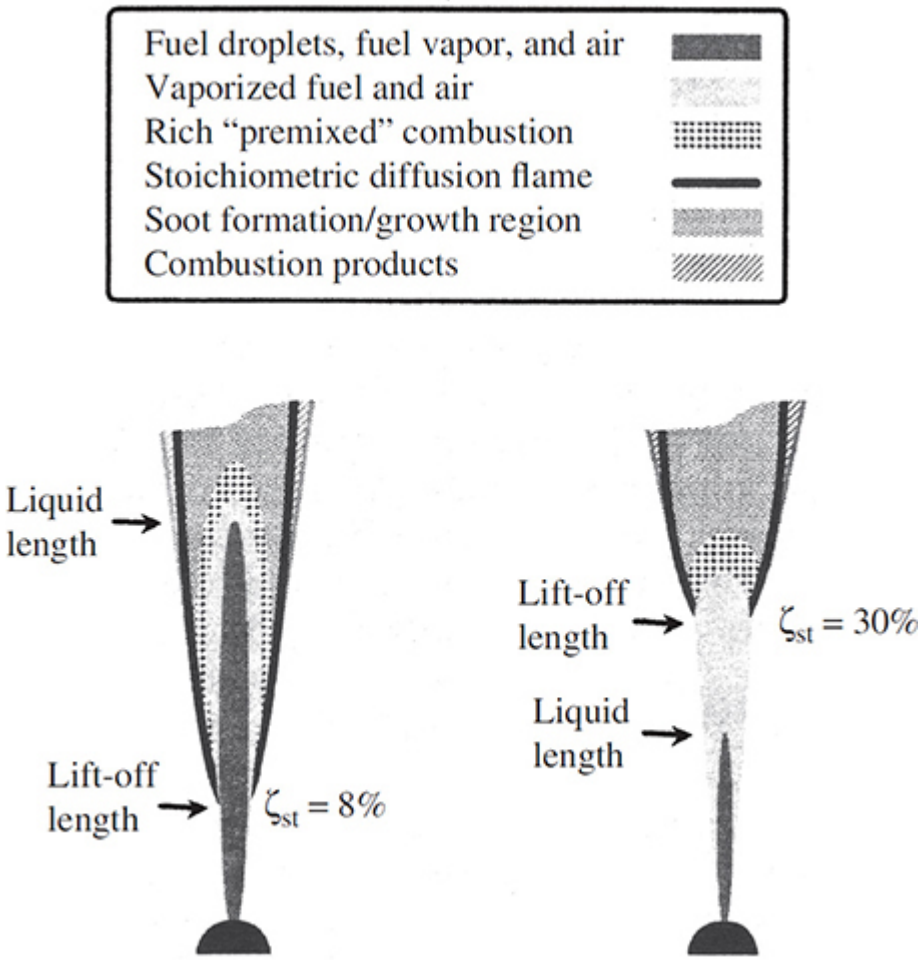


Figure 10.51 Schematics showing how the relative spatial relationship between fuel vaporization and the location of the DI diesel combustion zones may vary. See discussion in text. On the left, ambient gas conditions are 1100 K and 23 kg/m ^{3}, the orifice pressure drop is 40 MPa, and orifice diameter is

250 μm. On the right, ambient gas conditions are 1000 K and 20 kg/m ^{3}, orifice pressure drop is 200 MPa, and orifice diameter is 100 μm. [49]

The role of air swirl in promoting more rapid fuel-air mixing in medium-size and smaller diesel engines is evident from gas-sampling studies in engines with these different combustion systems. The variation of gas species and unburned hydrocarbon concentrations within critical regions of the combustion chamber have been mapped out, and these data show that during the earlier stages of injection and combustion, the boundaries of the individual sprays can be identified as they are convected around the combustion chamber bowl by the swirl. The fuel distribution within the combustion chamber is highly nonuniform. However, additional air has mixed into the spray due to swirl to bring the fuel/air equivalence ratios within the spray, in the outer regions of the chamber, closer to stoichiometric values. [50] This substantially different character of the spray with swirl is clear from the data in Fig. 10.52. While the local ϕ values obviously depend on the relation of the sample valve location to spray position at any given crank angle, the much lower values of moles CO per mole $CH_{1.75}$ (and thus of equivalence ratio) with swirl relative to values in a quiescent chambers, during injection and the early stages of combustion, clearly indicate that swirl enhances fuel-air mixing rates substantially. As combustion ends, the fuel distribution within the combustion chamber, at least on a gross geometric scale, becomes more uniform. However, early in the combustion process, the high CO levels found in these different diesel combustion systems, as shown in Fig. 10.52, indicate that the burned gases are only partially reacted. With quiescent chambers this is largely due to lack of oxygen. With swirl, however, significant oxygen is present; whether the high CO with swirl is due to kinetic limitations or to smaller-scale mixture non-homogeneities is unclear.

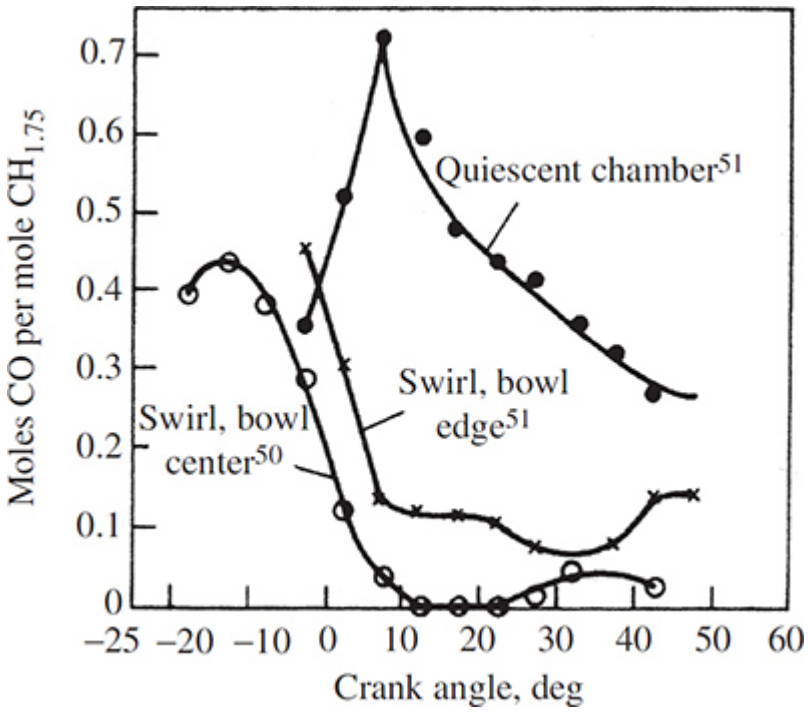


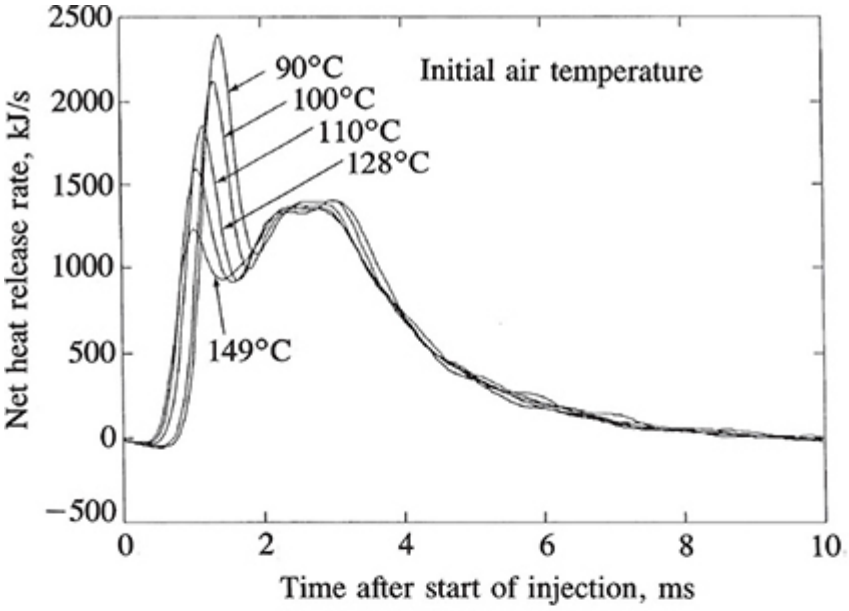
Figure 10.52 Time-resolved gas-composition data obtained from rapid-acting sampling valves from within the combustion chambers of quiescent and high-swirl bowl-in-piston DI diesel engines. CO concentrations (in moles CO per mole fuel, $CH_{1.75}$) are shown: (1) on the spray centerline with larger quiescent chamber diesel engine; (2) at edge of deep bowl, and within shallow bowl, in engines with swirl. [50, 51]

10.6.3 Fuel-Air Mixing and Burning Rates

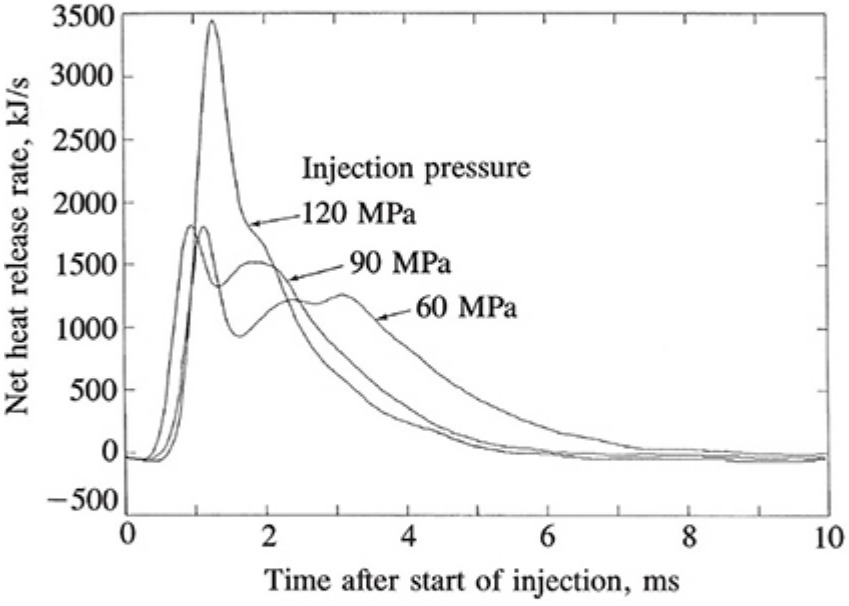
The energy-release profile during diesel combustion obtained from heat-release analyses of cylinder pressure data identifies two main stages of combustion (see Fig. 10.8). The first is the premixed-combustion phase, when the fuel which has mixed sufficiently with air within the spray to form an ignitable fuel-rich mixture during the delay period, releases part of its chemical energy. The second is the mixing-controlled combustion phase, where a diffusion flame surrounds the spray and completes the fuel oxidation process. Experimental evidence from heat-release analysis indicates that the majority of the fuel (usually more than 80%) burns during this second mixing-controlled phase.

That the fuel-burning or heat-release rate during the main diffusion flame combustion process is predominantly mixing controlled is supported by many types of evidence. Estimates of the rate at which fuel-air mixture with composition within the combustible limits is produced in diesel sprays under typical engine conditions, based on a variety of turbulent-jet models of the spray, show that mixing rates and burning rates are comparable in magnitude. Estimates of characteristic times for the turbulent-jet mixing processes in diesel combustion chambers show these to be comparable to the duration of the heat-release process, and much longer than characteristic times for fuel evaporation and the thin diffusion flame chemical kinetics.

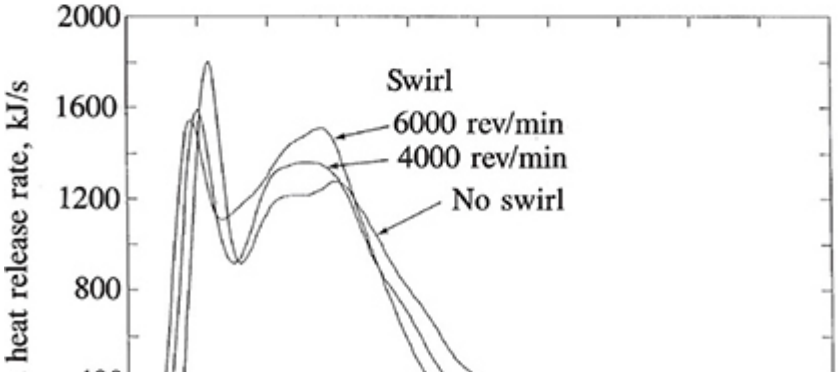
Then, measured diesel heat-release profiles during this main part of combustion show trends with engine design and operating parameter changes that correspond to fuel-air mixing being the primary controlling factor. Examples of heat-release profiles measured in rapid-compression-machine studies of diesel combustion, shown in Fig. 10.52, illustrate this clearly. The rapid-compression machine had a disc-shaped chamber of 10 cm diameter with a 3.1-cm clearance height at the end of a compression process through a volume ratio of 15.4; a five-hole centrally located fuel-injector nozzle was used. Figure 10.53 *a* shows the heat-release profiles for different initial (prior to compression) air temperatures, which produce different ignition delays. Longer delays allow more fuel to mix to within combustible limits during the delay, so the peak premixed heat-release rate increases. However, the mixing-controlled-phase heat-release-rate magnitudes are essentially the same because the spray-mixing processes are little affected by these changes in air temperature. Figures 10.53 *b* and *c* show that heat-release rates throughout the combustion process are increased by increased fuel-injection rate (achieved by increasing the fuel-injection pressure) and by swirl. Both these changes increase the fuel-air mixing rates within the fuel spray and therefore increase the heat-release rate during the mixing-controlled combustion phase.


Net heat release rate, kJ/s
2500
2000
1500
1000
500
0
−500
0
2
4
6
8
10
Time after start of injection, ms
Initial air temperature
90°C
100°C
110°C
128°C
149°C

(*a*)


Net heat release rate, kJ/s
3500
3000
2500
2000
1500
1000
500
0
−500
0
2
4
6
8
10
Time after start of injection, ms
Injection pressure
120 MPa
90 MPa
60 MPa

(*b*)


heat release rate, kJ/s
2000
1600
1200
800
Swirl
6000 rev/min
4000 rev/min
No swirl

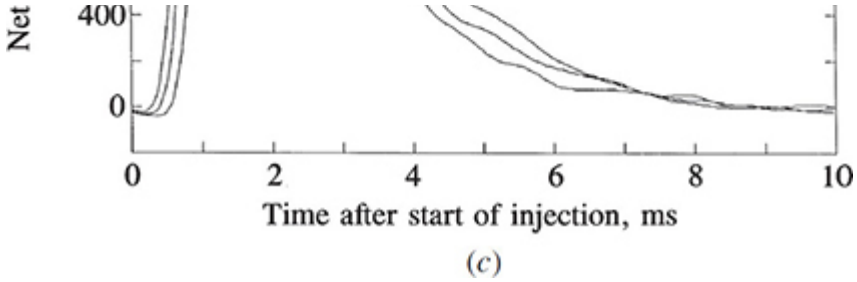


Figure 10.53 Net heat-release rates, as a function of time after start of injection, calculated from cylinder pressure data from rapid-compression-machine studies of DI diesel combustion. (*a*) Effect of varying initial air temperature: 4000 rev/min swirl, injection pressure 60 MPa, 100°C initial temperature corresponds, via Eq. (10.37 *a*) , to about 900 K. (*b*) Effect of varying injection pressure: no swirl. (*c*) Effect of varying swirl: injection pressure 60 MPa. [14]

Diesel engine heat-release-rate trends, as design and operating variables are changed, can be related to mixing rates in analogous fashion. Table 10.4 summarizes the trends that have been investigated. The directional effects of changes in engine parameters on the ignition delay period and the fuel-air mixing rate are all consistent with the measured changes in premixed and mixing-controlled heat-release rates. The controlling role of fuel-air mixing in the diesel engine fuel spray on combustion is clear.

TABLE 10.4 Effects of engine design and operating variables on heat-release rates [1]

Parameter varied	Effect on			
	τ_{id}	$\dot{m}_m$	$\dot{Q}_p$	$\dot{Q}_m$
Injection rate ↑	0	↑	↑	↑
Turbocharger boost ↑	↓	0	↓	0
Compression ratio ↓	↑	0	↑	0
Number of injector holes ↑	0	↑	↑	↑
Injection advance ↑	↑	0	↑	0
Swirl ↑	0	↑	↑	↑
Intake-air temperature ↓	↑	0	↑	0
Injection pressure ↑	0	↑	↑	↑
Speed ↑	0	↑	↑	↑

τ_{id}, ignition delay; $\dot{m}_m = (dm/dt)_m$, fuel-air mixing rate; $\dot{Q}_p = (dQ/dt)_p$, heat-release rate during premixed-combustion phase; $\dot{Q}_m = (dQ/dt)_m$, heat-release rate during mixing-controlled combustion phase; ↑ increase; ↓ decrease; 0 minor effect.
Source: From Plee and Ahmad.[39]

10.7 ALTERNATIVE COMPRESSION-IGNITION COMBUSTION APPROACHES

Modern diesel engine injection equipment permits multiple fuel injection pulses and control of the fuel injection rate during each pulse. So the basic diesel combustion process of a fuel-rich "premixed flame" within each individual fuel spray, followed by the burning to completion of this rich mixture in a diffusion flame around each spray, can be modified.

The various approaches of interest can be categorized into two broad groups: multiple fuel injections intended to improve the basic diesel combustion process, and alternative combustion concepts which are significantly different from the standard mode. Table 10.5 provides a framework for categorizing these alternative compression-ignition combustion strategies, and a summary of their key features and operating characteristics. Additional discussion of diesel combustion with multiple fuel injections, and of what are often called *advanced combustion concepts* where the fuel-air mixing and burning processes differ significantly from the normal diesel engine processes, follows in Secs. 10.7.1 and 10.7.2, respectively.

TABLE 10.5 Summary of alternative compression-ignition combustion and injection strategies

Feature	Diesel, with multiple injections	Advanced combustion	
		Premixed	Late
Primary objective:	Reduce the noise, emissions of standard diesel combustion	Achieve low burned gas temperatures (with attendant emissions reduction benefits)	
Approach:	Pilot injection to improve main diesel combustion process. Additional injections to control emissions (via exhaust treatment devices)	Early fuel injection, lean or dilute mixture, long induction period, premixed fuel-air mixture	Late fuel injection, high EGR levels and mixing rates to achieve faster, after TC, combustion process
Key features:	Common rail high-pressure injection system with piezoelectric injector	Early injection (intake or early compression), well- or pre-mixed fuel-air mixture, spontaneous ignition after long induction period	Close to TC injection, high levels of cooled EGR, high swirl, fast fuel-air mixing, moderate ignition delay
Combustion characteristics:	More flexible and controlled diesel combustion (premixed burning and diffusion flame)	Lean and/or dilute premixed autoignition process with low burned-gas temperatures	Dilute, fast-burning; spontaneously ignited, premixed and mixing-controlled burning processes, early in the expansion stroke
Efficiency improvement potential:	Moderate	High	Moderate
Emissions levels:	Improved PM and NO_x compared to standard diesel	Lowest PM and NO_x emissions	Lower NO_x, PM improved, compared to standard diesel
Fuels:	Standard diesel fuel	Needs low cetane number fuel	Standard diesel fuel

As Table 10.5 indicates, use of multiple injections has several objectives. The different injection pulses can modify the in-cylinder gas state to reduce the ignition delay of the primary main combustion process (through use of pilot injection), enhance the burning rate of the latter portion of the main injected fuel pulse (split injection), or reduce the in-cylinder particulate loading (post injection). As use of diesel exhaust after treatment devices to oxidize hydrocarbons, reduce nitrogen oxides, and filter and burn up particulates expands, late fuel injection pulses are being used to release additional fuel chemical energy at appropriate times in the engine cycle to raise gas temperatures locally, in the bulk charge, or in the exhaust. [52] Figure 10.54 illustrates these injection strategies: it also indicates the relative magnitude (proportion) of these several fuel injection pulses.

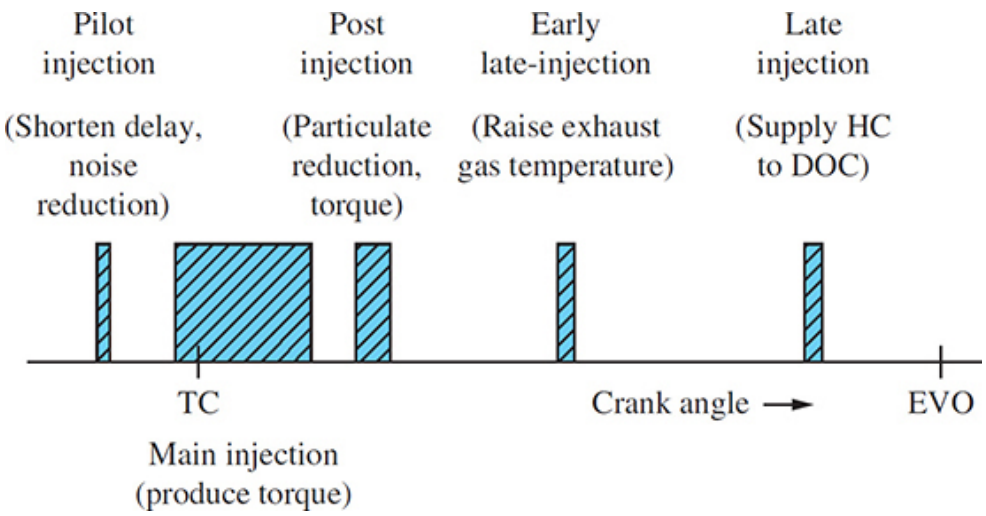


Figure 10.54 Schematic of multiple fuel injection strategy options identifying the objectives of the different injection pulses. DOC, diesel oxidation catalyst. [52]

Table 10.5 divides the advanced combustion strategies into two classes with substantive differences in basic approach: (1) early injection of (some) fuel so the fuel-air mixture burned is closer to being *premixed* and (2) where injection is retarded to achieve *late* (and usually fast) combustion, largely to reduce in-cylinder pollutant formation.

10.7.1 Multiple-Injection Diesel Combustion

Figure 10.54, as briefly discussed in the preceding section, illustrates several different multiple fuel-injection strategies in relation to the main fuel injection pulse, which normally occurs around TC. The primary role of each injection event is indicated as is their relative injection duration.

Pilot Injection

As mentioned, pilot injection is increasingly used to shorten the ignition delay between start of injection, and start of combustion of the main injection of the standard diesel injection strategy. Its objective is primarily to reduce engine noise (see Sec. 15.7.1). The combustion of the pilot injection fuel with its entrained air raises the temperature of the mixture around the injector, thus reducing the ignition delay of the main-injection fuel. The shorter the delay, the less rapid the initial burning rate of the main fuel pulse.

Because there is less time to "prepare" the early injected portion of the main injection for rapid combustion, the initial combustion-generated rate of pressure rise is reduced resulting in lower pressure-rise-rate generated noise levels. This pilot-injection effect is most advantageous at the lighter engine loads and prior to the engine fully warming up. At higher engine loads and speeds, the in-cylinder gases at time of injection are hotter so the ignition delays are inherently shorter so the impact of pilot injection is significantly reduced.

A second effect of pilot injection is that combustion of the pilot fuel adds a diluent—burned gases—to the "air" surrounding the injector. Thus, the flame temperatures in the diffusion flame surrounding each main-injection fuel spray are reduced, because the oxygen combusted in these diffusion flames has this additional dilution beyond the nitrogen in the air and the burned gas residual from the previous cycle. This reduces the NO_x formation rate in the diffusion flame (see Sec. 11.2.4): it may also reduce the soot oxidation rate in the diffusion flame and adversely affect the particulate emissions (Sec. 11.5.5).

Typically, pilot fuel injection amounts are a few percent (5 to 15%) of the total fuel injected, depending on load, and its injection is started some 15 to 35 degrees CA before TC. Pilot fuel injection results in significant reductions in the ignition delay—by a factor of about two.

Post Injection

Post or after injection primarily impacts particulate emissions. Post injection with a relatively short delay (some 10 to 15° CA) between end of main injection and start of post injection pulses usefully reduces soot emissions (up to 40%). Particulate emissions are usually increased with post injections closer to the end of main injection, and with significantly later post injections. Use of post injections is primarily a strategy for reducing particulate emissions at lighter loads and lower engine speeds.

The late injection pulses illustrated in Fig. 10.54 are used to enable the performance of exhaust system aftertreatment devices. Fuel injection (and its combustion) part-way through the expansion stroke raises the exhaust gas temperature and is used to initiate the particulate burn-up or regeneration process in particulate traps/filters. Injection late in the expansion stroke is used when a reducing agent (hydrocarbons) is needed to reduce NO_x in a

lean exhaust gas in a *selective catalytic reduction* (SCR) system, which is used for NO_x removal (see Sec. 11.6.3).

10.7.2 Advanced Compression-Ignition Combustion Concepts

Spontaneous fuel-air mixture ignition (compression ignition) can be used to initiate engine combustion using a variety of fuels, and fuel injection and mixing strategies. These alternative approaches are being explored for their high efficiency, and low particulate and oxides of nitrogen emissions potential. In the late 1970s, homogeneous charge two-stroke cycle engines were successfully operated with compression ignition. [53] Najt and Foster [54] analyzed homogeneous-charge compression ignition (HCCI) fundamentals, concluding that the ignition process is controlled by the low-temperature hydrocarbon oxidation kinetics while the main energy releasing process is controlled by the high-temperature hydrocarbon oxidation kinetics. See Sec. 9.6.2 for a more detailed discussion of the autoignition chemistry of hydrocarbon-air mixtures.

The low temperature oxidation reactions in engines start to occur at about 800 K. Intermediate species such as aldehydes are produced through abstraction of H-atoms from the fuel molecules. These peroxides react with OH to produce a growing number of active radicals, converting chemical energy into thermal energy at an accelerating rate. The more direct oxidation path then commences at temperatures between 950 and 1000 K, and intermediates such as olefins and alkyl radicals are increasingly rapidly consumed. Depending on the relative rates of the low- and high-temperature paths, which are governed by the fuel's composition and the molecular structure of the individual hydrocarbons involved and mixture temperature, a two-stage ignition process may occur (when the low-temperature oxidation path is slower). Or, the two processes may effectively merge into one continuous process. [55]

In 1984, Oppenheim [57] (see also Sec. 10.5.3) proposed the concept of HCCI where the unburned mixture conditions necessary for compression ignition in engines were achieved by controlling temperature (and dilution) by varying the amounts of residual gases and EGR. It also became clear that

the fuel's autoignition characteristics were a critical factor in this controlled autoignition combustion process.

Figure 10.55 illustrates the basic features of this HCCI combustion process, by comparing a conventional diesel combustion process with the HCCI process. The HCCI combustion process occurs in a (fairly) well-mixed lean charge achieved by early injection, in this case of a fuel with a long chemical induction time (ignition delay). The smooth two-stage heat-release-rate profile of HCCI is clear, and the HCCI profiles contrast with the conventional diesel heat-release rate profile of a rapid/premixed initial burning phase followed by the mixing-controlled diffusion-flame phase.

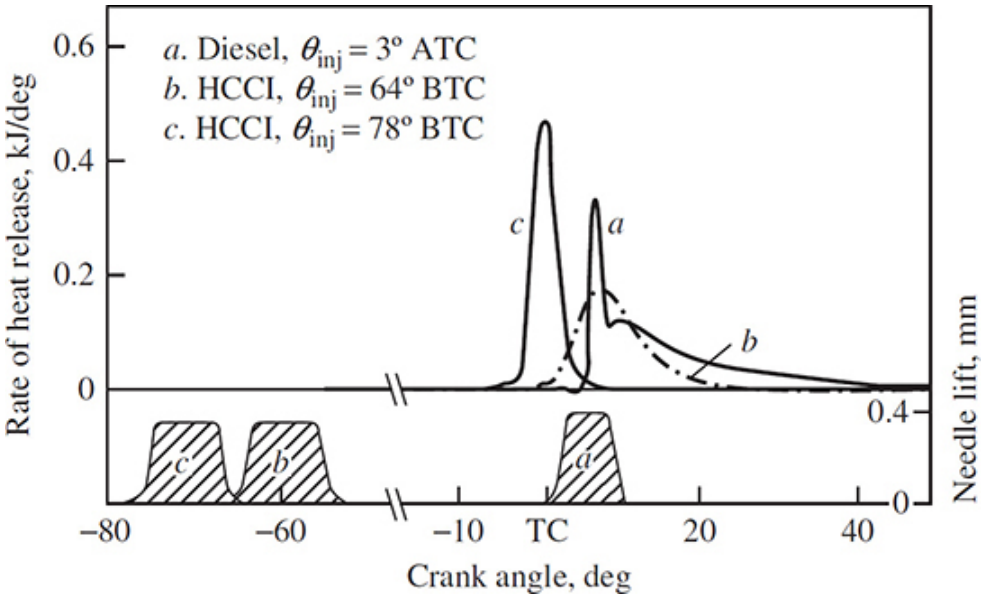


Figure 10.55 Comparison of (*a*) the heat-release rate profile of a standard diesel combustion process with start of injection (SOI) 3° ATC and start of combustion (SOC) a few degrees later, with two HCCI engine cases: (*b*) HCCI heat-release rate profile with SOI at 64° BTC and SOC shortly after TC; (*c*) with SOI at 78° BTC and SOC 4° BTC.[56]

The advantages of spontaneously igniting a homogeneous lean and/or dilute mixture are the low burned gas temperatures that result which avoid the fuel-air equivalence ratio (ϕ) and temperature (T) regimes where soot formation and where NO formation occur. As shown in Fig. 10.56, the cylinder gas temperatures (average in-cylinder values) for the HCCI combustion are below about 1600 K, and are relatively uniform. With

conventional diesel combustion, in-cylinder gas temperatures are significantly higher (and within the diffusion flame that surrounds each fuel spray, see Fig. 10.13, reach about 2700 K), thus forming significant soot within the rich reacting spray as well as NO in the close-to-stoichiometric diffusion flame. The NO_x emissions reduction potential of the HCCI type of combustion process is illustrated by the relative engine-out NO_x emission levels of the different combustion process examples shown in Fig. 10.55. The base diesel case (*a*) had NO_x levels of 400 ppm: the HCCI cases (*b*) and (*c*) had levels of 250 ppm, and 20 ppm, respectively. [56]

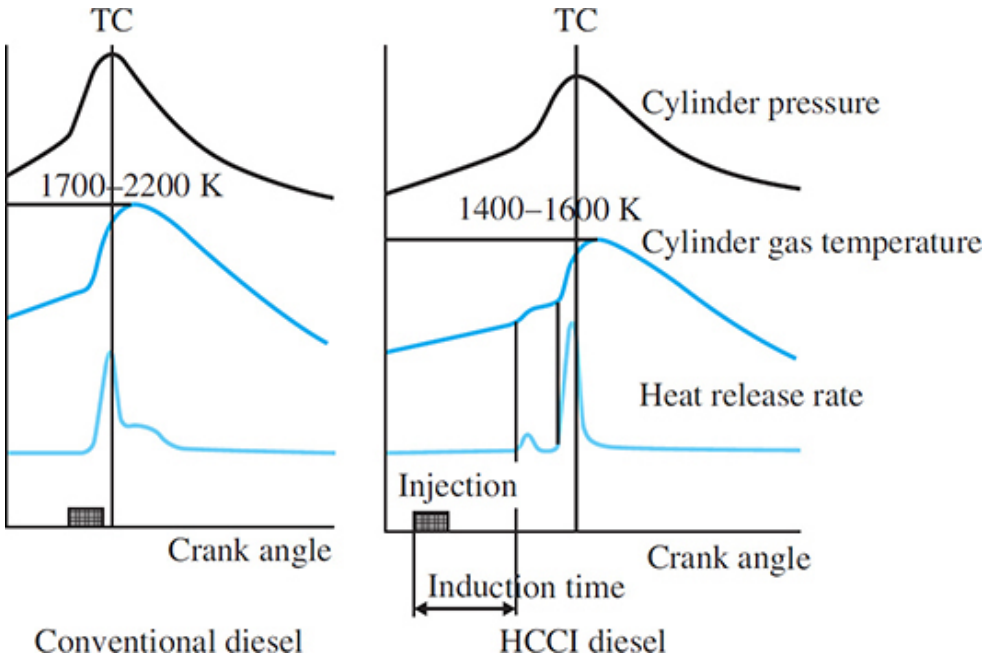


Figure 10.56 Profiles of cylinder pressure, average gas temperature, and heat-release rate versus crank angle for a standard diesel combustion process, and an HCCI combustion process, are shown. The much longer ignition delay with HCCI combustion enables more complete mixing between fuel vapor and air, and thus eliminates the mixing-controlled phase of diesel combustion. [55]

Many different approaches to achieving these objectives through spontaneous ignition of well-mixed lean (and dilute) mixtures have been proposed and examined. These are predominantly *low temperature combustion* (LTC) approaches for the reasons that Fig. 10.56 illustrates.

They can be grouped in two categories as "premixed" or "late" combustion modes (see Table 10.5). These correspond to *early* fuel injection approaches which allow much more time for premixing the fuel and air, and *late* injection which, due to the lengthening of the ignition delay, results in better fuel-air mixing than with the standard diesel injection and combustion processes. However, with late injection, dilution with EGR is usually needed to further reduce the peak burned-gas temperatures, and both of these changes require a speeding up of the combustion process to maintain high engine efficiency.

Many different labels have been proposed for the various approaches: for example, premixed charge compression ignition (PCCI), reaction controlled compression ignition (RCCI), modulated kinetics (MK) (a late combustion concept). Each of these titles abbreviates the essential feature of these advanced combustion concepts—more complete mixing (pre-mixing) of the charge, control of combustion initiation and progression through managing the kinetics of the combustion chemistry, and reducing the burned gas (flame) temperatures through dilution with air, residual gas, or EGR. See Ref. 55 for an overview.

The fuel-air equivalence ratio temperature paths of an early injection and late injection low-temperature combustion concept identify both the opportunities and problems of these advanced combustion approaches. Figure 10.57 shows the gas temperatures that result from burning a homogeneous mixture of diesel-like fuel and air at the specified fuel-air equivalence ratio and temperature of 1000 K. For very rich mixtures ($\phi \geq 2$) this temperature can exceed the equilibrium burned gas temperature by several hundred degrees K before endothermic reactions lower the temperature to its equilibrium value. The soot formation region (richer than $\phi \approx 2$ with T between 1600 K and 2200 K) with percentages of fuel carbon converted to soot, and the NO formation region, are shown in Fig. 10.58. Also shown are lines of 21% (air), 15%, and 10% oxygen (corresponding to increasing levels of EGR). A typical path for a fuel parcel in a conventional diesel combustion process (with air) ignites at a ϕ in the 2 to 4 range, and enters the soot formation region as rich reacting fuel-air mixture; then in the diffusion flame ($\phi \approx 1$, $T = 2700\ K$) the burned gases are in the NO formation region. A and B indicate the premixed (rapid) burning phase and mixing-controlled (diffusion) burning phase of the diesel combustion process.

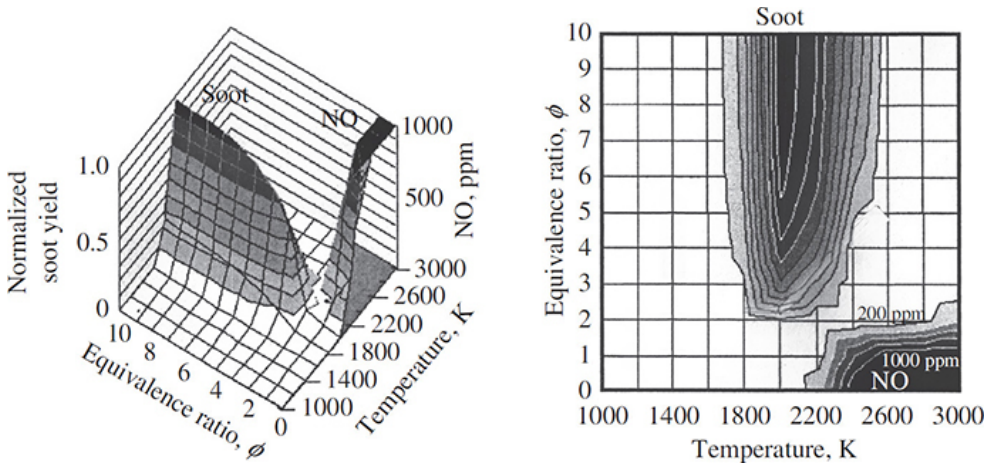


Figure 10.57 Fuel-air equivalence ratio versus burned gas temperature plot relevant to diesel combustion analysis showing soot formation (rich mixtures) and NO formation (high temperature, near stoichiometric) regions.[58]

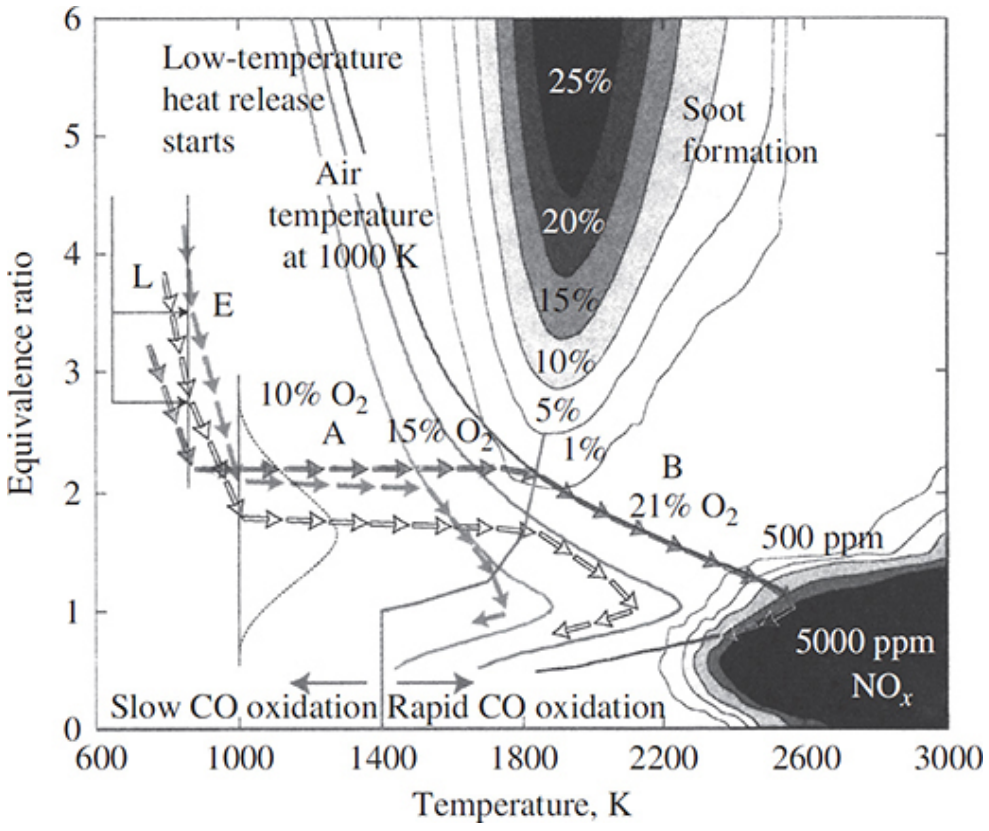


Figure 10.58 Paths for two low-temperature combustion modes in the fuel-air equivalence ratio (f) and burned gas temperature (T) plane: early

injection (E), late injection (L), and standard diesel combustion (A). High EGR contours (15, 10% O_2) are also shown. [59]

A path characteristic of an early-injection high-dilution combustion concept is shown in Fig. 10.58 by the filled-in arrows (E). Typical operating conditions for this path are 10% oxygen concentration (65% EGR) and SOI at 20° crank-angle BTC. For the late-combustion example (L, open arrows), the SOI was close to TC and the O_2 concentration was about 15%. The longer delay (relative to conventional diesel combustion, A, solid heavier arrows) gives a "less rich" mixture composition at SOC, but temperatures are lower during the bulk of the combustion process due to the EGR dilution. There is still a diffusion-controlled burning process with the late-injection case but its NO formation impact is reduced due to the increased dilution and its later occurrence, and thus lower peak flame temperature. The temperature boundary between more rapid and less rapid CO oxidation regimes is indicated. This is important, because for rich mixtures ($\phi > 1$) the products of premixed combustion (CO, H_2, and other organic compounds) must be fully oxidized to utilize the fuel's chemical energy completely.

Successful implementation of these various HCCI and low-temperature combustion concepts does lead to higher engine efficiencies and lower particulate and NO emissions in those regions of the engine-operating map where these advanced combustion concepts can be used. However, these regions are generally limited to lighter loads (mep values well below the maximum) by the very rapid HCCI combustion rate, which follows ignition, which leads to knock-like pressure oscillations. With extremely lean operation (and at high dilution rates with EGR) misfire becomes a problem resulting in high levels of engine-out hydrocarbon and carbon monoxide emissions. Also, engine control depends on managing the charge temperature and composition in each cycle appropriately, and is made difficult by the strong dependence of spontaneous ignition of fuel-air mixtures to these variables, especially temperature. Use of "partially premixed" fuel-air-residual gas mixtures with temperature and composition stratification alleviates such challenges to a degree, but these limits to a broad operating range still remain.

A basic issue is that engines usually need to operate over a wide speed range (e.g., SI engines 600 to 6000 rev/min). Diesels and spark-ignition

engines employ a combustion process whose rate scales fundamentally with the in-cylinder fluid turbulence, which scales with engine speed. Kinetically-controlled engine combustion concepts do not have that inherent combustion-rate scaling with engine speed.

PROBLEMS

10.1. Describe the sequence of processes which must occur before the liquid fuel in the injection system in a direct-injection compression-ignition engine is fully burned.

10.2. Small high-swirl direct-injection CI engines have fuel conversion efficiencies which are about 10% higher than values typical of equivalent indirect-injection engines. (IDI engines were used previously.) What combustion-system-related differences between DI and IDI diesels contribute to this higher efficiency?

10.3. In a diesel engine, because the fuel distribution is nonuniform the burned gas temperature is nonuniform. Consider small fuel-air mixture elements initially at 1000 K and 6.5 MPa at top-center with a range of equivalence ratios. Each element burns at essentially constant pressure. Calculate (using the charts in Chap. 4 , or an appropriate chemical equilibrium thermodynamic computer code) the burned gas temperature for mixture equivalence ratios of 0.4, 0.6, 0.8, 1.0, 1.2. (Assume the fuel is isooctane.) Note: the reaction zone in the main diesel combustion phase (a diffusion flame) is stoichiometric.

10.4. The levels of combustible species in the exhaust of a direct-injection diesel engine are: HC, 0.8 g/kW·h; CO, 3 g/kW·h; hydrogen, negligible; particulates, 0.7 g/kW·h. If the specific fuel consumption is 210 g/kW·h, calculate the combustion efficiency.

10.5. Consider a turbocharged 8-liter six-cylinder direct-injection diesel engine (see Fig. 15.34) operating at maximum power (210 kW), 2500 rev/min and an equivalence ratio of 0.6. Bore = 114 mm, stroke = 130 mm, r_c = 16.5. Estimate the following:

(a) Mass of air in each cylinder per cycle, mass and volume (as liquid) of diesel fuel injected per cylinder per cycle.

(b) Estimate the spray penetration as a function of time, the liquid core length (of an evaporating spray), the lift-off length, and the average drop size. Use equations from the relevant sections of Chap. 10 . You will need to make several assumptions about parameters that define the engines operating conditions and fuel spray details. List and briefly explain your assumptions.

(c) Assuming all fuel droplets are the same size as the average drop, how many drops are produced per injection? If these drops are uniformly distributed throughout the air charge at TC, what is the approximate distance between drops? (Neither of these assumptions is correct; however, the calculations illustrate the challenges of the fuel-air mixing process.)

10.6. Estimate the following quantities for a typical direct-injection diesel fuel spray. The injection pressure is 500 bar; the cylinder pressure during injection is 50 bar.

(a) Assuming that the flow through the nozzle orifice is incompressible and quasi-steady, estimate the liquid fuel velocity at the orifice exit. At this velocity, how long would the fuel take to reach the cylinder wall? The bore is 125 mm.

(b) Each nozzle orifice diameter d_n is 0.34 mm and $L_n/d_n = 4$. Determine the spray spreading angle and plot spray tip penetration versus time.

(c) Use Eq. (10.28) to estimate the initial average drop size assuming that the injection process in (*a*) above continues for 1 millisecond and the injector nozzle has four orifices.

10.7. Diesel fuel is injected as a liquid at room temperature into air at 50 bar and 800 K, close to TC at the end of compression. If the overall equivalence ratio is 0.7, estimate the reduction in average air temperature which would occur when the fuel is fully vaporized and uniformly mixed. Assume such mixing takes place at constant volume prior to any combustion.

10.8. Using Eq. (10.35) estimate the ignition delay in milliseconds and crank-angle degrees for these operating conditions: low swirl DI diesel at 600 rev/min; high swirl DI diesel 1800 rev/min, with lower and higher compression ratio, respectively: other details as in Table 10.3. The fuel cetane number is 45; stroke = 135 and 83 mm, respectively. Discuss differences in the predicted ignition-delay values.

10.9. The compression ratio of truck diesel engines must be set at about 16 so that the engine will start when cold. Using Eqs. (10.35) to (10.37) develop a graph of τ_{id} (in degrees) as a function of compression ratio for r_c = 12 to 20. Assume p_i = 1 bar, T_i = 255 K, n = 1.13, speed = 100 rev/min, bore = stroke = 120 mm, fuel cetane number = 45. If the ignition delay must be less than 20° CA for satisfactory starting, what compression ratio is required?

10.10. Data and equations in this chapter can be used to estimate the fraction of the fuel injected into a direct-injection diesel engine which burns in the premixed phase. Use this information to explain why, for turbocharged DI diesel engines the ignition delay and relative amount of fuel burned during the premixed combustion phase is less important than it is for naturally-aspirated engines.

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[a] Note that the term *compression-ignition*, widely used to characterize the ignition of fuel-air mixture in diesel engines, is really a much abbreviated characterization of the actual process that occurs. That process is the spontaneous ignition of fuel-vapor, air, mixture within the combustion chamber close to top center (TC), due to its high temperature and pressure (which are a consequence of the compression process) shortly after fuel injection commences).

[b] Note that this latter definition (usually called *net* heat release) combines the combustion energy release with heat losses to the cylinder walls.

[c] These gas conditions in the combustion vessel were realized by burning an appropriate mixture of nitrogen, oxygen, acetylene, and hydrogen in the vessel before diesel fuel injection occurred, allowing

time for equilibrium and, if necessary, cooling, to achieve the desired gas temperature, pressure, density, and oxygen concentration.

[d] The heat-release rate plotted here is the net heat-release rate (see Sec. 10.4). It is the sum of the change of sensible internal energy of the cylinder gases and the work done on the piston. It differs from the rate of fuel energy released by combustion by the heat transferred to the combustion chamber walls, and the thermal energy absorbed by fuel vaporization. The heat loss to the walls is 10 to 25% of the fuel heating value in smaller engines; it is less in larger engine sizes. This net heat release can be used as an indicator of the fuel's chemical energy release when the heat loss is small.

[e] This two-step model illustrates why the initial premixed rapid-burning part of the heat release rate profile occurs. However, its second step—becomes ready for burning—involves several stages as discussed more fully in Sec. 10.3.4 .

[f] That is, $U = U_S = U(T) - U(298\ \text{K})$ and $h_f = h_{S,\,f} = h_f(T) - h_f(298\ \text{K})$; see Sec. 5.5 for definitions.

[g] Typical numbers: pump and distribution system, up to 1500 bar; common rail system, up to 1700 bar; unit injectors, up to 2500 bar; piezoelectric injectors, up to 1800 bar.

[h] The back lighting identifies regions where sufficient liquid fuel (as ligaments or drops) is present to attenuate the light. The shadowgraph technique responds to density gradients in the test section, so it identifies regions where fuel vapor exists.

[i] In the original procedure α-methyl-napthalene ($C_{11}H_{10}$) with a cetane number of zero represented the bottom of the scale. Heptamethyl-nonane, a more stable compound, has replaced it.

[j] API (American Petroleum Institute) gravity is based on specific gravity and is calculated from: API gravity (degrees) = 141.5/(specific gravity at 60°F) − 131.5.

[k] These are usually fatty acid methyl ester (FAME) and fatty acid ethyl ester (FAEE).

[l] The spray and flame tracings shown in Fig. 10.49 do not show this flame lift-off. This is likely due to the fuel injection details in this older study resulted in short lift-off distances.

CHAPTER 11

Pollutant Formation and Control

11.1 NATURE AND EXTENT OF PROBLEM

Spark-ignition (SI) and diesel engines are major sources of air pollutants. The spark-ignition engine exhaust gases contain oxides of nitrogen (nitric oxide, NO, and small amounts of nitrogen dioxide, NO_2—collectively known as NO_x), carbon monoxide (CO), and organic compounds, which are unreacted or partially reacted fuel hydrocarbons (HC). The relative amounts of these *engine-out emissions* that enter exhaust depend on engine design and operating conditions but are of order: NO_x, 500 to 1000 ppm or 20 g/kg fuel; CO, about 1% or 150 g/kg fuel; and HC, 2000 ppm (as C_1) or 15 g/kg fuel. [a] Catalytic converters in the exhaust system now reduce these engine-out emissions by over 90%. Piston blowby gases, and fuel evaporation and release to the atmosphere through vents in the fuel system, especially after engine shut-down, were historically additional sources of unburned hydrocarbons. However, in modern engines these nonexhaust sources are controlled by returning the blowby gases from the crank-case to the engine intake system and by venting the fuel tank through a vapor-absorbing carbon cannister which is purged with some of the engine intake air during normal engine operation.

In diesel engine exhaust, concentrations of NO_x are comparable to those from SI engines. Diesel hydrocarbon emissions are significant, though exhaust concentrations are much lower than typical SI engine levels. The hydrocarbons in the exhaust may also condense to form white smoke during engine starting and warm-up. Specific hydrocarbon compounds in the exhaust

gases are the source of diesel odor. Diesel engines are an important source of particulate emissions; between about 0.2 and 0.5% of the fuel mass is emitted as small (10 to 500 nm diameter) particles, which consist primarily of soot with additional hydrocarbon material. Diesel engines are not a significant source of carbon monoxide.

Use of alcohol fuels in either of these engines substantially increases aldehyde emissions. Aldehydes become a significant contributor to photochemical smog when these oxygenates are blended with gasoline (e.g., 10% ethanol with gasoline, E10). The dominant fuels, gasoline and diesel, contain sulfur, the amount varying depending on world region. The sulfur is oxidized (i.e., burned) to produce sulfur dioxide, SO_2, of which a fraction can be oxidized to sulfur trioxide, SO_3. The latter combines with water to form a sulfuric acid aerosol. Sulfur in these fuels, over time, degrades the effectiveness of the exhaust catalysts now widely used to substantially reduce engine-out emissions. In many parts of the world, the sulfur content of transportation fuels is tightly regulated, and "sulfur-free fuels" with <10 ppm (by weight) sulfur are required in the major developed-world regions. Fuel quality varies widely in other parts of the world and, historically, levels have been much higher (of order 600 ppm in gasoline, up to 0.5% in diesel).

In general, the concentrations of these pollutants in internal combustion engine exhaust gases differ from values calculated assuming chemical equilibrium. Thus the detailed chemical mechanisms by which these pollutants form and the kinetics of these processes are important in determining emission levels. For some pollutant species, for example, carbon monoxide, organic compounds, and particulates, the formation and destruction reactions are intimately coupled with the primary fuel combustion process. Thus an understanding of the formation of these species requires knowledge of the combustion chemistry. For nitrogen oxides and sulfur oxides, the formation and destruction processes are not part of the fuel combustion process. However, the reactions, which produce these species, take place in an environment created by the combustion reactions, so the two processes are still linked. A summary of the mechanisms by which these pollutants form in internal combustion engines provides an introduction to this chapter. In subsequent sections, the details of the basic formation mechanisms of each pollutant and the application of these mechanisms to the combustion process in both spark-ignition and compression-ignition engines will be developed.

As noted above, catalytic converters and particulate traps are now built into engine exhaust systems to *clean up* the exhaust gases before they leave the tailpipe and enter the atmosphere. These exhaust treatment devices will be described in Sec. 11.6 . The efficiency of these exhaust treatment devices is very high: the percent of the engine-out emissions removed is usually over 90%. However, to be effective, the temperature of the catalysts used for HC, CO, and NO_x removal must be at about 250°C for the catalyst to be effective. Thus engine-out emissions levels during this catalyst warm-up period are especially important, as is achieving the catalyst *light-off* state rapidly. Once the catalyst or particulate trap has fully warmed-up, the engine's task is to feed exhaust gas of the appropriate composition (e.g., relative air/fuel or fuel/air ratio) and temperature to realize very high efficiency and low vehicle emissions levels. This is the context for our review of the mechanisms within the engine by which these pollutants form.

The processes by which pollutants form within the cylinder of a conventional spark-ignition engine are illustrated in Fig. 11.1. The schematic shows the combustion chamber during four different phases of the engine operating cycle: compression, combustion, expansion, and exhaust. Nitric oxide (NO) forms in the high-temperature burned gases behind the flame through chemical reactions involving nitrogen and oxygen atoms and molecules: this nitrogen/oxygen system does not attain chemical equilibrium. The higher the burned gas temperature, the higher is the rate of formation of NO. As the burned gases cool during the expansion stroke the reactions involving NO freeze, and leave NO concentrations well in excess of levels corresponding to equilibrium at exhaust conditions. Carbon monoxide also forms during the combustion process. With rich fuel-air mixtures, there is insufficient oxygen to burn fully all the carbon in the fuel to CO_2: also in the high-temperature products, even with lean mixtures, dissociation produces significant CO levels. Later in the expansion stroke, the CO oxidation process also freezes as the burned gas temperature falls.

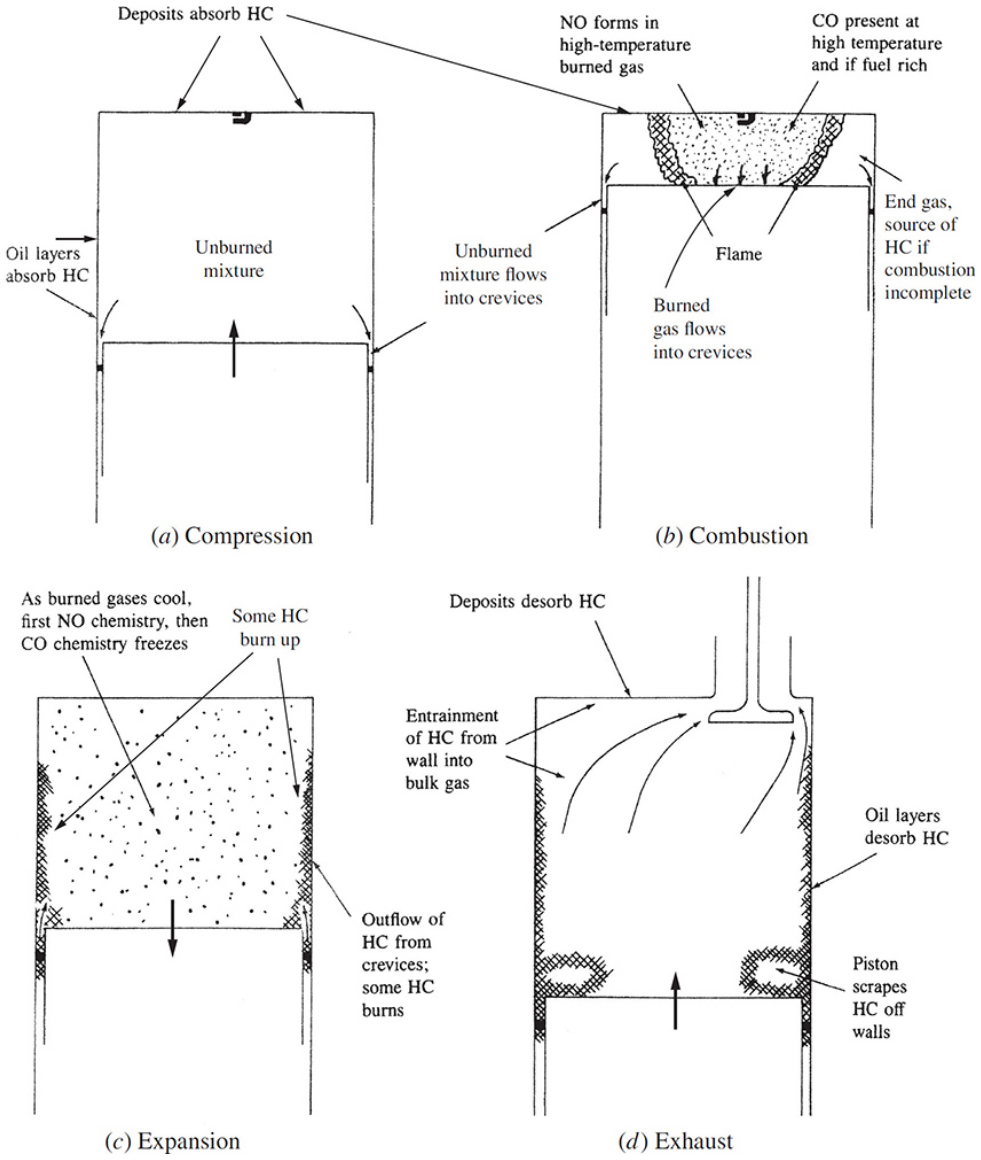


Figure 11.1 Summary of HC, CO, and NO pollutant formation mechanisms in a spark-ignition engine.

The hydrocarbon emissions have several different sources. During compression and combustion, the increasing cylinder pressure forces some of the gas in the cylinder into crevices—narrow volumes connected to the combustion chamber: the volumes between the piston, rings, and cylinder wall are the largest of these. Most of this gas in the crevices is unburned fuel-air mixture, which escapes the primary combustion process because the entrance to these crevices is too narrow for the flame to enter. This gas, which leaves these crevices during the expansion and exhaust processes, is

the major source of hydrocarbon emissions. Another source is flame quenching on the combustion chamber walls. A quench layer containing unburned and partially burned fuel-air mixture is left at the wall when the flame, as it approaches the wall, extinguishes. However, the HC in these thin (≤0.1 mm) layers burn up rapidly after flame quenching so this is not a large source. It has been shown that the porous deposits on the walls of engines in actual operation do increase engine HC emissions, due to a flame quenching process. An additional source of hydrocarbons is believed to be the engine oil left in a thin film on the cylinder liner. This oil layer absorbs and desorbs fuel hydrocarbon components, before and after combustion, respectively, thus permitting a fraction of the fuel to escape the primary combustion process unburned. A final source of HC in engines is incomplete combustion due to bulk quenching of the flame in that fraction of engine cycles where combustion is especially slow (see Sec. 9.4.3). Such conditions are most likely to occur during transient engine operation when the air/fuel ratio, spark timing, and the fraction of the exhaust recycled for emission control are not properly matched.

These as yet unburned hydrocarbons exit the cylinder by being entrained into the bulk-gas flow during blowdown, and at the end of the exhaust stroke as the piston pushes gas scraped off the wall out of the exhaust valve. Oxidation of the hydrocarbons which escape the primary combustion process by any of the above processes can occur during expansion and exhaust. The amount of oxidation depends on the temperature and oxygen-concentration time histories of these HC as they mix with the bulk gases.

One of the most important variables in determining spark-ignition engine emissions is the relative air/fuel ratio, l. Figure 11.2 shows how NO, CO, and HC exhaust emissions vary with this parameter. The spark-ignition engine has historically been operated close to stoichiometric, or slightly fuel-rich, to ensure smooth and reliable operation. Figure 11.2 shows that leaner mixtures give lower CO and HC emissions until the combustion quality becomes poor (and eventually misfire occurs), when HC emissions rise sharply and engine operation becomes erratic. However, NO emissions peak about 10% lean of stoichiometric. The shapes of these curves indicate the complexities of emission control. In a cold engine, when fuel vaporization is slow, the fuel flow may be increased to provide an easily combustible slightly-rich mixture in the cylinder. But then CO and HC emissions are higher. At part-load conditions, lean mixtures could be used which would

produce lower HC and CO emissions (at least until the combustion quality deteriorates), but NO emissions would be high. Use of recycled exhaust to dilute the engine intake mixture lowers the NO levels, but also deteriorates combustion quality. Exhaust gas recirculation (EGR) is used with stoichiometric mixtures in many engine control systems. Note that the highest power levels are obtained from the engine with slightly rich-of-stoichiometric mixtures and no recycled exhaust to dilute the incoming charge. As we will see, several emission control techniques are required to reduce engine-out emissions of all three pollutants, over all engine operating modes.

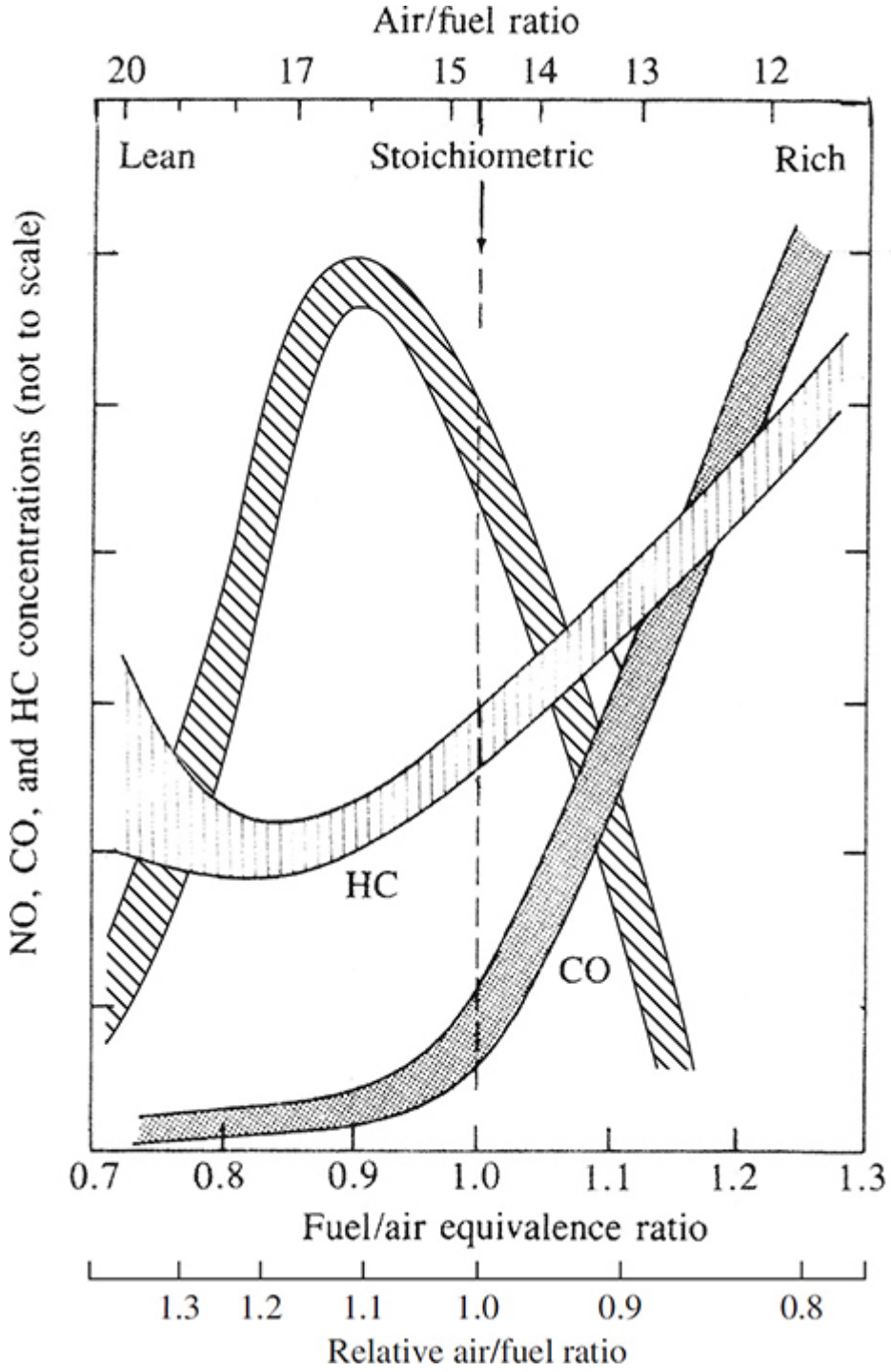


Figure 11.2 Variation of HC, CO, and NO concentration in the exhaust of a conventional spark-ignition engine with relative air-fuel ratio and fuel/air equivalence ratio.

In the diesel engine, the fuel is injected into the cylinder just before combustion starts, so throughout most of the critical parts of the cycle the fuel distribution is nonuniform. The pollutant formation processes are strongly dependent on the fuel distribution and how that distribution changes with time due to mixing with hot air. Figure 11.3 illustrates how various parts of the fuel spray and diffusion flame affect the formation of NO, unburned HC, and soot (or particulates) during the "premixed" and "mixing-controlled" phases of diesel combustion in a direct-injection (DI) engine with swirl. Nitric oxide forms in the high-temperature burned gas regions as before, but temperature and fuel/air ratio distributions within the burned gases are now nonuniform and formation rates are highest in the close-to-stoichiometric diffusion flame reaction zone. Soot forms in the rich unburned-fuel-containing core of fuel spray, after the fuel has vaporized through mixing with hot entrained air (and later with burned gas). Soot oxidizes in the diffusion region flame when it contacts oxygen, giving rise to the yellow luminous character of the flame. Hydrocarbons and aldehydes originate in regions where the flame quenches both on the walls and where excessive dilution with air prevents the combustion process from either starting or going to completion. Fuel that vaporizes from the nozzle sac volume during the later stages of combustion is also a source of HC. Combustion-generated noise is controlled by the early part of the premixed (rich) combustion process, the initial rapid heat release immediately following the ignition delay.

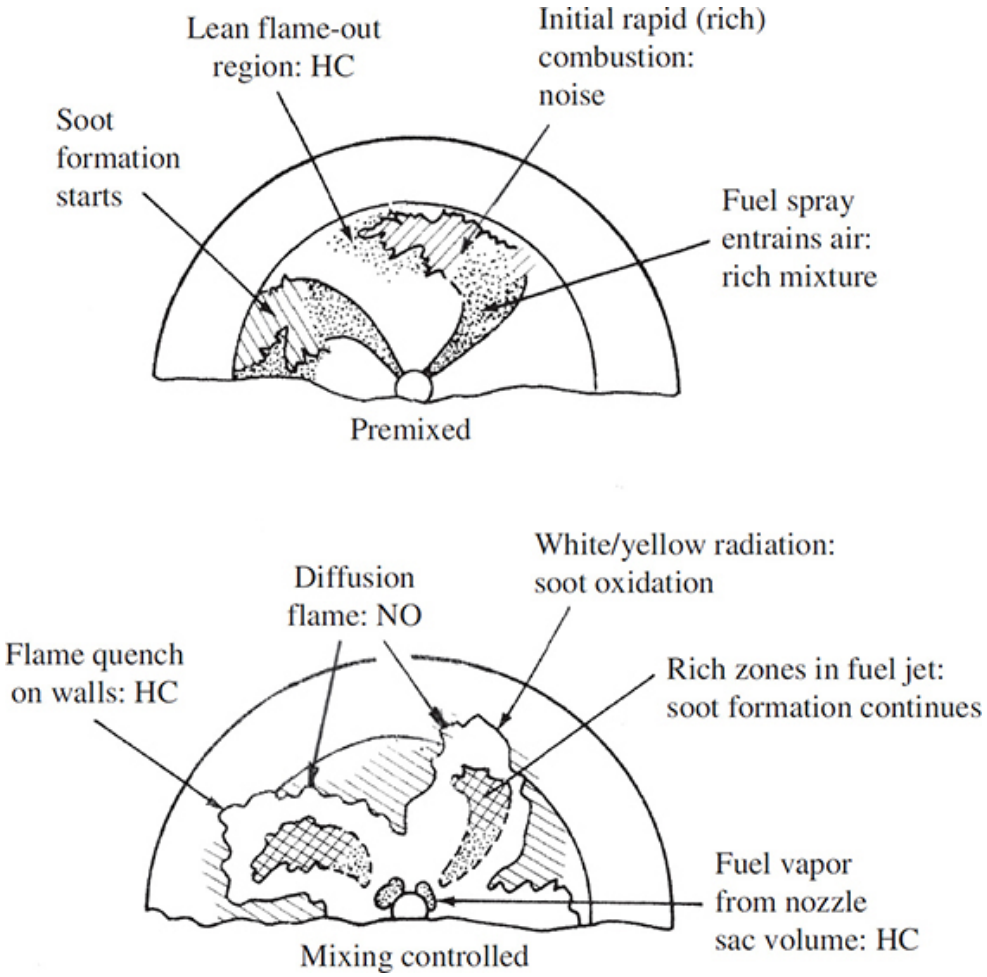


Figure 11.3 Summary of pollutant formation mechanisms in a direct-injection diesel engine during "premixed" and "mixing-controlled" combustion phases.

11.2 NITROGEN OXIDES

11.2.1 Kinetics of NO Formation

While nitric oxide (NO) and nitrogen dioxide (NO_2) are usually grouped together as NO_x emissions, nitric oxide is the predominant oxide of nitrogen produced inside the engine cylinder. The principal source of NO is the oxidation of atmospheric (molecular) nitrogen. The mechanism of NO formation from atmospheric nitrogen has been studied extensively. [1] It is

generally accepted that in combustion of near-stoichiometric fuel-air mixtures, the principal reactions governing the formation of NO from nitrogen (and its destruction) are [b]

$$N_2 + O = NO + N \tag{11.1}$$

$$N + O_2 = NO + O \tag{11.2}$$

$$N + OH = NO + H \tag{11.3}$$

The forward and reverse rate constants k_i^+ and k_i^-, respectively) for these reactions have been measured in numerous experimental studies. Recommended values for these rate constants taken from a critical review of this published data are given in Table 11.1. Note that the equilibrium constant for each reaction, $K_{c,i}$ (see Sec. 3.7.2), is related to the forward and reverse rate constants by $K_{c,i} = k_i^+/k_i^-$. The rate of formation of NO via reactions (11.1) to (11.3) is given by [see Eqs. (3.55) and (3.58)]

$$\frac{d[NO]}{dt} = k_1^+[N_2][O] + k_2^+[N][O_2] + k_3^+[N][OH] - k_1^-[NO][N] - k_2^-[NO][O] - k_3^-[NO][H] \tag{11.4}$$

TABLE 11.1 Rate constants for NO formation mechanism [1]

Reaction	Rate constant, $cm^3/mol \cdot s$	Temperature range, K	Uncertainty, factor of or %
(1) $N_2 + O \rightarrow NO + N$	$7.6 \times 10^{13} \exp[-38{,}000/T]$	2000–5000	2
(−1) $N + NO \rightarrow N_2 + O$	1.6×10^{13}	300–5000	±20% at 300 K 2 at 2000–5000 K
(2) $N + O_2 \rightarrow NO + O$	$6.4 \times 10^9\ T \exp[-3150/T]$	300–3000	±30% at 300–1500 K 2 at 3000 K
(−2) $O + NO \rightarrow O_2 + N$	$1.5 \times 10^9\ T \exp[-19{,}500/T]$	1000–3000	±30% at 1000 K 2 at 3000 K
(3) $H + OH \rightarrow NO + H$	4.1×10^{13}	300–2500	±80%
(−3) $H + NO \rightarrow OH + N$	$2.0 \times 10^{14} \exp[-23{,}650/T]$	2200–4500	2

where [ ] denotes species concentrations in moles per cubic centimeter when k_i have the values given in Table 11.1. The forward rate constant for reaction (11.1) and the reverse rate constants for reactions (11.2) and (11.3) have

large activation energies, which result in a strong temperature dependence of NO formation rates.

A similar relation to (11.4) can be written for $d[N]/dt$:

$$\frac{d[\mathrm{N}]}{dt} = k_1^+[\mathrm{N_2}][\mathrm{O}] - k_2^+[\mathrm{N}][\mathrm{O_2}] - k_3^+[\mathrm{N}][\mathrm{OH}] - k_1^-[\mathrm{NO}][\mathrm{N}] - k_2^-[\mathrm{NO}][\mathrm{O}] - k_3^-[\mathrm{NO}][\mathrm{H}] \tag{11.5}$$

Since [N] is much less than the concentrations of other species of interest ($\sim 10^{-8}$ mole fraction), the steady-state approximation *c* is appropriate: $d[\mathrm{N}]/dt$ is set equal to zero and Eq. (11.5) used to eliminate [N]. The NO formation rate then becomes

$$\frac{d[\mathrm{NO}]}{dt} = 2k_1^+[\mathrm{N_2}][\mathrm{O}] \frac{1 - [\mathrm{NO}]^2/(\mathrm{K}[\mathrm{O_2}][\mathrm{N_2}])}{1 + k_1^-[\mathrm{NO}]/(k_2^+[\mathrm{O_2}] + k_3^+[\mathrm{OH}])} \tag{11.6}$$

where $K = (k_1^+/k_1^-)(k_2^+/k_2^-)$.

NO forms in both the flame front and the post-flame gases. In engines, however, combustion occurs at high pressure so the flame reaction zone is extremely thin (~0.1 mm) and residence time within this zone is short. Also, the cylinder pressure rises during most of the combustion process, so burned gases produced early in the combustion process are compressed to a higher temperature than they reached immediately after combustion. Thus, NO formation in the post-flame gases almost always dominates any flame-front–produced NO. It is appropriate, therefore to assume that the combustion chemistry and NO formation processes are decoupled and to approximate the concentrations of O, O_2, OH, H, and N_2 by their equilibrium values at the local pressure and equilibrium temperature.

To introduce this equilibrium assumption it is convenient to use the notation $R_1 = k_1[\mathrm{N_2}]_e[\mathrm{O}]_e = k_1^-[\mathrm{NO}]_e[\mathrm{N}]_e$, where $[\ ]_e$ denotes equilibrium concentration, for the one-way equilibrium rate for reaction (11.1), with similar definitions for $R_2 = k_2^+[\mathrm{N}]_e[\mathrm{O_2}]_e = k_2^-[\mathrm{NO}]_e[\mathrm{O}]_e$ and $R_3 = k_3^+[\mathrm{N}]_e[\mathrm{OH}]_e = k_3^-[\mathrm{NO}]_e[\mathrm{H}]_e$. Substituting $[\mathrm{O}]_e$, $[\mathrm{O_2}]_e$, $[\mathrm{OH}]_e$, $[\mathrm{H}]_e$, and $[\mathrm{N_2}]_e$ for [O], $[O_2]$, [OH], [H], and $[N_2]$ in Eq. (11.6) yields

$$\frac{d[NO]}{dt} = \frac{2R_1\{1-([NO]/[NO]_e)^2\}}{1+([NO]/[NO]_e)R_1/(R_2+R_3)} \tag{11.7}$$

Typical values of R_1, R_1/R_2, and $R_1/(R_2+R_3)$ are given in Table 11.2. The difference between R_1/R_2 and $R_1/(R_2+R_3)$ indicates the relative importance of adding reaction (11.3) to the mechanism.

TABLE 11.2 Typical values of R_1, R_1/R_2, and $R_1/(R_2+R_3)$ *

Equivalence ratio	$R_1^{\dagger}$	R_1/R_2	$R_1/(R_2+R_3)$
0.8	5.8×10^{-5}	1.2	0.33
1.0	2.8×10^{-5}	2.5	0.26
1.2	7.6×10^{-6}	9.1	0.14

*At 10 atm pressure and 2600 K.
†Units gmol/cm³·s.

The strong temperature dependence of the NO formation rate can be demonstrated by considering the initial value of $d[NO]/dt$ when $[NO]/[NO]_e \ll 1$. Then, from Eq. (11.7),

$$\frac{d[NO]}{dt} = 2R_1 = 2k_1^+[N_2]_e[O]_e \tag{11.8}$$

The equilibrium oxygen atom concentration is given by

$$[O]_e = \frac{K_{p(O)}[O_2]_e^{1/2}}{(\tilde{R}T)^{1/2}} \tag{11.9}$$

where $K_{p\,(O)}$ is the equilibrium constant for the reaction

$$\tfrac{1}{2}O_2 = O$$

and is given by

$$K_{p(O)} = 3.6 \times 10^3 \exp\left(\frac{-31{,}090}{T}\right) \quad \text{bar}^{1/2} \tag{11.10}$$

The initial NO formation rate may then be written [combining Eqs. (11.8) , (11.9), and (11.10) with k_1^+ from Table 11.1] as

$$\frac{d[\text{NO}]}{dt} = \frac{6 \times 10^6}{T^{1/2}} \exp\left(\frac{-69{,}090}{T}\right) [\text{O}_2]_e^{1/2} [\text{N}_2]_e \quad \text{mol/cm}^3 \cdot \text{s} \tag{11.11}$$

The strong dependence of $d[\text{NO}]/dt$ on temperature in the exponential term is evident. High temperatures and high oxygen concentrations result in high NO formation rates. Figure 11.4 shows the NO formation rate as a function of gas temperature and fuel/air equivalence ratio in post-flame gases. Also shown is the adiabatic flame temperature attained by a fuel-air mixture initially at 700 K at a constant pressure of 15 bar. For adiabatic constant-pressure combustion (an appropriate model for each element of fuel that burns in an engine), this initial NO formation rate peaks about 10% lean of the stoichiometric composition, and decreases rapidly as the mixture then becomes leaner or richer.

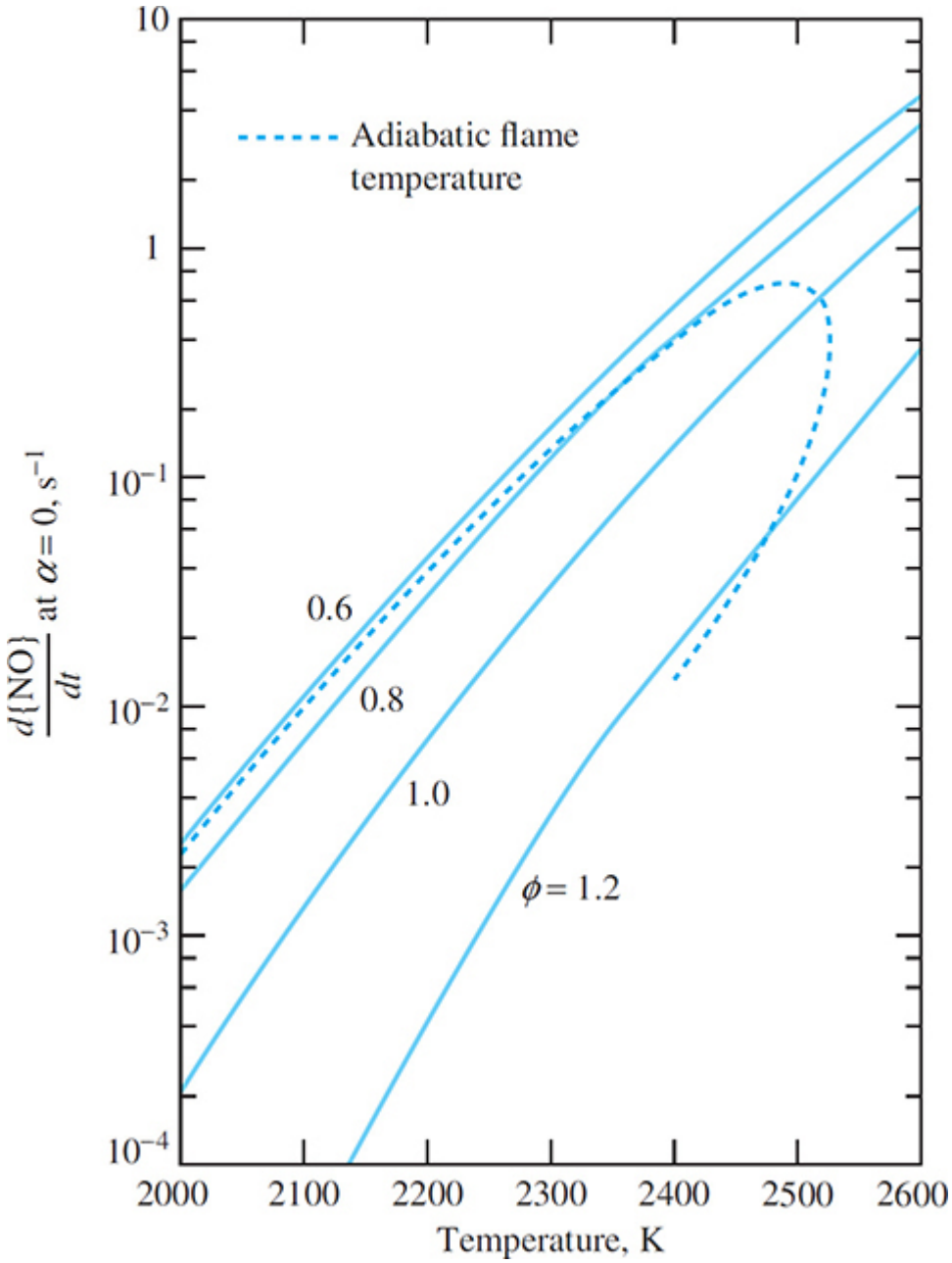


Figure 11.4 Initial NO formation rate, mass fraction per second (for $[NO]/[NO]_e << 1$) as a function of temperature for different equivalence ratios (ϕ) and 15 bar pressure. Dashed line shows adiabatic flame temperature for kerosene combustion with 700 K, 15 bar air. [3]

A characteristic time for the NO formation process, τ_{NO}, can be defined by

$$\tau_{NO}^{-1} = \frac{1}{[NO]_e}\frac{d[NO]}{dt} \tag{11.12}$$

$[NO]_e$ can be obtained from the equilibrium constant

$$K_{NO} = 20.3 \times \exp(-21{,}650/T)$$

as $[NO]_e = (K_{NO}[O_2]_e[N_2]_e)^{1/2}$. Equations (11.11) and (11.12) can be combined to give

$$\tau_{NO} = \frac{8\times10^{-16}T\exp(58{,}000/T)}{p^{1/2}} \tag{11.13}$$

where τ_{NO} is in seconds, T in kelvins, and p in bar. Use has been made of the fact that $\tilde{x}_{N_2} \approx 0.71$. For engine combustion conditions, τ_{NO} is usually comparable to or longer than the times characteristic of changes in engine conditions so the formation process is kinetically controlled. However, for close-to-stoichiometric conditions at the maximum pressures and burned gas temperatures, τ_{NO} is of the same order as typical combustion times (1 ms) and equilibrium NO concentrations may be attained. There is good evidence that this NO formation model is valid under conditions typical of those found in engines. [4]

11.2.2 Formation of NO_2

Chemical equilibrium considerations indicate that for burned gases at typical flame temperatures, NO_2/NO ratios should be negligibly small. While experimental data show this is true for spark-ignition engines, in diesels NO_2 can be 10 to 30% of the total oxides of nitrogen, engine-out, emissions. [5] A plausible mechanism for the persistence of NO_2 is the following. [6] NO formed in the flame zone can be rapidly converted to NO_2 via reactions such as

$$NO + HO_2 \rightarrow NO_2 + OH \tag{11.14}$$

Subsequently, conversion of this NO_2 to NO occurs via

$$NO_2 + O \rightarrow NO + O_2 \qquad (11.15)$$

unless the NO_2 formed in the flame is quenched by mixing with cooler fluid. This explanation is consistent with the highest NO_2/NO ratio occurring at light load in diesels, when cooler regions which could quench the conversion back to NO are widespread. [5]

Figure 11.5 shows examples of NO and NO_2 emissions data from a spark-ignition and a diesel engine. The maximum value for the ratio (NO_2/NO) for the SI engine is 2 percent, at a relative air-fuel ratio of about 1.2. For the diesel this ratio is higher, and is highest at light load, when the engine is operating very lean; it also depends on engine speed.

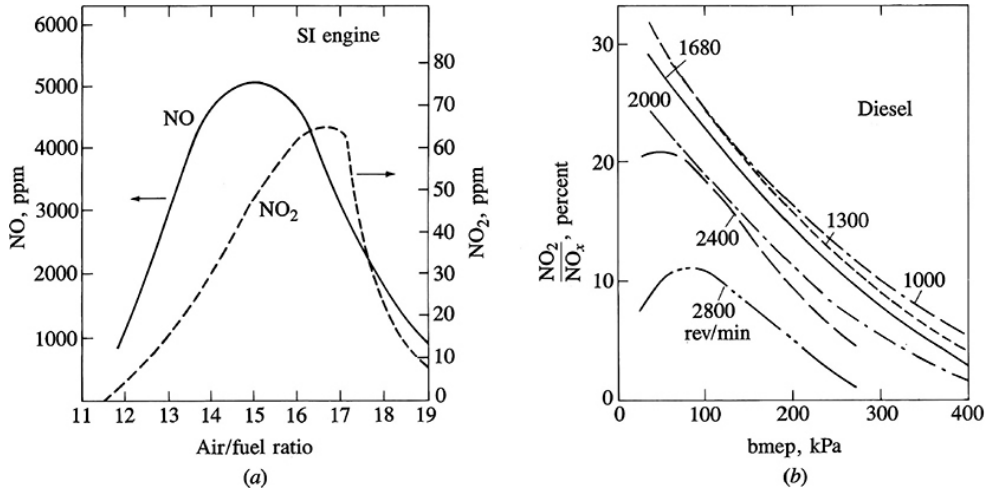


Figure 11.5 (*a*) NO and NO_2 concentrations in SI engine-out exhaust as function of air/fuel ratio, 1500 rev/min, wide-open throttle; (*b*) NO_2 as percent of total NO_x in diesel exhaust as function of load and speed.[5]

It is customary to measure total oxides of nitrogen emissions, NO plus NO_2, with a chemiluminescence analyzer and call the combination NO_x. It is always important to check carefully whether specific emissions data for NO_x are given in terms of mass of NO or mass of NO_2, which have molecular weights of 30 and 46, respectively.

11.2.3 NO Formation in Spark-Ignition Engines

In conventional spark-ignition engines the fuel and air (and any recycled exhaust) are mixed together during the engine intake process. Vigorous mixing with the residual gas within the cylinder also occurs during the intake. Thus the fuel/air ratio and the amount of diluent (residual gas plus any recycled exhaust) are close to uniform throughout the charge within the cylinder during combustion. [d] Since the composition is essentially uniform, the nature of the NO formation process within the cylinder can be understood by coupling the kinetic mechanism developed in Sec. 11.2.1 with the burned gas temperature distribution and pressure in the cylinder during the combustion and expansion processes. The temperature distribution which develops in the burned gases due to the passage of the flame across the combustion chamber has been discussed in Sec. 9.2.1 . Mixture which burns early is compressed to higher temperatures after combustion, as the cylinder pressure continues to rise; mixture which burns later is compressed primarily as unburned mixture and ends up after combustion at a lower burned gas temperature. Figures 11.6 *a* and *b* show measured cylinder pressure data from an operating engine, with estimates of the mass fraction burned (x_b) and the temperatures of a gas element which burned just after spark discharge and a gas element which burned at the end of the burning process. The model used to estimate these temperatures assumed no mixing between mixture elements which burn at different times. This assumption is more realistic than the alternative idealization that the burned gases mix rapidly and are thus uniform (see Sec. 9.2.1). Using the NO formation kinetic model [Eq. (11.7)] to calculate NO concentrations in these burned gas elements, with the equilibrium concentrations of the species O, O_2, N_2, OH, and H corresponding to the average fuel/air equivalence ratio and burned gas fraction of the mixture, and these pressure and temperature profiles, the rate-limited concentration profiles in Fig. 11.6*c* are obtained. Also shown are the NO concentrations that would correspond to chemical equilibrium at these conditions. The rate-controlled concentrations rise from the residual gas NO concentration, lagging the equilibrium levels, then cross the equilibrium levels, and “freeze” well above the equilibrium values corresponding to exhaust conditions. Depending on details of engine operating conditions, the rate-limited concentrations may or may not come close to equilibrium levels at peak cylinder pressure and gas temperature. Also, the amount of

decomposition from peak NO levels which occurs during expansion depends on engine conditions as well as whether the mixture element burned early or late. [7]

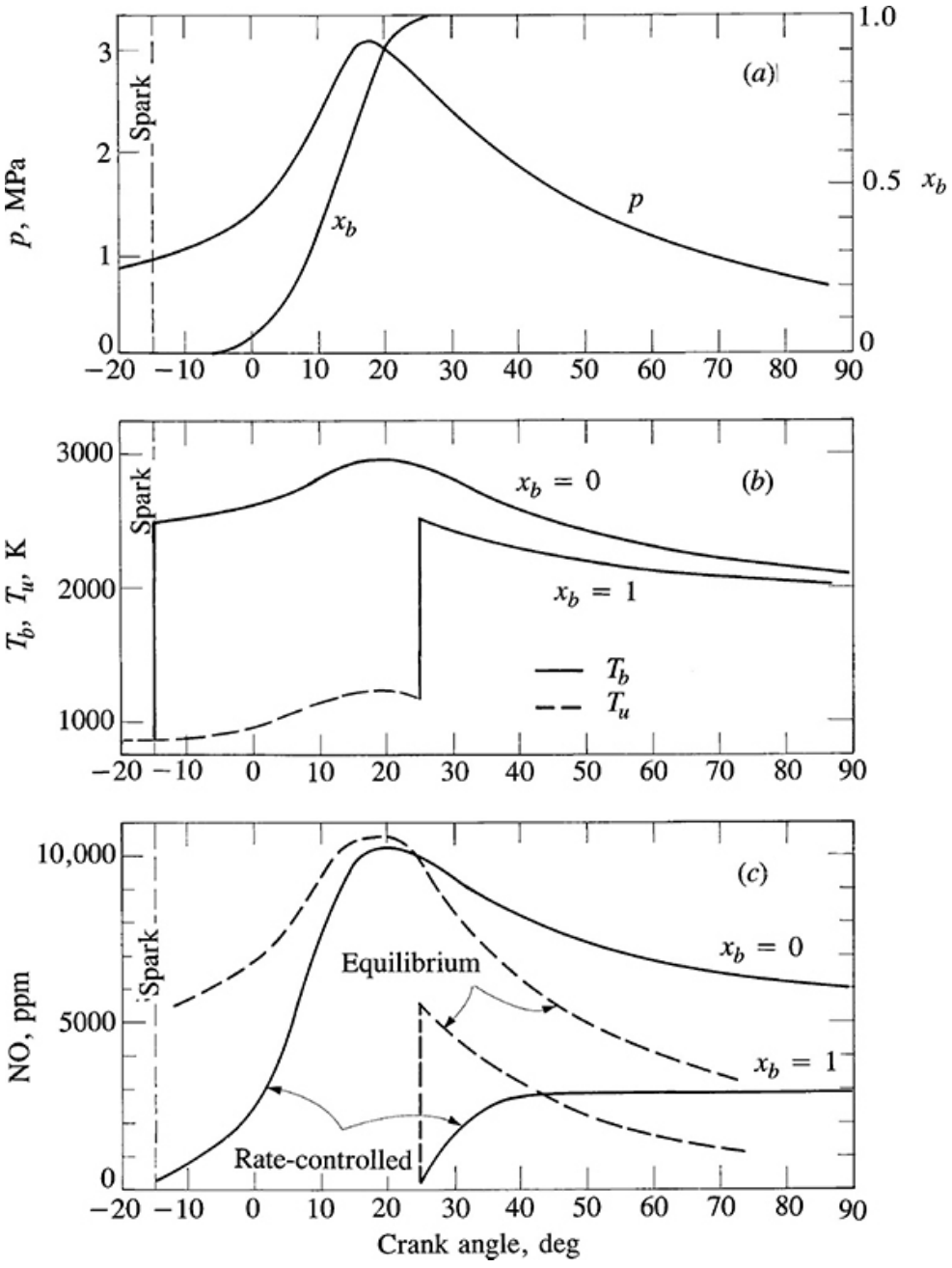


Figure 11.6 Illustration of SI engine NO formation process: (*a*) measured cylinder pressure p and calculated mass fraction burned x_b; (*b*) calculated temperature of unburned gas T_u and burned gas T_b in early- and late-burning

elements; (*c*) calculated NO concentrations in early- and late-burning elements for rate-controlled formation model and at equilibrium.[7]

Once the NO chemistry has frozen during the early part of the expansion stroke, integration over all elements will give the final average NO concentration in the cylinder which equals the exhaust concentration. Thus, if {NO} is the local mass fraction of NO, then the average exhaust NO mass fraction is given by

$$\{\overline{\mathrm{NO}}\} = \int_0^1 \{\mathrm{NO}\}_f \, dx_b \tag{11.16}$$

where $\{\mathrm{NO}\}_f$ is the final frozen NO mass fraction in the element of charge which burned when the mass fraction burned was x_b. Note that $\{\mathrm{NO}\} = [\mathrm{NO}] M_{\mathrm{NO}}/\rho$, where $M_{\mathrm{NO}} = 30$, the molecular weight of NO. The average exhaust concentration of NO as a mole fraction is given by

$$\tilde{x}_{\mathrm{NO}_{av}} = \{\overline{\mathrm{NO}}\}\frac{M_{\mathrm{exh}}}{M_{\mathrm{NO}}} \tag{11.17}$$

and the exhaust concentration in ppm is $\tilde{x}_{\mathrm{NO}_{av}} \times 10^6$. The earlier burning fractions of the charge contribute much more to the exhausted NO than do later burning fractions of the charge: frozen NO concentrations in these early-burning elements can be an order of magnitude higher than concentrations in late-burning elements. In the absence of vigorous bulk gas motion, the highest NO concentrations occur nearest the spark plug.

Substantial experimental evidence supports this description of NO formation in spark-ignition engines. The NO concentration gradient across the burned gas in the engine cylinder, due to the temperature gradient, has been demonstrated using gas sampling techniques [8,9] and using measurements of the chemiluminescent radiation from the reaction $\mathrm{NO} + \mathrm{O} \rightarrow \mathrm{NO}_2 + h\nu$ to determine the local NO concentration. Figure 11.7 shows NO concentration data as a function of crank angle, taken by Lavoie[10] through two different windows in the cylinder head of a specially constructed L-head engine where each window was a different distance from the spark plug. The stars indicate

the estimated initial NO concentration that results from mixing of the residual gas with the fresh charge, at the time of arrival of the flame at each window. The observed NO mole fractions rise smoothly from these initial values and then freeze about one-third of the way through the expansion process. NO levels observed at window W_2, closest to the spark plug, were substantially higher than those observed at window W_3. The dashed lines show calculated NO concentrations obtained using the NO formation kinetic model with an "unmixed" thermodynamic analysis for elements that burned at the time of flame arrival at each window. Since the calculated values started from zero NO concentration at the flame front (and not the diluted residual gas NO level indicated by the star), the calculations initially fall below the data. However, the difference between the two measurement locations and the frozen levels are predicted with reasonable accuracy. Thus, the rate-limited formation process, freezing of NO chemistry during expansion, and the existence of NO concentration gradients across the combustion chamber have all been observed.

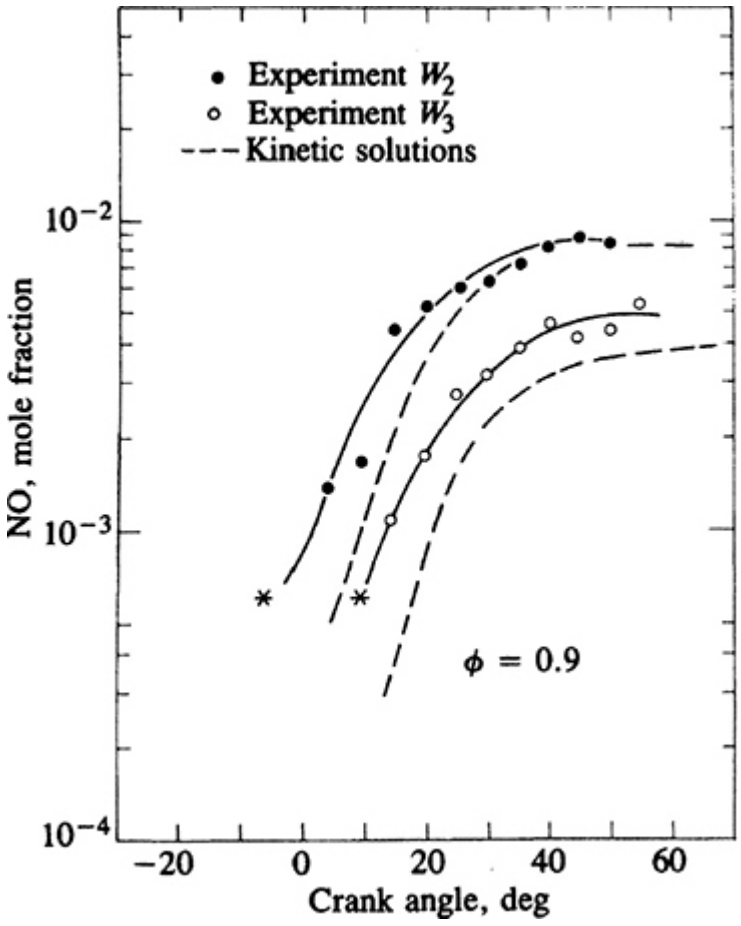


Figure 11.7 Spectroscopically measured NO concentrations through two windows W_3 and W_2 in special L-head SI engine (W_2 is closer to spark than W_3). The asterisks mark estimated initial conditions and flame arrival times. The dashed lines are calculated rate-limited concentrations for portions of the charge burning at these flame arrival times, with zero initial NO concentration. [10]

The most important engine variables that affect NO emissions are the relative air/fuel ratio (or fuel/air equivalence ratio), the burned gas fraction of the *in-cylinder* unburned mixture, and spark timing. The burned gas fraction depends on the amount of diluent such as recycled exhaust gas (EGR) used for NO_x emissions control, as well as the residual gas fraction. Fuel properties will also affect burned gas conditions; the effect of normal variations in gasoline properties is modest, however. The effect of variations in these parameters can be explained with the NO formation mechanism described above: changes in the time history of temperature and oxygen concentration in the burned gases during the combustion process and early part of the expansion stroke are the important factors. [11]

Equivalence Ratio

Figure 11.8 shows the effect of variations in the fuel/air ratio on NO emissions. Maximum burned gas temperatures occur at $\lambda \approx 0.9$; however, at this relative air/fuel ratio oxygen concentrations are low. As the mixture is further enriched, burned gas temperatures fall. As the mixture is leaned out, increasing oxygen concentration initially offsets the falling gas temperatures and NO emissions peak at $\lambda \approx 1.1$. Detailed predictions of NO concentrations in the burned gases suggest that the concentration versus time histories under fuel-lean conditions are different in character from those for fuel-rich conditions. In lean mixtures, NO concentrations freeze early in the expansion process and little NO decomposition occurs. In rich mixtures, substantial NO decomposition occurs from the peak concentrations present when the cylinder pressure is a maximum. Thus in lean mixtures, gas conditions at the time of peak pressure are especially significant. [7]

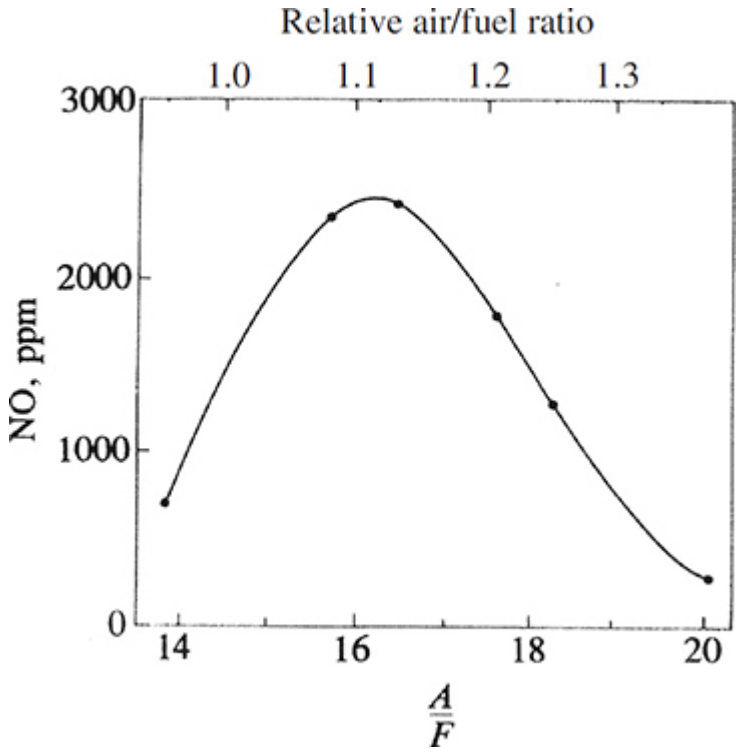


Figure 11.8 Variation of exhaust (engine-out) NO concentration with *A/F* and relative air/fuel ratio. Spark-ignition engine, 1600 rev/min, n_v = 50%, MBT timing. [12]

Burned Gas Fraction

The unburned mixture in the cylinder contains fuel vapor, air, and burned gases. The burned gases are residual gas from the previous cycle and any exhaust gas recycled to the intake for NO_x emissions control. The residual gas fraction is influenced by load, valve timing (especially the extent of valve overlap), and, to a lesser degree, by speed, air/fuel ratio, and compression ratio as described in Sec. 6.4 . The burned gases act as a diluent in the unburned mixture; the absolute temperature reached after combustion varies inversely with the burned gas mass fraction. Hence, increasing the burned gas fraction reduces NO emissions levels. However, it also reduces the combustion rate and, therefore, makes stable combustion more difficult to achieve (see Sec. 9.4).

Figure 11.9 shows the effect of increasing the burned gas fraction by recycling exhaust gases to the intake system just below the throttle plate. Substantial reductions in NO concentrations are achieved with 10 to 15%

EGR, which is about the maximum amount of EGR the engine will tolerate under normal part-throttle conditions. Of course, increasing the EGR at fixed engine load and speed increases the inlet manifold pressure, while fuel flow and airflow remain approximately constant.

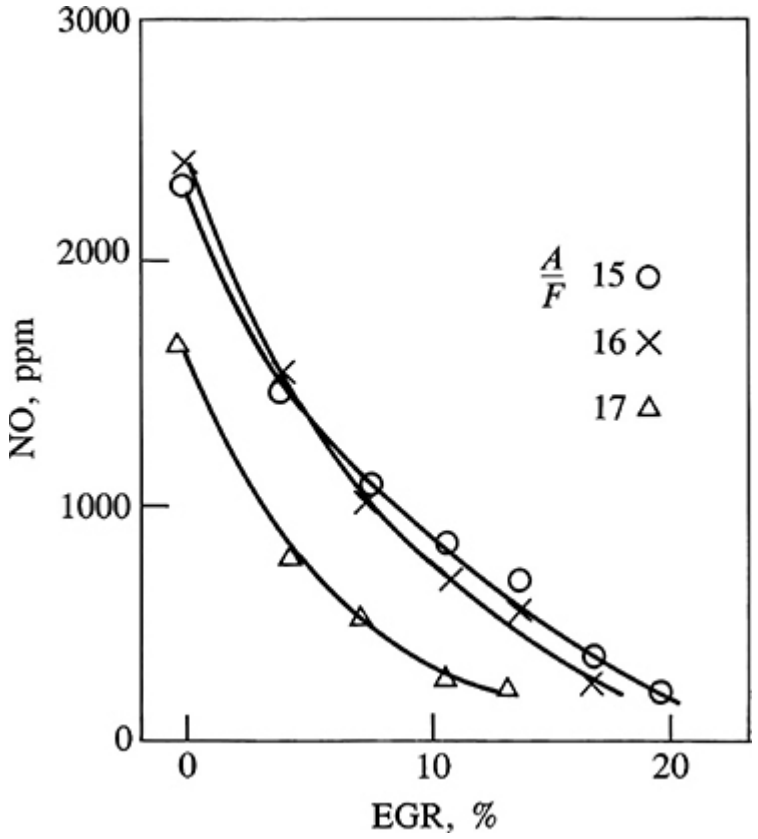


Figure 11.9 Variation of exhaust (engine out) NO concentration with percent recycled exhaust gas (EGR). Spark-ignition engine, 1600 rev/min, h_v = 50%, MBT timing. [12]

The primary effect of the burned gas diluent in the unburned mixture on the NO formation process is that it reduces flame temperatures by increasing the heat capacity of the cylinder charge, per unit mass of fuel. Figure 11.10 shows the effect of different diluent gases added to the engine intake flow, in a single-cylinder engine operated at constant speed, fuel flow, and air flow.[13] The data in Fig. 11.10*a* show that equal volume percentages of the different diluents produce different reductions in NO emissions. The same data when

plotted against diluent heat capacity (diluent mass flow rate × specific heat, c_p) collapse to close to a single curve.

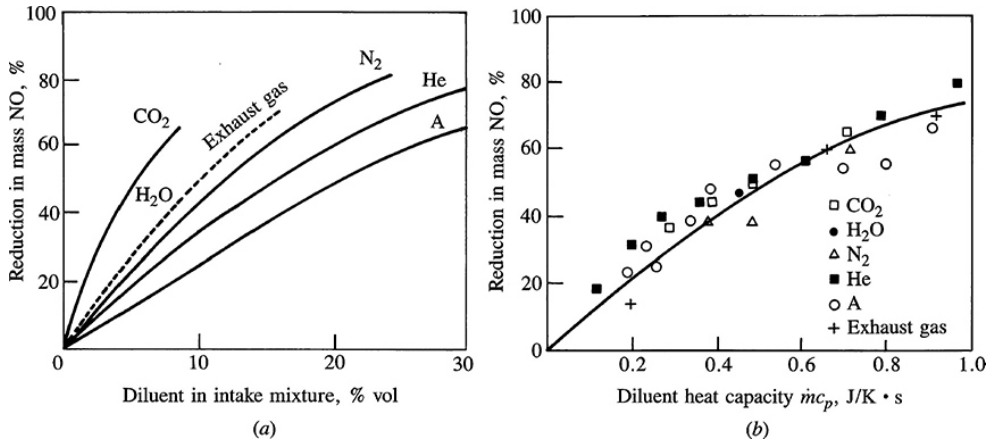


Figure 11.10 (*a*) Percentage reduction in mass NO emissions with various diluents. (*b*) Correlation of NO reduction with diluent heat capacity. Spark-ignition engine operated at 1600 rev/min, constant brake load (intake pressure ~0.5 atm), with MBT spark timing. [13]

A similar study where the burned gas fraction in the unburned charge was varied by changing the valve overlap, compression ratio, and EGR, separately, showed that, under more realistic engine operating conditions, it is the heat capacity of the total diluent mass in the in-cylinder mixture that is important. Whether the diluent mass is changed by varying the valve overlap, EGR, or even the compression ratio, was not important. [14] Application of the ideal gas law to the in-cylinder burned gases, first without EGR and then with EGR, at the same crank angle, yields a pair of equations

$$pV = M_{\text{noEGR}} RT_{\text{noEGR}}; \quad pV = M_{\text{EGR}} RT_{\text{EGR}} \tag{11.18}$$

Together, these illustrate the impact of adding EGR at constant engine torque. At the same crank angle (say close to the end of combustion), the volumes will be the same and the pressures will be closely comparable for the same torque. The gas constant will be little affected by the EGR since the burned gas molecular weight changes are small. With 10% EGR, the in-cylinder mass will be some 10% higher since, at constant torque, the fuel plus air mass will be little different. Thus, temperatures will be about 10% lower

which is a significant reduction (see Fig. 11.4). Note that with exhaust gas as the diluent, the impact on NO is through the temperature reduction resulting from the increased in-cylinder mass with EGR, for essentially the same fuel mass. The change in specific heat per unit mass is modest.

Excess Air and EGR

Because of the above, it is possible to correlate the influence of engine operating variables (such as air/fuel ratio, EGR, and engine speed and load) on NO emissions with two parameters which define the in-cylinder mixture composition: the relative air/fuel ratio (or the fuel/air equivalence ratio) and the gas/fuel ratio. The gas/fuel ratio (G/F) is calculated from the air plus residual plus EGR mass by

$$\frac{G}{F} = \frac{\text{total mass in cylinder}}{\text{fuel mass in cylinder}} = \frac{A}{F}\left(1 + \frac{x_b}{1 - x_b}\right) \tag{11.19}$$

where x_b is the burned gas fraction [Eq. (4.3)]. These together define the relative proportions of fuel, air, and burned gases in the in-cylinder mixture, and hence will correlate NO emissions. [e] Figure 11.11 shows the correlation of specific NO emissions, from a four-cylinder SI engine, over a range of operating conditions as a function of air/fuel ratio and gas/fuel ratio. Lines of constant air/fuel ratio and volumetric efficiency are shown; the direction of increasing dilution with residual gas and EGR at constant air/fuel ratio is downward and to the right. Excessive dilution results in poor combustion quality, partial burning, and, eventually, misfire (see Sec. 9.4.3). Lowest NO emissions consistent with good fuel consumption (avoiding the use of rich mixtures) are obtained with a stoichiometric mixture, with as much dilution as the engine will tolerate without excessive deterioration in combustion quality. [15] Note that increasing mixture dilution (through higher residual or EGR) at the stoichiometric A/F decreases NO emissions faster than does leaner operation (increasing A/F above the stoichiometric value of about 14.6) at constant residual plus EGR. This is largely due to the fact that when lean, a portion of the residual and EGR can be thought of as “air” since the burned gases contain oxygen.

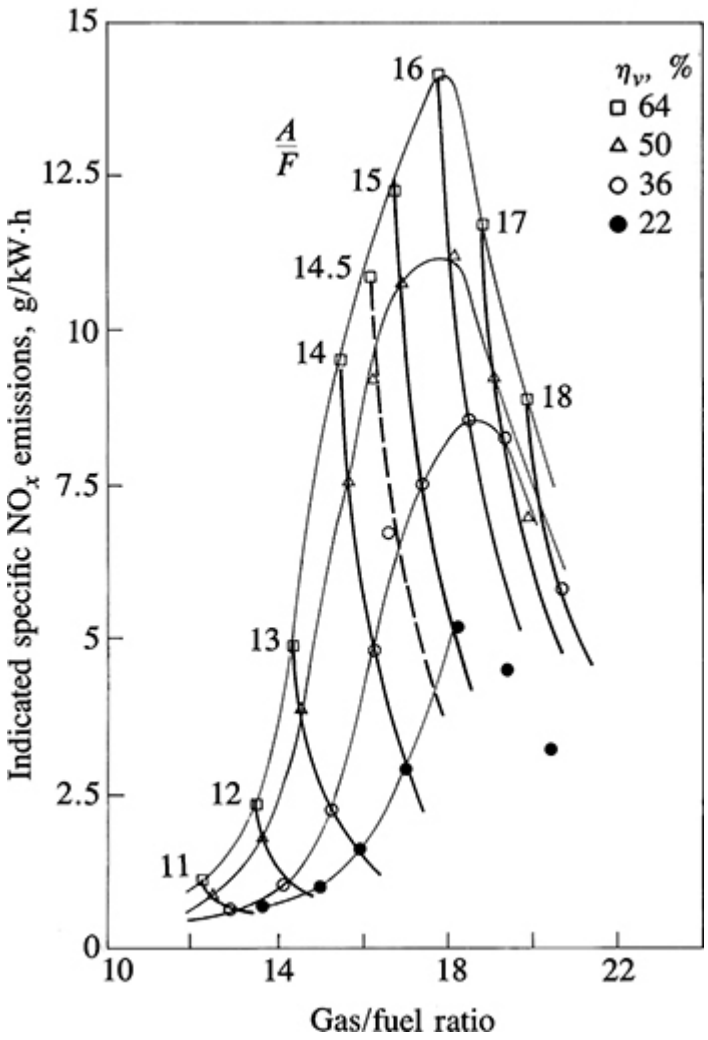


Figure 11.11 Correlation between gas/fuel ratio (G/F) and indicated specific NO_x emissions at various air/fuel ratios (A/F) and volumetric efficiencies (n_v). Spark-ignition engine operated at 1400 rev/min with spark timing retarded to give 0.95 of maximum brake torque. [15]

Comparisons between predictions made with the NO formation model (described at the beginning of this section) and experimental data show good agreement with normal amounts of dilution. [16] With extreme dilution, at engine-out NO levels of 100 ppm or less, the NO formed within the flame reaction zone cannot always be neglected. Within the flame, the concentrations of radicals such as O, OH, and H can be substantially in excess of equilibrium levels, resulting in much higher formation rates within the flame than in the post-flame gases. It is believed that the mechanism

[reactions (11.1) to (11.3)] and the formation rate equation (11.6) are still valid. However, neglecting the faster flame-front-formation rate of NO due to high concentrations of O, OH, and H may no longer be an appropriate assumption. [17]

Spark Timing

Spark timing significantly affects NO emission levels. Advancing the timing so that combustion occurs earlier in the cycle increases the peak cylinder pressure (because more fuel is burned before TC and the peak pressure moves closer to TC where the cylinder volume is smaller); retarding the timing decreases the peak cylinder pressure (because more of the fuel burns after TC). Higher peak cylinder pressures result in higher peak burned gas temperatures, and hence higher NO formation rates. For lower peak cylinder pressures, lower NO formation rates result. Figure 11.12 shows typical NO engine-out emission data for a spark-ignition engine as a function of spark timing. NO emission levels steadily decrease as spark timing is retarded from maximum brake torque (MBT) timing and moved closer to TC. Since exact determination of MBT timing is difficult (and not critical for fuel consumption and torque where the variation with timing around MBT is modest), there is always some uncertainty in NO emissions at MBT timing. Often, therefore, an alternative reference timing is used, where spark is retarded from MBT timing to the point where torque is decreased by 1 or 2% from the maximum value (usually about 5 degrees retard). Great care with spark timing is necessary to obtain accurate NO emissions measurements under MBT-timing operating conditions.

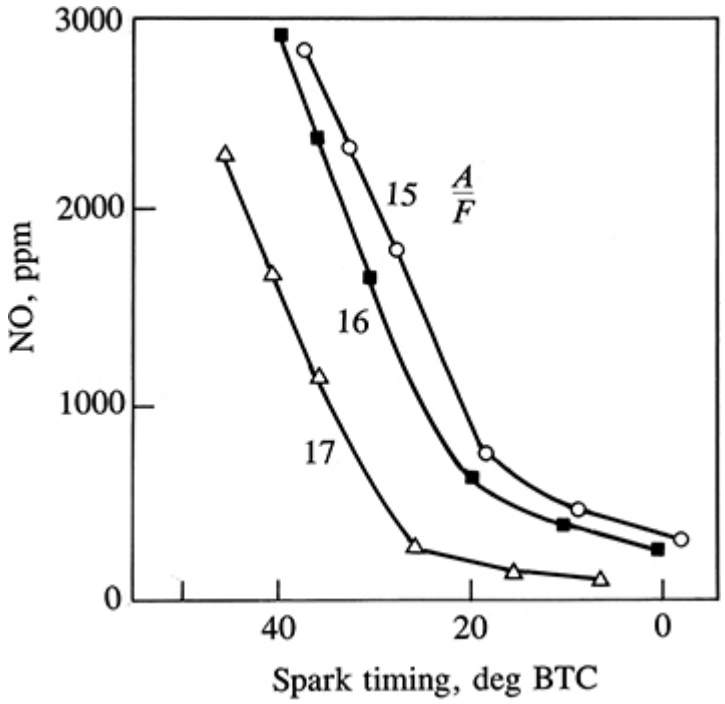


Figure 11.12 Variation of exhaust (engine-out) NO concentration with spark retard. 1600 rev/min, $n_v = 50\%$; left-hand end of curve corresponds to MBT timing for each *A/F*. [12]

11.2.4 NO_x Formation in Compression-Ignition Engines

The kinetic mechanisms for NO and NO_2 formation described in Secs. 11.2.1 and 11.2.2 and the assumptions made regarding equilibration of species in the C-O-H system apply to diesels as well as to spark-ignition engines. The critical difference, of course, is that injection of fuel into the cylinder occurs just before combustion starts, and that nonuniform burned gas temperature and composition result from this nonuniform fuel distribution during the diesel combustion process. The fuel-air mixing and combustion processes are extremely complex. During the "premixed" combustion phase immediately following the ignition delay, rich fuel-air mixture with a fuel-air equivalence ratio between 2 and 4 reacts due to spontaneous ignition within each diesel fuel spray. During the mixing-controlled combustion phase, the burning mixture is close to stoichiometric (the flame structure is that of a turbulent diffusion flame). However, throughout the combustion process

mixing between fuel vapor, air, and lean, stoichiometric, and rich burned mixture is occurring, thus changing the composition of gas elements in the cylinder. In addition to these composition (and hence temperature) changes due to mixing, temperature changes due to compression and expansion occur as the cylinder pressure rises and falls.

The discussion in Sec. 11.2.1 showed that the critical equivalence ratio for NO formation in high-temperature high-pressure burned gases is close to stoichiometric. Figure 11.4 is relevant here: it shows the initial NO formation rate in combustion products formed by burning a mixture of a typical hydrocarbon fuel with air (initially at 700 K, at a constant pressure of 15 bar). NO formation rates are high, and within a factor of 2 of the maximum value for $0.85 \lesssim \phi \lesssim 1.1$. Thus little NO will form during the fuel-rich premixed burning phase; almost all the NO will form in the mixing-controlled diffusion-flame burning process. In this diffusion flame, which surrounds each fuel spray, the fuel-rich partially reacted mixture diffuses into the flame from inside the spray and air diffuses into the flame from the outside. Chemical reaction is centered where the mixture composition is close to stoichiometric, producing the highest NO formation rates. These burned gases then diffuse away from the flame front, both inward into the fuel-rich spray, and outward into the surrounding air (see Fig. 10.5).

The critical time period is when burned gas temperatures are at a maximum. This will be close to the time of peak cylinder pressure. As combustion proceeds and cylinder pressure increases, unburned mixture temperatures will rise. After the time of peak pressure, burned gas temperatures will decrease as the cylinder gases expand. The decreasing temperature due to expansion *and* due to mixing of the high-temperature gases with air slows down and then freezes the NO chemistry. This second effect (which occurs only in the diesel) means that freezing occurs more rapidly than in the spark-ignition engine, and less decomposition of the NO occurs.

Figure 11.13 illustrates the above discussion of the diesel's NO formation process. It shows profiles of NO concentration, gas temperature, oxidizer concentration, and NO formation rate, with distance, moving outward from the center of the diffusion flame surrounding each diesel fuel spray into the outside air. The peak NO formation rate occurs slightly lean of stoichiometric, (see Fig. 11.4), and dominates the production process. [18]

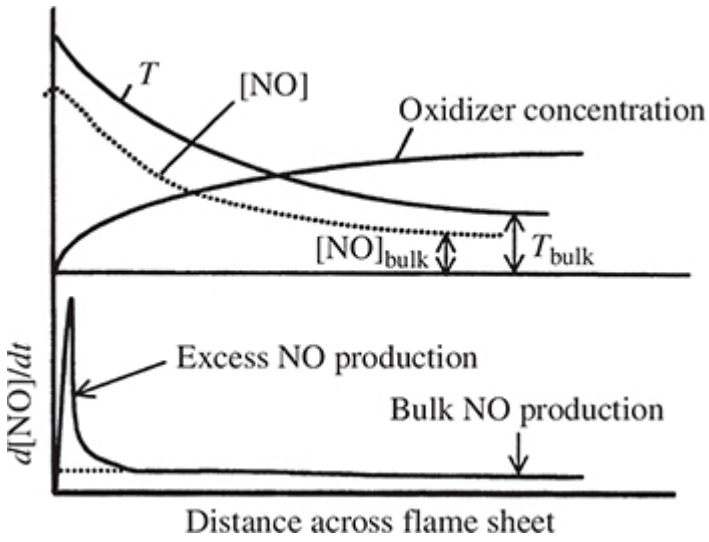


Figure 11.13 Schematic of NO formation in the diffusion flame-sheet surrounding each diesel fuel spray, with temperature, oxygen, and NO concentration profiles. Rich side of flame sheet on left; bulk air in cylinder on right. [18]

The above description is supported by species concentration data obtained from experiments where gas was sampled from within the cylinder of a normally operating diesel engine and analyzed. Figure 11.14 shows time histories of the major species concentrations, through the combustion process, determined with a rapid-acting sampling valve (1 ms open time) in a quiescent *direct-injection diesel* engine. Concentrations at different positions in the combustion chamber were obtained; the sample valve location for the Fig. 11.14 data is shown. Local NO concentrations rise from the residual gas value following start of combustion, to a peak at the point where the local burned gas equivalence ratio transitions from rich to lean (where the CO_2 concentration has its maximum value). As the local burned gas equivalence ratio becomes leaner due to mixing with excess air, NO concentrations decrease primarily due to dilution with air since formation becomes much slower as dilution occurs. At the time of peak NO concentrations within the bowl (15° ATC), most of the bowl region temperatures fall.

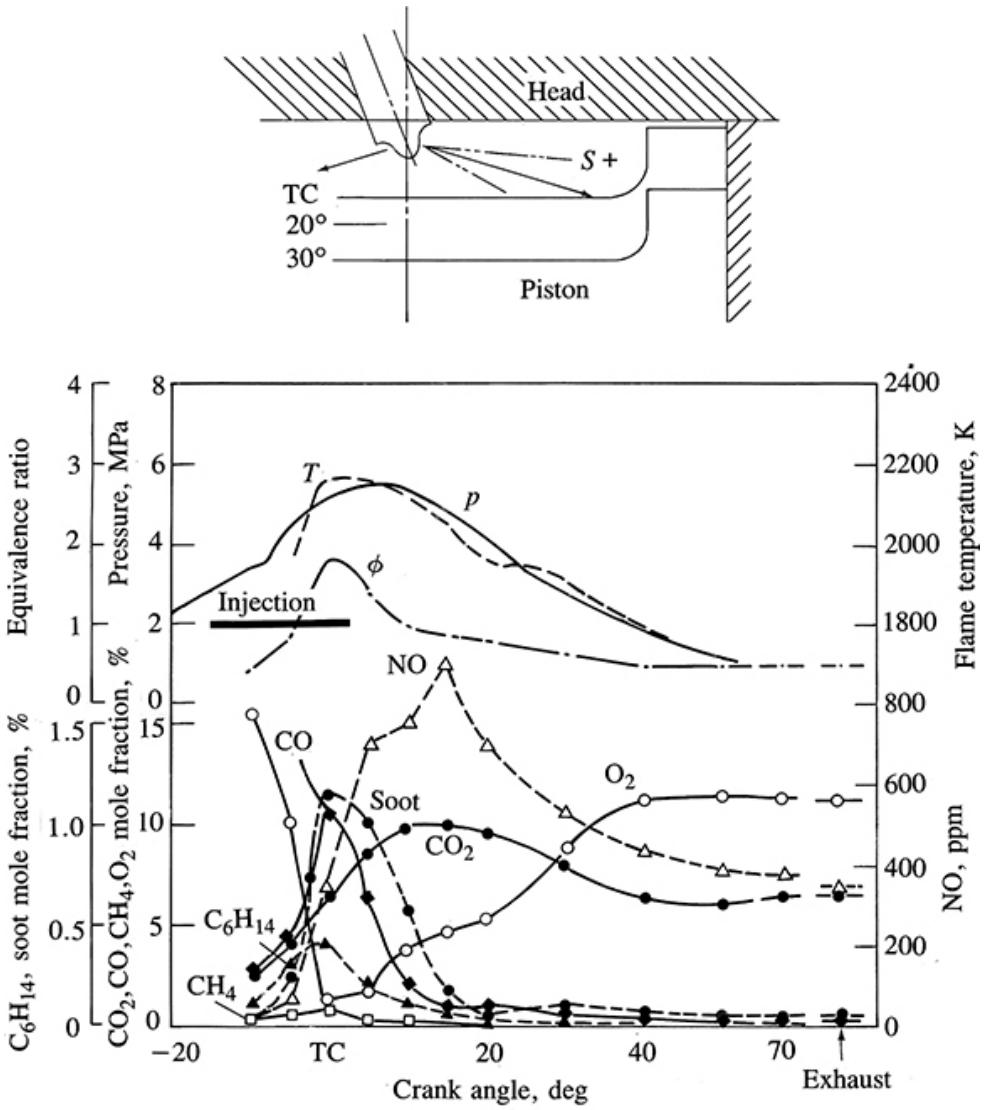


Figure 11.14 Concentrations of soot, NO, and other combustion product species measured at outer edge of the bowl-in-piston combustion chamber (location *S*) of a quiescent DI diesel with a rapid sampling valve. Cylinder gas pressure p, mean temperature T, and local equivalence ratio ϕ are also shown. Bore = 95 mm, stroke = 110 mm, r_c = 14.6. Four-hole nozzle with hole diameter = 0.2 mm. [20]

The total amount of NO within the cylinder of this type of direct-injection diesel during the NO formation process has also been measured. [19] At a predetermined time in one cycle, once steady-state warmed-up engine operation had been achieved, the contents of the cylinder were dumped into

an evacuated tank by rapidly cutting open a diaphragm which had previously sealed off the tank system. Figure 11.15 shows how the ratio of the average cylinder NO concentration divided by the exhaust concentration varies during the combustion process. NO concentrations reach a maximum shortly after time of peak pressure. There is a modest amount of NO decomposition. Variations in engine speed have little effect on the shape of this curve. The 20 crank-angle degrees after the start of combustion is the critical time period.

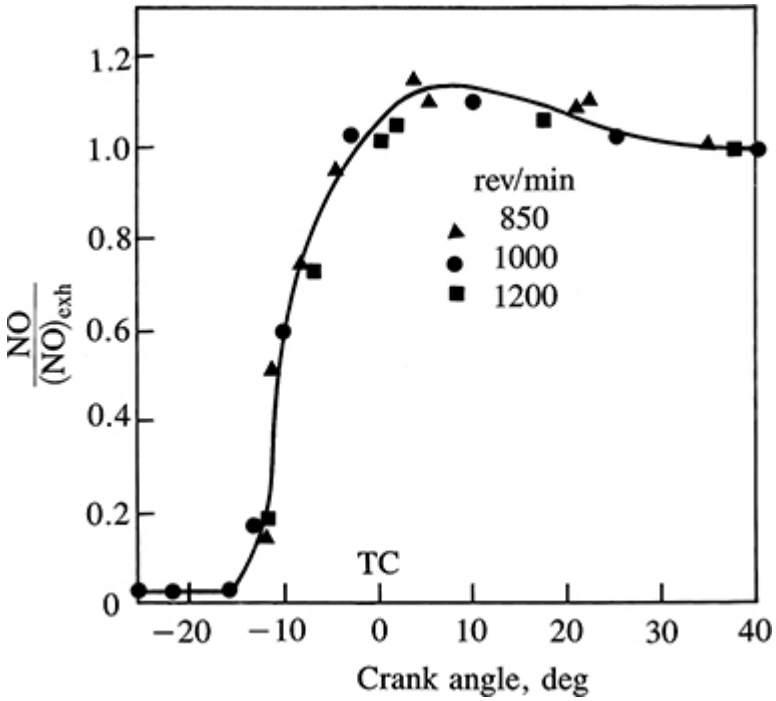


Figure 11.15 Ratio of cylinder-average NO concentration at given crank angle (determined from cylinder-dumping experiments) to exhaust NO concentration. DI diesel, equivalence ratio = 0.6, injection timing at 27° BTC. [19]

Results from similar cylinder-dumping experiments where injection timing and load (defined by the overall equivalence ratio) were varied also showed that almost all of the NO forms within 20° following the start of combustion. As injection timing is retarded, so the combustion process is retarded: NO formation occurs later, and concentrations are lower since peak temperatures are lower. The effect of the overall equivalence ratio, or load, on NO_x concentrations is shown in Fig. 11.16. At high load, with higher peak pressures, and hence temperatures, and larger area and longer lasting

regions of close-to-stoichiometric diffusion-flame-produced burned gas, NO levels increase. Both NO and NO_x concentrations were measured; NO_2 is 10 to 20% of total NO_x (see also Fig. 11.5 *b*). Though NO levels decrease with a decreasing overall equivalence ratio, they do so much less rapidly than do spark-ignition engine NO emissions (see Fig. 11.8) since, in the diesel, the zones where NO formation rates are highest are stoichiometric. Though the amount of fuel injected decreases proportionally as the overall equivalence ratio is decreased, almost all the fuel still burns close to stoichiometric. Thus NO emissions should be roughly proportional to the mass of fuel injected, provided burned gas pressures and temperatures in the diffusion flame do not change greatly.

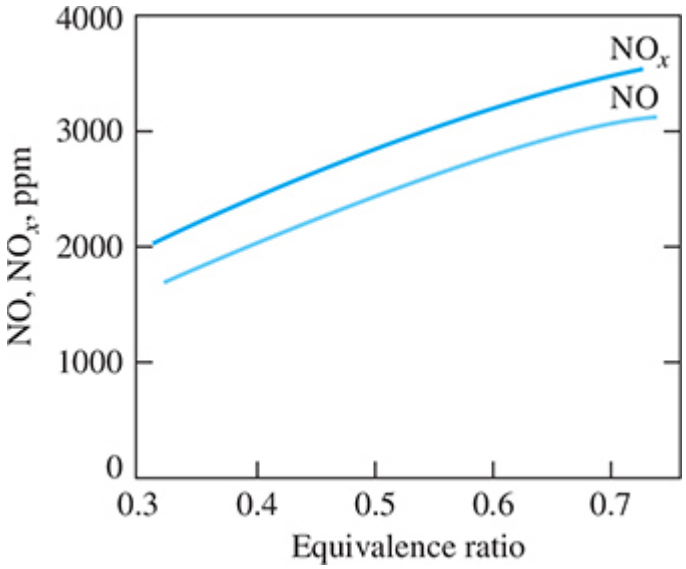


Figure 11.16 Exhaust NO_x and NO concentrations as a function of overall equivalence ratio or engine load. DI diesel, 1000 rev/min, injection timing at 27° BTC. [19]

Diluents added to the intake air (such as recycled exhaust) are effective at reducing the NO formation rate and NO_x emissions. As with spark-ignition engines, the effect is primarily one of reducing the burned gas temperature for a given mass of fuel and oxygen burned. Figure 11.17 *a* shows the effect of dilution of the intake air with N_2, CO_2, and exhaust gas on NO_x exhaust levels.[21] The heat capacity of CO_2 (per mole) at the temperatures relevant to

diesel combustion is about twice that of N_2. That of exhaust gas is slightly higher than that of N_2. Thus these data show that the effect is primarily one of reduced burned gas temperatures. Note that the composition of the exhaust gas of a diesel varies with load. At idle, there is little CO_2 and H_2O, and the composition does not differ much from that of air. At high load the diluent capacity (mass times specific heat of inert gas) increases since the concentrations of CO_2 and H_2O in the recycled exhaust are substantially higher. Figure 11.17 *b* shows the effect of increasing EGR at higher, mid-, and lower-loads.[22] The engine can tolerate high EGR levels (50%) at low load since at very lean operating condition ($\phi \sim 0.2$), a major fraction of this EGR is effectively "air." As load and overall equivalence ratio increases, the mixture becomes less lean, and a larger fraction of the EGR is effectively "stoichiometric burned gas" and less is "air." Thus EGR behavior in the diesel combustion process is significantly different than that in a spark-ignition engine. Only a portion of the EGR acts as an inert diluent in the close-to-stoichiometric region of the diffusion flame where almost all of the NO is formed.

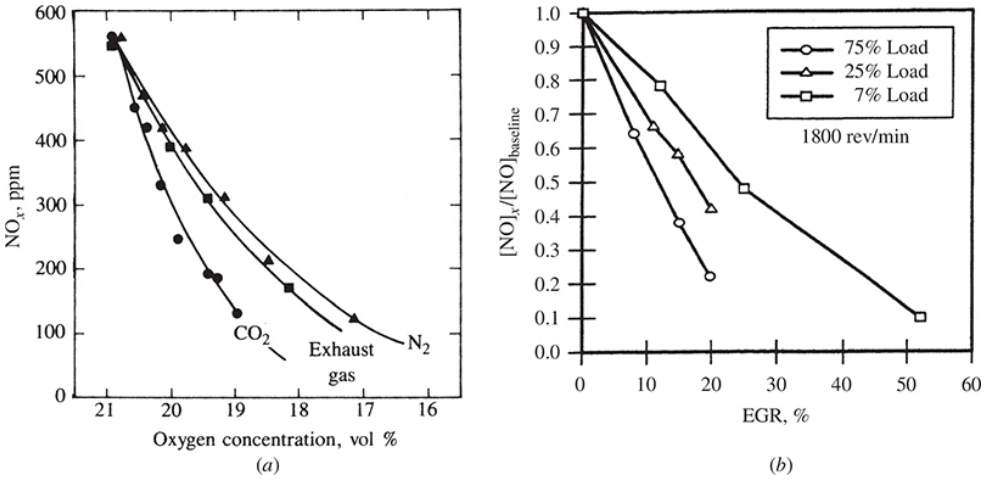


Figure 11.17 (*a*) Effect of reduction in oxygen concentration by different diluents (exhaust gas, CO_2, N_2) on NO_x emissions in DI diesel. Bore = 140 mm, stroke = 152 mm, r_c = 14.3, speed = 1300 rev/min, fuel rate = 142 mm^3/stroke, injection timing at 4° BTC.[21] (*b*) Effect of recycled exhaust gas (EGR) at light, mid, and high loads: Heavy-duty diesel engine at 1800 rev/min.[22]

Confirmation that NO forms in the close-to-stoichiometric burned gas regions, and that the magnitude of the stoichiometric burned gas temperature controls NO_x emissions, is given by the following. Plee et al . [23,24] have shown that the effects of changes in intake gas composition (with EGR, nitrogen, argon, and oxygen addition) and temperature on NO_x emissions can be correlated by

$$EI_{NO_x} = \text{constant} \times \exp\left(\frac{E}{\tilde{R}T_f}\right) \tag{11.20}$$

T_f (kelvins) is the stoichiometric adiabatic flame temperature (evaluated at a suitable reference point: fuel-air mixture at top-center pressure and air temperature) and E is an overall activation energy. Figure 11.18 shows EI_{NO_x} for a range of intake air compositions and temperatures, from two DI engines for several loads and speeds, normalized by the engine NO_x level obtained for standard air, plotted on a log scale against the reciprocal of the stoichiometric adiabatic flame at TC conditions. A single value of $E/\tilde{R}$ correlates EI_{NO_x} over more than an order of magnitude. [23] While there is some scatter since the model used is overly simple, and load, speed, and other engine design and operating parameters affect the process, the overriding importance of the close-to-stoichiometric diffusion-flame burned gas temperature is clear.

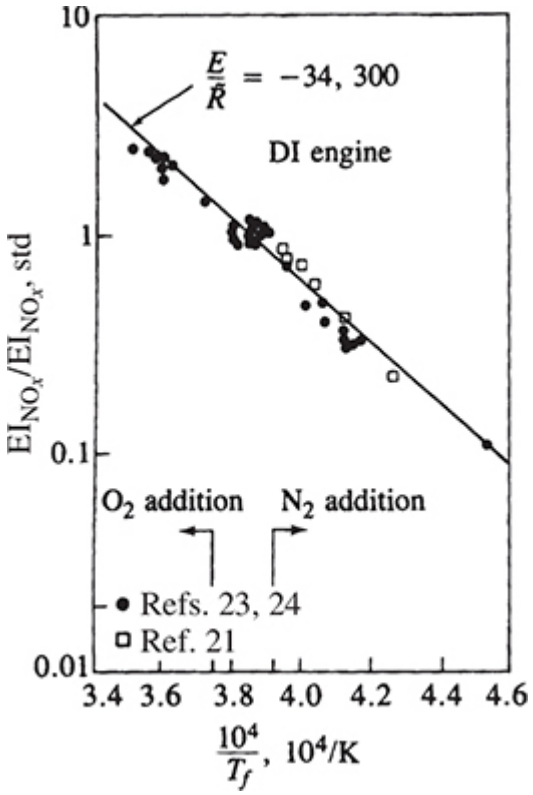


Figure 11.18 Correlation of NO_x emissions index EI_{NO_x} for a wide range of operating conditions with the reciprocal of the stoichiometric mixture flame temperature for DI diesel engines: Flame temperatures varied by addition of different diluents and oxygen.[21,23,24] Values of EI_{NO_x} normalized with value at standard conditions.

11.3 CARBON MONOXIDE

Carbon monoxide (CO) emissions from internal combustion engines are controlled primarily by the relative air/fuel ratio or fuel/air equivalence ratio. Figure 11.19 shows CO levels in the exhaust of a spark-ignition engine for several different fuel compositions.[25] When the data are plotted against the relative air/fuel ratio, they are correlated by a single curve. For fuel-rich mixtures CO concentrations in the exhaust increase steadily with decreasing relative *A/F* ratio, as the amount of excess fuel increases. For fuel-lean mixtures, CO concentrations in the exhaust vary little with relative *A/F* and are low, of order 10^{-3} mole fraction.

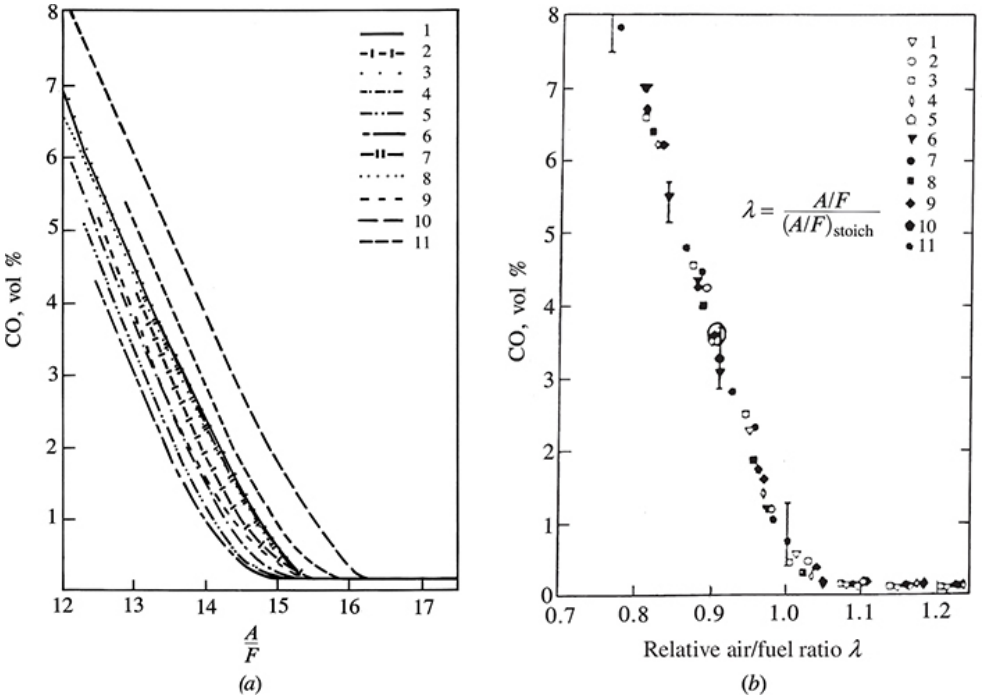


Figure 11.19 Variation of SI engine CO emissions with 11 fuels of different H/C ratio: (*a*) with air/fuel ratio and (*b*) with relative air/fuel ratio λ.[25]

Since spark-ignition engines often operate close to stoichiometric at part load and, depending on calibration, may be fuel-rich at high load (see Sec. 7.1), CO emissions are significant and need to be controlled. Diesels, however, always operate well on the lean side of stoichiometric; CO emissions from diesels are low enough to be unimportant, and will not be discussed further.

The levels of CO observed in spark-ignition engine exhaust gases are lower than the maximum values measured within the combustion chamber, but are significantly higher than equilibrium values for the exhaust conditions. Thus the processes which govern CO exhaust levels are kinetically controlled. In a premixed hydrocarbon-air flame, the CO concentration increases rapidly in the flame zone to a maximum value, which is larger than the equilibrium value for adiabatic combustion of the fuel-air mixture. CO formation is one of the principal reaction steps in the hydrocarbon combustion mechanism, which may be summarized by [1]

$$RH \rightarrow R \rightarrow RO_2 \rightarrow RCHO \rightarrow RCO \rightarrow CO \qquad (11.21)$$

where R stands for the hydrocarbon radical. The CO formed in the combustion process via this path is then oxidized to CO_2 at a slower rate. The principal CO oxidation reaction in hydrocarbon-air flames is

$$CO + OH = CO_2 + H \tag{11.22}$$

The rate constant for this reaction is [1]

$$k_{CO}^{+} = 6.76 \times 10^{10} \exp\left(\frac{T}{1102}\right) \text{ cm}^3/\text{gmol} \tag{11.23}$$

It is generally assumed that in the post-flame combustion products in a spark-ignition engine, at conditions close to peak cycle temperatures (2500 to 2800 K) and pressures (15 to 40 atm), the carbon-oxygen-hydrogen system is equilibrated. Thus CO concentrations in the immediate post-flame burned gases are close to equilibrium. However, as the burned gases cool during the expansion and exhaust processes, depending on the temperature and cooling rate, the CO oxidation process [reaction (11.22)] may not remain locally equilibrated.

Newhall carried out a series of kinetic calculations for an engine expansion stroke assuming the burned gas at the time of peak cylinder pressure was uniform and in equilibrium. [26] Of the reactions important to CO chemistry, only three-body radical-recombination reactions such as

$$H + H + M = H_2 + M \tag{11.24}$$

$$H + OH + M = H_2O + M \tag{11.25}$$

$$H + O_2 + M = HO_2 + M \tag{11.26}$$

were found to be rate controlling. The bimolecular exchange reactions and the CO oxidation reaction (11.22) were sufficiently fast to be continuously equilibrated. Only during the later stages of the expansion stroke was the CO concentration predicted to depart from equilibrium, as shown in Fig. 11.20. Using this technique to predict average CO levels at the end of expansion, over a range of equivalence ratios (rich to lean), Newhall [26] obtained a good

match to experimental data (see Fig. 11.21). The kinetically controlled aspects of the CO emissions mechanism have thus been demonstrated.

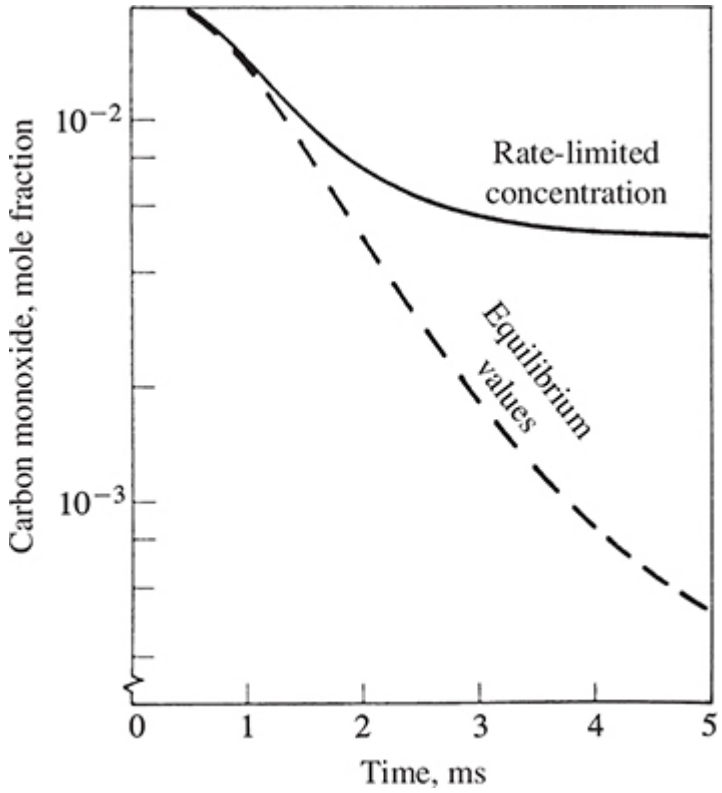


Figure 11.20 Results of kinetic calculations of CO concentrations, and equilibrium concentrations, during expansion stroke following TC combustion in SI engine; stoichiometric mixture.[26]

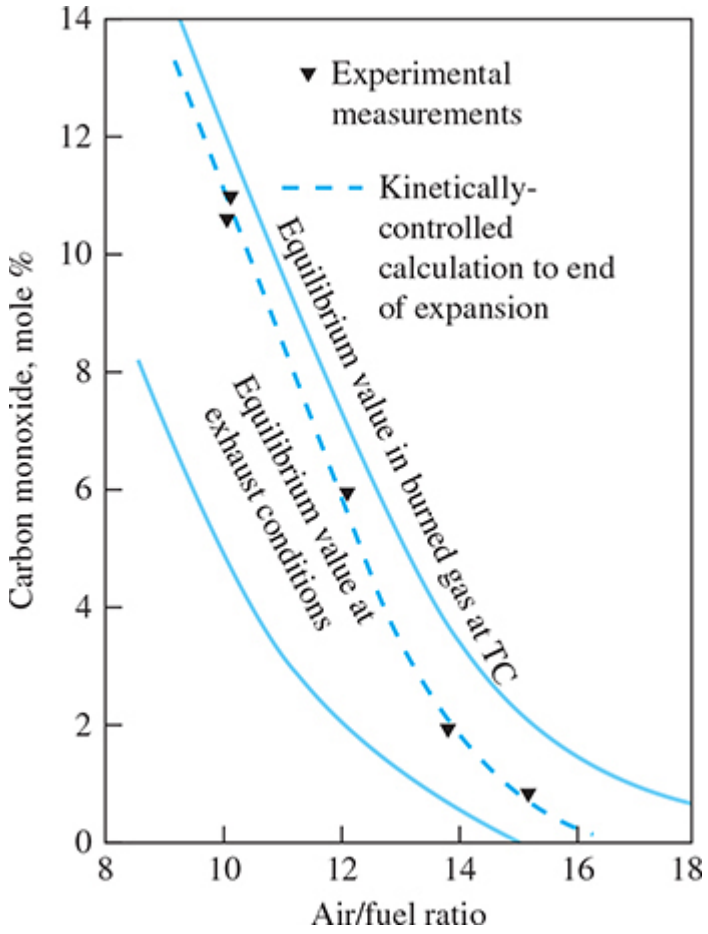


Figure 11.21 Predicted CO concentration at the end of expansion stroke, compared with measured exhaust concentrations, as function of air/fuel ratio. Equilibrium levels at TC combustion and exhaust conditions also shown. [26]

Conclusions from these detailed studies show that exhaust CO concentrations for fuel-rich mixtures are close to equilibrium concentrations in the burned gases during the expansion process. For close-to-stoichiometric mixtures, the CO engine-out emissions are much higher than CO equilibrium concentrations corresponding to exhaust conditions. For fuel-lean mixtures, measured CO emissions are substantially higher than predictions from kinetically-controlled analysis of the chemistry bulk gas chemistry. One possible explanation of this lean-mixture discrepancy is that only partial oxidation to CO may occur as some of the hydrocarbons emerging during expansion and exhaust from crevices in the combustion chamber, and from oil layers or deposits on the chamber walls, burn up.

As a practical matter, exhaust CO emissions are determined by the relative air/fuel ratio. The degree of control achieved within the engine to date has come from improving mixture uniformity and leaning-out the intake mixture. In multi-cylinder engines, because CO increases rapidly as the inlet mixture becomes richer than stoichiometric, cylinder-to-cylinder variations in equivalence ratio about the mean value are important: nonuniform distribution of fuel and air about stoichiometric increases average emissions. Thus improved cylinder-to-cylinder fuel/air ratio distribution has become essential. Also, because it is necessary to enrich the intake mixture when the engine is cold, CO emissions during engine warm-up may be higher than emissions in the fully warmed-up state. Further, in transient engine operation during acceleration and deceleration, control of fuel metering has had to be improved. Major reductions in CO beyond what can be achieved in the engine are possible with exhaust catalysts, which are reviewed in Sec. 11.6 . Oxidation of CO in the exhaust system without use of special exhaust treatment devices does not normally occur to any significant extent because the exhaust gas temperature is too low: CO oxidation reactions effectively freeze as the burned gases pass into the exhaust system. [27]

11.4 HYDROCARBON EMISSIONS

11.4.1 Background

Hydrocarbons, or more appropriately organic emissions, are the consequence of incomplete combustion of the hydrocarbon fuel. The level of unburned hydrocarbons (HC) in the exhaust gases of gasoline fueled spark-ignition engines is typically 1 to 2% of the fuel: in diesels it is much less. The levels are generally specified in terms of the total hydrocarbon concentration expressed in parts per million carbon atoms (C_1). [f] While total hydrocarbon emission is a useful measure of combustion inefficiency, it is not necessarily an appropriate index of pollutant emissions impacts. Engine exhaust gases contain a wide variety of hydrocarbon compounds. Table 11.3 shows the average breakdown by class of the hydrocarbons in spark-ignition engine exhaust gases, both with and without a catalytic converter, with gasoline fuel. Some of these hydrocarbons are essentially inert

physiologically and some are virtually unreactive from the standpoint of photochemical smog. Others are highly reactive in the smog-producing chemistry. (Also, some hydrocarbons are known carcinogens; see Sec. 11.5.2). Based on their potential for oxidant formation in the photochemical smog chemistry, hydrocarbon compounds are characterized by their reactivity—their ozone forming potential in the atmosphere. Table 11.4 shows one reactivity scale which is commonly used to estimate the reactivity of exhaust gas hydrocarbon mixtures: the *Incremental Reactivity* scale. All hydrocarbons except methane react, given enough time. More detailed breakdowns of the composition of spark-ignition and diesel engine exhaust HC are available in the literature. The values in Table 11.4 for specific organic compounds are in gO $_3$ (the dominant oxidizing species present in photochemical smog) per gram of NMOG (non-methane organic gases) in the exhaust: the Maximum Incremental Reactive value. Ranges of values are given for the important classes of organic species. Lower molecular weight paraffinic compounds have the lowest values, less than or about 1. Olefins are highest, values in the 4 to 10 range. The aromatics that are in gasoline are also high, 2.7 to about 10. These are called *incremental reactivities* since they are weighted by the concentrations of the main individual hydrocarbon species in the hydrocarbon mix in engine or vehicle exhaust, and then summed to determine the overall atmospheric reactivity of the exhaust gas stream.

TABLE 11.3 Hydrocarbon composition of spark-ignition engine exhaust (by class) [29]

	Carbon, percent of total HC			
	Paraffins	Olefins	Acetylene	Aromatics
Engine-out emissions	33	27	8	32
Catalyst-out emissions	57	15	2	26

TABLE 11.4 Reactivity of different classes of hydrocarbons

Hydrocarbons	MIR value (gO_3/g NMOG)*
C_1–C_4 paraffins	0.25–1
C_5 and higher molecular weight paraffins	1–2
Cyclic paraffins	1–2.5
Ethylene	7.3
Olefins	4–10
Diolefins	10
Benzene	0.4
Toluene	2.7
Higher aromatics	6–10
Methanol	0.6
Ethanol	1.3
Formaldehyde	7.2
Acetaldehyde	5.5

*MIR: Maximum Incremental Reactivity values determined from standardized smog chamber testing. See Carter for more details.[30]

Fuel composition can significantly influence the composition and magnitude of the organic emissions. Fuels containing high proportions of aromatics and olefins produce relatively higher concentrations of reactive hydrocarbons. However, many of the organic compounds found in the exhaust are not present in the fuel, indicating that significant pyrolysis and synthesis occur during parts of the combustion process.

Oxygenates are present in engine exhaust, and are known to participate in the formation of photochemical smog. Some oxygenates are also irritants and odorants. Oxygenates are generally categorized as carbonyls, phenols, and other noncarbonyls. The carbonyls of interest are low-molecular-weight aldehydes and aliphatic ketones. The volatile aldehydes are eye and respiratory tract irritants. Formaldehyde is a major component ($\lesssim$20% of total carbonyls). Carbonyls account for about 10% of the HC emissions from diesel passenger car engines, but only a few percent of spark-ignition engine HC emissions. Phenols are odorants and irritants: levels are much lower than aldehyde levels. Other noncarbonyls include methanol, ethanol, nitromethane, methyl formate. Whether these are significant with hydrocarbon fuels is unclear. Use of alcohol fuels, as standalone fuels or blended with gasoline, increases oxygenate emissions. Both methanol and aldehyde emissions increase substantially above gasoline-fueled levels with methanol-fueled spark-ignition engines.

11.4.2 Flame Quenching and Oxidation Fundamentals

Flame quenching or extinction occurs at the walls of engine combustion chambers. The cool walls of the chamber act as a sink for the thermal energy and the active radical species generated within the flame. Quenching of the flame occurs under several different geometrical configurations: the flame may be propagating normal to, parallel to, or at an angle to the wall; the flame may quench at the entrance to a crevice—a thin volume with a narrow entrance to the combustion chamber, such as the region between the piston crown and the cylinder wall. When the flame quenches, it leaves a thin layer or volume of unburned mixture ahead of the flame. Whether this results in hydrocarbon emissions depends on the extent to which these quench region hydrocarbons can subsequently be oxidized.

Flame-quenching processes have been analyzed by relating the heat release within the flame to the heat loss to the walls under conditions where quenching just occurs. This ratio, called a Peclet number (Pe), is approximately constant for any given geometrical configuration. The simplest configuration for study is the two-plate quench process, where the minimum spacing between two parallel plates through which a flame will propagate is determined. The Peclet number for this two-plate configuration is given by:

$$\mathrm{Pe}_2 = \frac{\rho_u S_L c_{p,f}(T_f - T_u)}{k_f (T_f - T_u)/d_{q2}} = \frac{\rho_u S_L c_{p,f} d_{q2}}{k_f} \tag{11.27}$$

which is approximately constant over a wide range of conditions. ρ , S_L, c_p, T, and k are the density, laminar flame speed, specific heat at constant pressure, gas temperature, and thermal conductivity, respectively, with the subscripts u and f referring to unburned and flame conditions. $d_{q\,2}$ is the two-plate quench distance. The wall temperature and unburned gas temperature are assumed to be equal. This assumption is also appropriate in the engine context since there is ample time during the compression stroke for a thermal boundary layer to build up on the combustion-chamber walls to a thickness of at least the quench distance.

Lavoie [31] has developed empirical correlations for two-plate quench distance data for propane-air mixtures: only limited data for liquid

hydrocarbon fuels such as isooctane are available. The data in the pressure range of 3 to 40 atm are well fitted by

$$\mathrm{Pe}_2 = \frac{9.5}{\phi}\left(\frac{p}{3}\right)^{0.26\times\min(1,1/\phi^2)} \tag{11.28}$$

where p is the pressure in atmospheres and f is the fuel/air equivalence ratio. The two-plate quench distance $d_{q\,2}$ is then obtained from Eq. (11.27) and Prandtl number and viscosity relations for the flame conditions (see Sec. 4.8 or Ref. 31). Thus the minimum size crevice region into which a flame will propagate can be determined.

For the process of a flame front quenching on a single wall, there are many possible geometries. The simplest is where the flame front is parallel to the wall and approaches it head on. This one-wall quench distance $d_{q\,1}$, defined as the position of closest approach of the reaction zone to the wall, scales with flame properties in a similar way to the two-plate quench distance. Thus, a one-wall Peclet number relation can be formed:

$$\mathrm{Pe}_1 = \frac{\rho_u S_L c_{p,u} d_{q1}}{k_u} \approx 8 \tag{11.29}$$

where the subscript u denotes properties evaluated at unburned gas conditions.

Using the wall temperature as representative of the local unburned gas temperature (because the thermal boundary-layer thickness is greater than typical quench distances), Lavoie showed that

$$\frac{d_{q1}}{d_{q2}} = \frac{\mathrm{Pe}_1}{\mathrm{Pe}_2} = 0.2 \tag{11.30}$$

is a reasonable fit to the single-wall quench data. Single-wall quench distances are in the range of 0.04 to 0.15 mm. Typical two-wall quench distances for spark-ignition engine conditions are 0.2 to 0.6 mm; these distances indicate the minimum crevice size the flame will enter. However, the largest engine crevice, the gap between the piston crown (topland) and the cylinder liner, has a different geometry from either the two-wall or

single-wall quench situation. The piston crown is effectively a flat plate, and the liner is perpendicular to it. The flame is thus propagating into a "corner" and the liner would be expected to provide additionally local flame cooling. This would likely increase the quenching distance for the actual geometry above the two-wall quench distance value. More recent experimental data on crevice volume impacts on efficiency and HC emissions supports this discussion: see Fig. 11.27. [32]

While a fraction of the fuel hydrocarbons can escape the primary combustion process unburned, or only partially reacted, oxidation of some of these hydrocarbons can occur during the expansion and exhaust processes. Hydrocarbon oxidation rates have been determined in a number of different studies and several different empirical correlations of the data in the form of overall reaction rate equations have been proposed. A reasonable fit to the experimental data on unburned HC burnup is the rate expression: [31]

$$\frac{d[\mathrm{HC}]}{dt} = -6.7 \times 10^{15} \exp\left(\frac{-18{,}735}{T}\right) \tilde{x}_{\mathrm{HC}} \tilde{x}_{\mathrm{O}_2} \left(\frac{p}{RT}\right) \tag{11.31}$$

where [ ] denotes concentration in moles per cubic centimeter, $\tilde{x}_{\mathrm{HC}}$ and $\tilde{x}_{\mathrm{O}_2}$ are the mole fractions of HC and O_2, respectively, t is in seconds, T is in kelvins, and the density term (p/RT) has units of moles per cubic centimeter. The spread in the data about this equation is substantial, however.

Studies of combustion of premixed fuel-air mixtures at high pressure in closed vessels have been useful in identifying the mechanisms by which hydrocarbons escape complete combustion. The residual hydrocarbons left unburned in the vessel following a combustion experiment have been shown to come primarily from crevices in the vessel walls. Thus wall quench hydrocarbons apparently diffuse into the burned gases and oxidize following the quenching event. Analytical studies of the flame quenching process, and postquench diffusion and oxidation with kinetic models of the hydrocarbon oxidation process, are in agreement with this assessment. Flame quenching can be thought of as a two-stage process. The first step is the extinction of the flame a short distance from the cold wall, determined by a balance between conduction of thermal energy from the hot reaction zone to the wall and the thermal energy released in the reaction zone by the flame reactions. The second step is the post-quench HC diffusion and oxidation occurring on a

time scale of one or a few milliseconds after quenching. The diffusion and oxidation process ultimately reduces the mass of wall quench hydrocarbons to levels well below its value at the time of quenching.

Closed-vessel combustion experiments have also been used to show that oil layers on the walls of the vessel cause an increase in residual HC levels after combustion is complete. The additional HC that result in these experiments with oil films are primarily fuel molecules, and are proportional to the amount of oil placed on the walls of the reactor and the solubility of the specific fuel in that oil. Absorption of fuel in the oil occurs prior to ignition and the dissolved fuel is then desorbed into the burned gases after combustion is complete. Thus fuel absorption into and desorption from oil layers on the cylinder liner is a potentially important engine HC mechanism.[33]

11.4.3 HC Emissions from Spark-Ignition Engines

Engine-out hydrocarbon levels in the exhaust of a spark-ignition engine under normal operating conditions are typically in the range 1000 to 3000 ppm C_1. Normally, these HC emissions are between about 1 and 2% of the fuel flow into the engine. As indicated in Fig. 11.2, HC emissions rise steadily as the mixture becomes richer than stoichiometric, and level off at a relative (lean) *A/F* close to 1.15. With very lean mixtures, as combustion quality deteriorates, HC emissions rise rapidly due to incomplete combustion or misfire in a fraction of the engine's operating cycles. As outlined in Sec. 11.1, there are several sources that contribute to engine HC emissions. By sources, we mean any HC escaping the primary combustion process. Some of these HC may then fully or partly oxidize in the expansion and exhaust processes. While the complete HC emissions process is complex, there is sufficient information available to explain the overall process, and thus how major engine variables influence engine-out HC emission levels.

Several HC emissions formation mechanisms for spark-ignition engines (where the fuel-air mixture is essentially premixed) have been proposed. They are listed in Table 11.5, and illustrated in Fig. 11.22. The largest of these is the filling of crevice volumes with largely unburned mixture which, since the flame quenches near the crevice entrance, escapes the primary combustion process. All these processes together result in some 5 to 10% of

the fuel not burning during the normal premixed-flame combustion process. Thus, in addition to being the source of the engine's HC emissions, they constitute a significant loss of torque and efficiency. [32,34]

TABLE 11.5 Processes by which gasoline compounds escape burning during normal combustion

- Gasoline vapor-air mixture compressed into the combustion chamber crevice volumes
- Gasoline compounds absorbed in oil layers on the cylinder liner
- Gasoline absorbed by and/or contained within deposits on the cylinder head and piston crown
- Quench layers on the combustion chamber wall left as the flame extinguishes close to the wall
- Gasoline vapor-air mixture left unburned when the flame extinguishes prior to reaching the walls
- Liquid gasoline within the cylinder that does not evaporate and mix with sufficient air to burn prior to the end of combustion
- Leakage of unburned mixture through the (nominally) closed exhaust valve

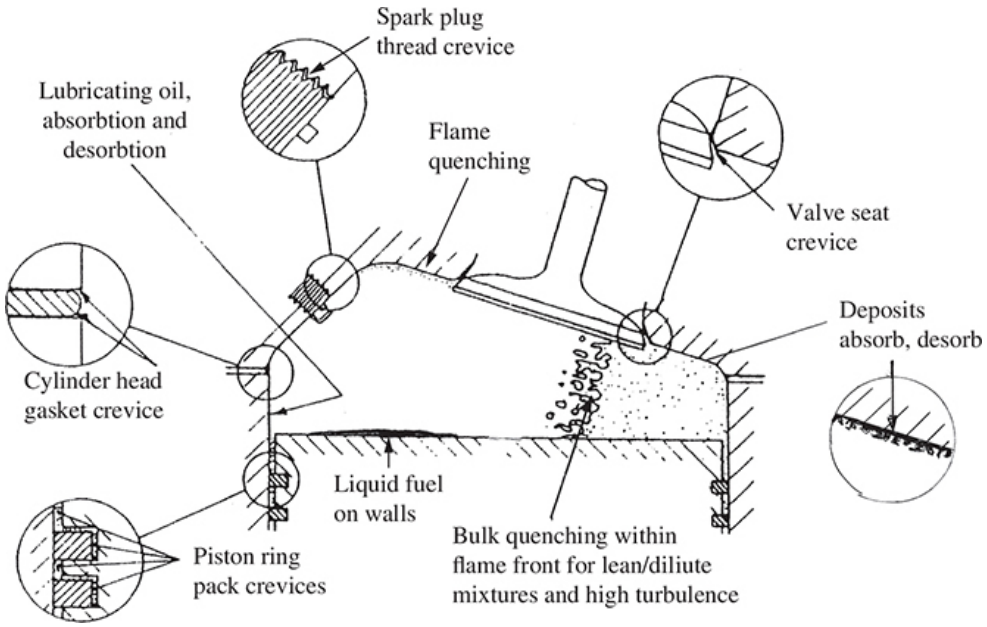


Figure 11.22 Schematic of SI engine cylinder head and piston during combustion identifying the sources of hydrocarbon emissions.

All these processes (except misfire) result in unburned hydrocarbons close to the combustion chamber walls rather than in the bulk of the in-cylinder gases. Thus, the distribution of these HC within the cylinder will not be uniform. For most of these sources, as combustion ends after peak pressure, these HC will flow out of the region where they were "stored", will diffuse into and mix with the hot burned gases in the bulk of the cylinder and, if their temperature becomes sufficiently high, will start to oxidize. Table 11.6 summarizes, for each of these sources the processes by which, the HC that escaped burning during the normal combustion process leave the crevices, oil layers, etc., mix with the burned gases, then oxidize (partially or fully) or exit the cylinder in the exhausting gases. They then may oxidize in the exhaust system, or become HC emissions. Figure 11.23 illustrates the total process leading to unburned HC emissions in the engine exhaust, from the various sources—mechanisms by which fuel escapes burning during the normal combustion process—through whether or not these HC oxidize in the cylinder or exhaust port, and if not, becoming engine-out HC emissions. The figure approximately quantifies the consequences of this many step process through the engine's (four-stroke) operating cycle by the width of each shaded bar. Engine-out HC are about one-third of the HC that escaped burning during combustion. Since engine-out HC [g] correspond to 1 to 2% of the fuel entering each cylinder each cycle, the loss of torque and efficiency is significant.

TABLE 11.6 Processes by which these unburned hydrocarbons can oxidize or exit the cylinder

- Outflow of unburned fuel-air mixture from crevice volumes; mixing with burned gases; some of the crevice HC will oxidize.
- Diffusion of hydrocarbon vapor out of oil layers and deposits into the burned gases; some of these HC will oxidize.
- Mixing of wall and bulk quench gases with burned gases; some will oxidize.

- Exhaust blowdown process will carry a fraction of the in-cylinder unburned HC into the exhaust.
- The displacement of gases by the piston during the exhaust stroke will transport an additional fraction of the in-cylinder unburned HC into the exhaust.
- The unburned HC, which exit the cylinder, will mix with hot exhaust gases; a fraction of these HC will oxidize in the exhaust port and manifold.

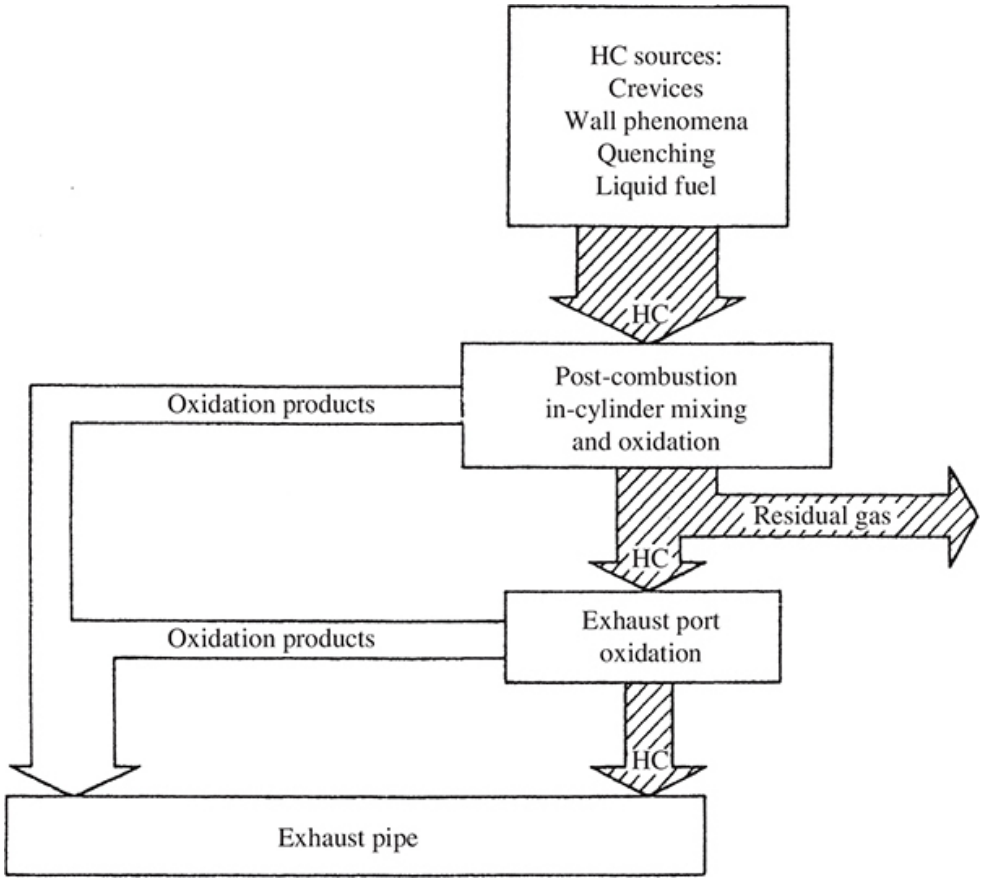


Figure 11.23 The various steps in the overall SI engine hydrocarbon mechanism: HC sources, post combustion mixing and oxidation, retention, and exhaust.[35] Width of shaded bars indicate HC concentration.

The HC concentration in the exhaust gases varies significantly during the exhaust process. Figure 11.24 illustrates our current understanding. The sources of HC are initially around the outside of the combustion chamber;

these hydrocarbons then diffuse from the walls into the bulk gas. Mixing of these HC with the bulk cylinder gases (and some oxidation) occurs during expansion and the exhaust blowdown process. The remaining unburned HC are exhausted in two peaks of approximately equal mass: the first of these coincides with the exhaust blowdown flow (which removes the majority of the *mass* from the cylinder). Then, during the final stages of the exhaust stroke, piston motion pushes most of the remaining fraction of the cylinder mass (with its high HC concentration) into the exhaust. This leaves a high concentration of HC in the residual gas in the cylinder, and also in the exhaust port and manifold. Approximately one-third of the hydrocarbons left unburned in an engine combustion event are retained in the cylinder and recycled, under typical driving conditions (light load). [36,37]

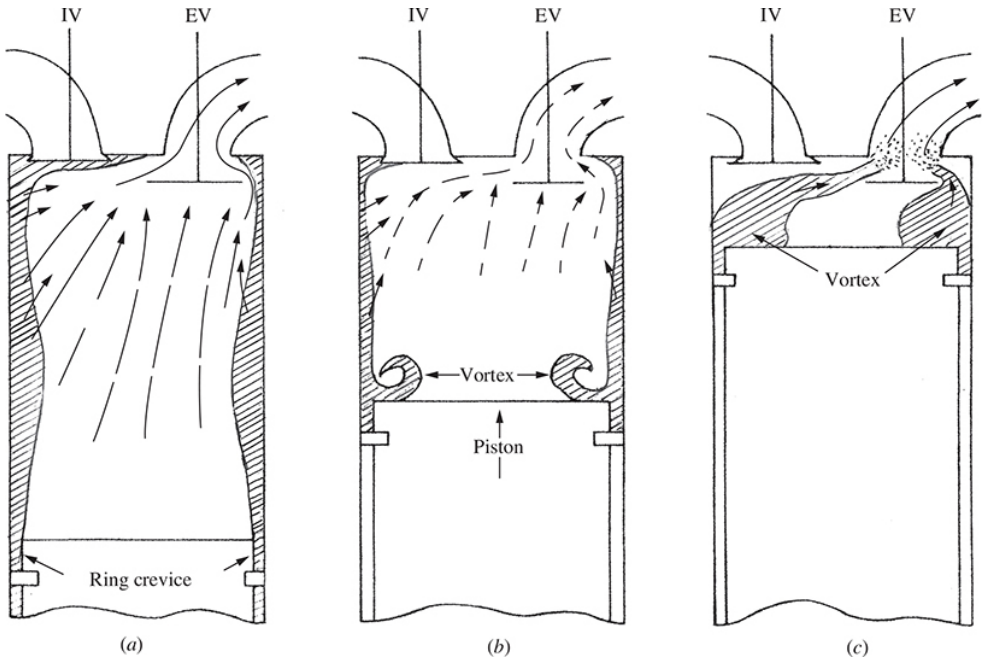


Figure 11.24 Schematic of in-cylinder flow processes by which hydrocarbons, post combustion, diffuse from the various sources into the bulk burned gases and are entrained into the blowdown exhaust flow (*a*), and then into the piston-driven outflow, (*b*) and (*c*).

These several schematics of the total spark-ignition engine HC emission process (Figs. 11.22, 11. 23 , and 11.24) explain its several sequential steps and parallel paths. We will return to the overall process after discussing the more important contributing mechanisms.

Crevice HC Mechanism

The crevices in the combustion chamber walls—small volumes with narrow entrances—into which the flame is unable to penetrate have been shown to be the largest source of unburned HC. The largest of these crevice regions is the volume between the piston, piston rings, and cylinder wall. Other crevice volumes in production engines are the threads around the spark plug, the space around the plug center electrode, crevices around the intake and exhaust valve heads, and the head gasket crevice. Table 8.1 shows the size and relative importance of these crevice regions in one cylinder of a typical production gasoline engine determined from measurements of cold-engine components. Total crevice volume is about 1.5 to 2.5% of the clearance volume and the piston and ring pack crevices are the dominant contributor. [32]

The sequence of crevice processes occurring during the engine cycle are the following. As the cylinder pressure rises during compression, unburned mixture is forced into the crevice regions. Since these volumes are thin, they have a large surface/volume ratio so the gas flowing into each crevice cools by heat transfer to close to the wall temperature. During combustion, while the pressure continues to rise, unburned mixture continues to flow into the crevice volumes. When the flame arrives at each crevice, it can either propagate into these crevices and partially burn the fuel and air within the crevice, or it can quench at the crevice entrance. Whether the flame quenches depends on crevice entrance geometry, the composition of the unburned mixture, and its thermodynamic state as described in Sec. 11.4.2 . After flame arrival at the crevice entrance, burned gases will flow into each crevice until the cylinder pressure starts to decrease. Once the cylinder pressure reaches its peak and starts to fall, gas flows back from each crevice into the cylinder.

The most important of these crevices, the volume between the piston, piston rings, and cylinder wall, is shown schematically in Fig. 8.32. The overall piston crevice region consists of a series of volumes, connected by flow restrictions such as the ring side clearance and ring gap, whose geometry changes as the ring moves up and down in the ring groove sealing either the top or (mostly) bottom ring surface. The gas flow, pressure distribution, and ring motion are therefore coupled, and their behavior during the compression and expansion strokes has already been discussed in Sec. 8.6 . During compression and most of combustion, mass flows *into* the volumes in this total crevice region. Once the cylinder pressure starts to decrease (at MBT spark timing, about 15° ATC), gas flows out of the top of

these crevice regions in Fig. 8.27 into the cylinder at low velocity adjacent to the cylinder wall. The important result is that a fraction of the total fuel mass in the cylinder (5 to 10%; see Example 11.1 below) is trapped in these regions at the time of peak cylinder pressure and escapes the primary combustion process. Most of this trapped gas flows back into the cylinder during the expansion process. Depending on spark plug location in relation to the position of the top ring gap, more than half of this gas is likely to be unburned fuel-air mixture. Its potential contribution to unburned HC emissions is obvious.

EXAMPLE 11.1

At the peak pressure point in the cycle, the mass of gas in the crevices is at its maximum. If the volume of the crevices (V_{crev}) is 2% of the cylinder clearance volume (V_c), the crevice gases are at the average wall temperature (e.g., liner 100°C, piston crown 200°C), the in-cylinder gas is 75% burned (burned gas temperature 2700 K, unburned gas at 800 K), and the cylinder volume at 15° ATC is 1.16 times the clearance volume, calculate the mass of gas in the crevices relative to total mass in the cylinder. If the crevice gas is two-thirds unburned mixture, what fraction of the fuel is in the crevices at this point in the cycle?

The appropriate scaling relationship is:

$$\frac{m_{crev}}{m_{cyl}} = \frac{V_{crev}\rho_{crev}}{V_{cyl}\rho_{cyl}} = \frac{V_{crev}}{V_{cyl}}\left(\frac{p}{RT_{crev}}\right)\left(\frac{RT_{cyl}}{p}\right)$$

where the ideal gas law, $p = \rho RT$, has been used. The average crevice gas temperature is approximately 150 °C (=423 K); the average cylinder gas temperature is (0.75 × 2700 + 0.25 × 800) which is 2225 K. Thus

$$\frac{m_{crev}}{m_{cyl}} = \frac{2}{100} \times \frac{2225}{423} = 0.105 \text{ or about } 10\%.$$

If the unburned gas fraction is two-thirds, then the fraction of fuel which escapes burning during combustion is 7%.

There is substantial evidence to support the above description of crevice HC phenomena and the piston/ring crevice region in particular. Visualization studies in a special engine have identified the spark plug crevice outflow, low-velocity gas expansion out of the volume above the first ring after the time of peak pressure, and the jet-like flow through the top ring gap later in the expansion process when the pressure difference across the top ring changes sign. [38] Gas sampling from the volume above the top ring has shown that the gas composition in this region corresponds to unburned fuel-air mixture until flame arrival at the crevice entrance closest to the sampling valve location. Next, burned gases enter the crevice as the cylinder pressure continues to rise. Then, during expansion as gas flows out of this region, the composition of the gas sampled reverts back toward that of the unburned mixture which entered the crevice region earlier. [39] Flame geometry calculations suggest that about two-thirds of the crevice gases are unburned fuel, air, residual gas mixture, though the percentage depends on the spark plug location and the symmetry of flame front propagation. [34]

Direct evidence that the piston and ring crevice regions are a major contributor to exhaust HC emissions comes from experiments where the volume of this crevice region has been substantially changed. HC emissions increase as the top-land crevice volume increases. The extent of the topland height has been decreased over time (from some 10 to 5 mm), to decrease this crevice volume. So has the radial clearance. HC emissions do scale with piston crevice size and reductions in other crevice volumes, but the relationship is more complex than a linear one due to partial oxidation of the crevice HC and there being other HC sources. The data in Fig. 11.25 confirm that the fuel stored in the piston crevices is significant through its impact on the engine's fuel conversion efficiency. [32] The least-squares linear fit to the data indicate that a 1000 mm^3 (1 cm^3) increase in piston top-land crevice results in a 1 to 1.3 percentage points decrease in net indicated fuel-conversion efficiency. This change in efficiency (due to a 1000-mm^3 change in piston top-land crevice volume) corresponds to 3% of the fuel entering the cylinder not burning early enough to produce torque. Allowing for in-cylinder HC oxidation (about two-thirds), HC retention in the residual gas (one-third), and exhaust port oxidation (also about one-third) this would result in 0.4 to 0.5% of the fuel as HC. Thus the piston top-land crevice, some three-quarters of the total cylinder crevice volume, is the origin for

about 40% of the engine-out HC. This percentage is in agreement with other estimates. [34]

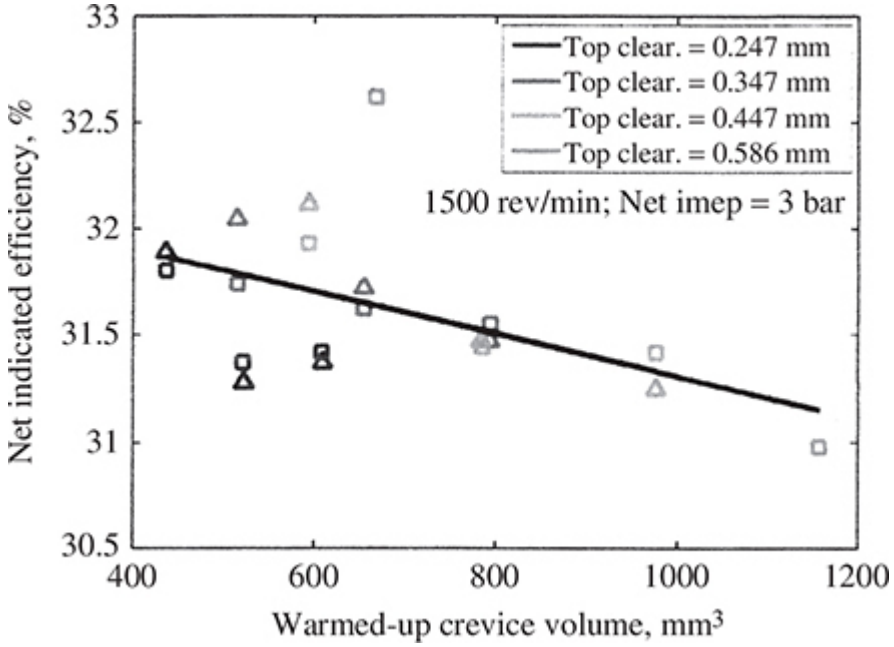


Figure 11.25 Effect of crevice volume on net indicated fuel conversion efficiency. Data from single-cylinder SI engine with several different piston ring-pack dimensions, each giving different piston crevice volumes: 1500 rev/min and net imep = 3.6 bar. [32]

Just how the flame quenches at crevice entrances is only partly understood. Figure 11.26 compares the warmed-up top-land clearances with the two-plate quench distance [Eqs. (11.27 and 11.28)] for a range of cold piston-radial-clearances, with the predicted two-plate quench distance. In this study, [32] the warmed-up radial clearance was 0.08 to 0.09 mm less than the cold clearance, and the comparison suggests that at the higher loads some flame penetration into the crevice may occur. An earlier study directly measured flame penetration: with the specific piston geometries and operating conditions used, flame penetration varied from essentially zero to a few mm (average just over 1 mm) so was not significant, most of the time. [40] Also, the geometry of the two-plate flame quenching situation is different from that of the piston crevice, as illustrated in Fig. 11.27. [32] The vertical cylinder-liner wall in the right-hand sketch (which is cooler than the piston

crown) would be expected to cool "half" of the flame close to the walls quicker, so quenching would occur at wider radial piston clearances than the basic two-plate quench studies would indicate. Thus we anticipate that under current engine circumstances, flame quenching close to the piston crevice entrances does occur.

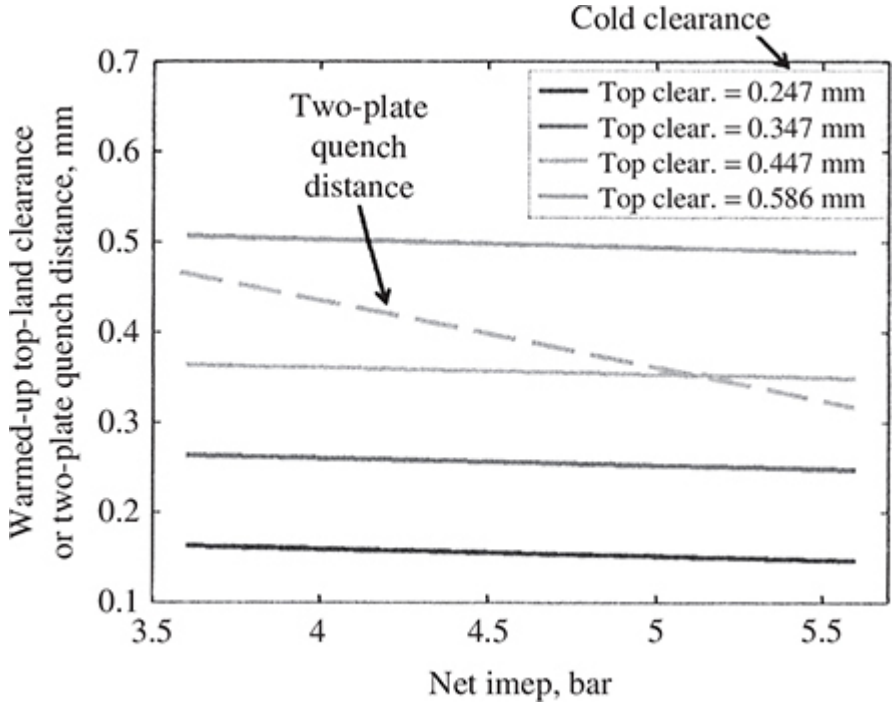


Figure 11.26 Comparison of warmed-up piston-to-liner top-land clearance for the four different pistons tested in the study shown in Fig. 11.25, with the two-plate quench distance, as a function of engine load. (Speed change effects are small.) [32]

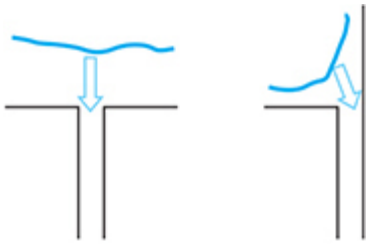


Figure 11.27 The geometries for the two-plate flame quenching process (left), and the piston-liner crevice flame quenching process (right).

The impact of the various crevice volumes identified in Fig. 11.22 can be inferred from their relative volumes. Table 11.7 compares the actual size of the different crevice volumes, and their percentage of the total, in a modern automobile spark-ignition engine. Through improvements in design, over time, these volumes have been decreased. The top-land height is constrained by piston rim strength considerations. The piston top-land dominates (~ 75%) though the spark plug crevices are about one-fourth of that. (Whether the piston second land volume, between the top and second rings, is a significant contributor is unclear. The gas pressure in this region is only a few atmospheres.) Earlier studies which converted the cylinder head, and the cylinder with piston close to TC, with steel plates to constant volume combustion vessels, then sealing and filling the various crevices in turn, produced HC concentrations after combusting the fuel air mixture inside each vessel which then allowed estimates of the relative contributor from each crevice. [41] These percentages were comparable in ranking and in magnitude to the volume percentages in Table 11.7.

TABLE 11.7 Volumes of the various crevices (mm^3 and percent of total) in single cylinder of modern spark-ignition engine [32]

	Volume (mm^3)	Percent
Spark plug crevice	187	18.0
Valve seat crevice	32	3.1
Head gasket crevice	37	3.6
Piston crevice	782	75.3
Total crevice volume	1038	100.0
Clearance volume (mm^3)	45,400	
Vol. piston crevice/Vol. clearance (%)		1.72
Vol. total crevice/Vol. clearance (%)		2.28

Blowby is the gas that flows from the combustion chamber, past the piston and rings, into the crankcase. It is forced through any leakage paths afforded by the piston-liner-ring assembly in response to high in-cylinder gas pressures during compression, combustion, and expansion. Blowby of gases from the cylinder to the crankcase removes gas from this crevice region and

thereby prevents some of the crevice gases from returning to the cylinder. Crankcase blowby gases were once vented directly to the atmosphere and constituted a significant source of HC emissions. The engine's crankcase is now vented to the intake system, the blowby gases are recycled, and this source of HC emissions is now fully controlled. Blowby at a given speed and load is controlled primarily by the greatest flow resistance in the flow path between the cylinder and the crankcase. This is the compression ring gap area. Figure 8.35 shows how blowby increases linearly with this smallest gap area. Figure 11.28 shows how exhaust HC levels decrease as blowby increases and more crevice HC flows to the crankcase. While blowby is usually, measured as a volume flow rate (at standard atmospheric pressure and temperature) out of the engine's crankcase, it is more meaningful to express it as a percentage (by mass) of the total flow rate into the cylinder. That flow in is air plus fuel plus any EGR. For a typical cylinder volume (500 cm^3) at mid-load and low-speed, 0.05 liter/s blowby corresponds to about 1% of this total flow. See light-load example below.

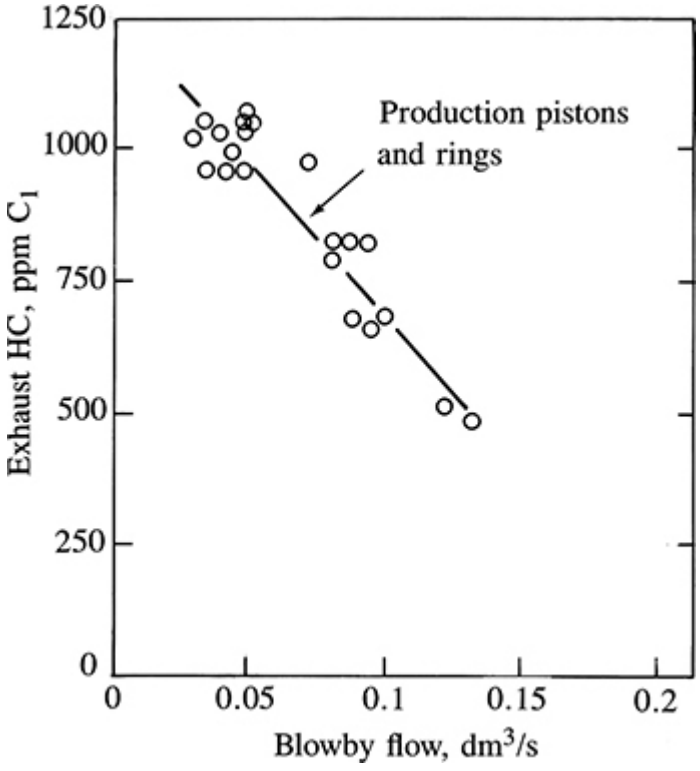


Figure 11.28 Effect of increasing crankcase blowby on exhaust hydrocarbon emissions. Production pistons and rings. SI engine at 1200 rev/min, intake manifold pressure 0.6 atm, $A/F = 14.2$. Blowby is per

cylinder: for a 500-cm ^{3} displaced-volume cylinder, 0.5 liters/s correspond to about 1.2% of the air, fuel, and EGR flow at this mid-load condition. [42]

EXAMPLE 11.2

A 500-cm ^{3} displaced-volume cylinder operating at a manifold air pressure of 0.5 bar and 1500 rev/min, a typical light-load operating condition, has a blowby of 0.05 dm ^{3} (liters)/s. Estimate this blowby as a percentage of the total mass flow rate into the cylinder.

The air flow into the engine is given by

$$\dot{m}_a = \eta_v \rho_{a,i} V_d N/2$$

where $\rho_{a,i}$ is the air density at intake air conditions: with this definition, $n_v \approx 0.8$. The density of air is obtained from the ideal gas law:

$$\rho_{a,i} = \frac{p}{RT} = \frac{0.5 \times 100}{287 \times 310} = 0.56 \text{ kg/m}^3$$

so

$$\dot{m}_a = 0.8 \times 0.56 \times 500 \times 10^{-6} \times 1500/60 \div 2 = 2.8 \times 10^3 \text{ kg/s}$$

The blowby flow of 0.05 liter/s, assuming that the blowby gas (a mixture of fuel and air, and burned gases) has a density close to that of air (1.18 kg/m ^{3}) at standard pressure and temperature, converts to a mass flow:

$$\dot{m}_{by} = 0.05 \times 10^{-3} \times 1.18 \times 10^3 = 0.059 \text{ g/s}$$

The cylinder intake flow of air, fuel (assumed stoichiometric), and EGR (assumed to be 10 percent of the exhaust flow) is given by

$$\dot{m} = \dot{m}_a + \dot{m}_f + \dot{m}_{EGR} = \dot{m}_a \left(1 + \frac{\dot{m}_f}{\dot{m}_a}\right) / (1 - x_{EGR})$$

Thus:

$$\dot{m}_i = 2.8(\text{g/s})(1+0.067)/(1-0.1) = 3.3\ \text{g/s}$$

and blowby flow as a fraction of total intake flow is

$$x_{by} = 0.059/3.3 = 0.018\ (=1.8\%)$$

Blowby is typically 1 to 2% of the in-cylinder mass, per cycle. This loss of gas from the cylinder has a modest effect on torque (the cylinder pressure is slightly lower due to blowby) but since the blowby is recycled, the impact on efficiency is negligible.

Crankcase blowby gases represent a performance loss. They are a smaller efficiency loss because crankcase gases are now recycled to the engine intake system.

The fate of the crevice HC when they flow back into the cylinder during expansion and exhaust is not well understood. Both low-velocity creeping flows (from the piston top-land crevice) and jet-like flows (from the top ring gap) have been observed (see Fig. 8.34). While the latter could mix rapidly with the high-temperature bulk burned gases, the latter will enter the cooler gases in the cylinder wall boundary layer and (probably) mix and burn much more slowly. Hydrocarbon transport and oxidation processes are discussed more fully below.

HC Absorption and Desorption in Engine Oil

Lubricating oil absorbs gasoline components from the in-cylinder unburned mixture. Absorption and desorption occur within the cylinder each engine cycle thus providing another mechanism for some of the fuel to escape burning during the primary combustion process.

This absorption/desorption mechanism is thought to work as follows. An oil layer a few microns thick on the cylinder liner is uncovered by the descending piston during the intake stroke, and remains exposed to the in-cylinder gases until covered by the piston during the compression stroke. During this time the oil layer is exposed to unburned fuel-vapor, air, residual gas mixture (and possibly, during the intake process, to liquid fuel). Figure 11.29 *a* shows the expected concentration profiles of gasoline vapor components that are soluble in the oil during intake and compression. As a consequence, such vapor components are transferred from the bulk gas into

the oil layer. It is usually assumed that Henry's Law, which relates the partial pressure of each vapor component p_f to the dissolved mole fraction of that component in the oil $\tilde{x}_f$, is valid:

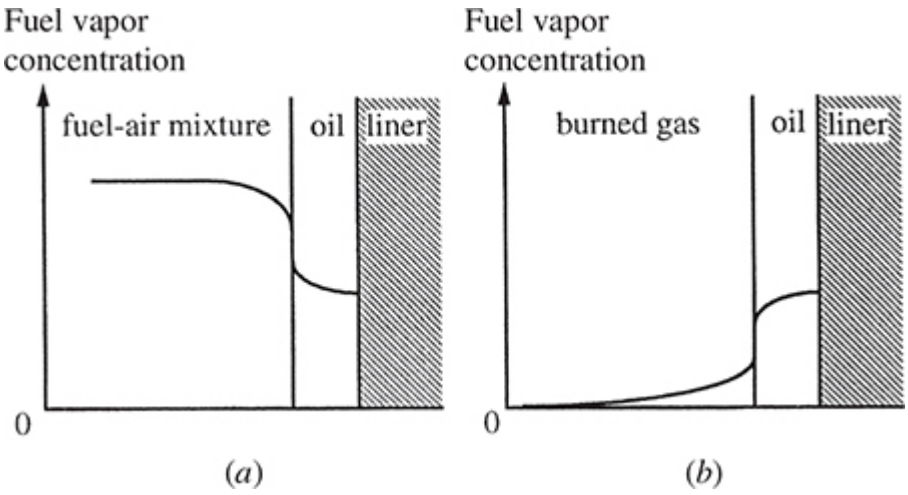


Figure 11.29 Schematic of HC absorption into, and desorption out of, oil layers on the liner. Gasoline component vapor concentration profile in the in-cylinder gases and in the lubricating oil on the cylinder liner: (*a*) during intake and compression and (*b*) post combustion and during exhaust.

$$\tilde{x}_f = \frac{p_f}{H} \tag{11.32}$$

where $\tilde{x}_f$, is the mole fraction of fuel vapor in the oil, p_f is the partial pressure of fuel vapor in the gas, and H is Henry's constant. Henry's constant, which relates these two properties, varies strongly with temperature so the amount of vapor absorbed into the oil is significantly higher at lower oil temperatures and (via each component's partial pressure) at higher gas pressures.

During the expansion and exhaust strokes, the oil layer on the liner, uncovered by the downward moving piston, has higher concentrations of gasoline compounds in the oil than correspond to equilibrium with the close-to-zero levels of gasoline in the burned gases which fill the cylinder: see Fig. 11.29 *b*. Thus hydrocarbons are desorbed out of the oil film and diffuse into the cylinder gases. If they do not then oxidize, they can exit the cylinder and contribute to the engine-out unburned hydrocarbon emissions. If the oil layer

is sufficiently thin, and thus diffusion is rapid, Eq. (11.32) can be used to estimate the mole fraction of the fuel dissolved in the oil. Since $p_f = n_{f,c}\tilde{R}T/V$ (where $n_{f,c}$ is the number of moles of fuel in the cylinder, T is the temperature, and V the cylinder volume) and $x_f = n_f/(n_{f,o} + n_o) = n_{f,o}/n_o$ for $n_o \gg n_{f,o}$ (where $n_{f,o}$ is the number of moles of fuel dissolved in the oil and n_o is the number of moles of oil), [43] then

$$\frac{n_{f,o}}{n_{f,c}} = \frac{n_o \tilde{R}T}{HV} \tag{11.33}$$

Diffusion is sufficiently rapid for Eq. (11.33) to be valid if the diffusion time constant τ_d is much less than characteristic engine times: that is,

$$\tau_d \approx \frac{\delta^2}{D} \ll N^{-1}$$

where δ is the oil layer thickness, D is the diffusion coefficient for fuel vapor in the oil, and N is engine speed. D for a hydrocarbon diffusing through a motor oil is of order 10^{-6} cm^2/s at 300 K and of order 10^{-5} cm^2/s at 400 K. Oil film thicknesses on the cylinder wall vary during the operating cycle between about 1 μm and 10 μm.[44,45] Thus diffusion times for engine conditions are 10^{-1} to 10^{-3} s; thus for the thinnest oil layers, equilibration would essentially be achieved.

Evidence for oil absorption and desorption at the high end of this range come from experiments where a specially designed spark-ignition engine was operated normally with an oil film on the liner to lubricate the piston rings and skirt, and then oil free. [46] For stoichiometric operation in the oil-free mode, at light load and low speed, engine out emissions were about 25% less than with normal lubrication. Other studies [47,48] indicate that in a warmed-up engine the contribution may be less than about 10%. In a cold engine, when the solubility of key fuel components in the oil is significantly higher, the contribution is likely to be higher. Under colder engine conditions, gasoline compounds are present in the oil in the sump (1 to 2%), and at higher levels than this in the oil residing in the ring pack region. [49] Under these conditions, it is likely that liquid fuel impinges on the liner oil layer,

and flows into the ring pack. This additional gasoline in the oil is likely to desorb HC into the burned exhausting gases as the engine warms up and be a source of additional HC emissions.

The different organic compounds in gasoline vary significantly in their solubility in engine oils, which makes accurate predictions of the amount of HC this mechanism produces difficult. However, there is now substantial experimental and theoretical evidence that oil layers within the cylinder, primarily on the cylinder liner, are a source of some 10 to 25 percent of the unburned HC emissions. [34]

Impact of Deposits

Over extended mileage, deposits build up on the intake valves, and on the combustion chamber walls of spark-ignition engines fueled with standard gasolines. Both these types of deposits affect hydrocarbon emissions. The deposits on the outside of the intake valve, which are porous, absorb some of the liquid fuel injected onto the back (intake side) of the valve with port fuel injection systems, thereby delaying its entry into the cylinder during engine transients. The in-cylinder mixture is therefore leaner than intended during accelerations (throttle openings) and richer than intended during decelerations (throttle closings). Changes in mixture composition affect engine-out HC emissions. Intake valve deposits primarily affect HC emissions during changes in engine load and, during starting and warm-up. They do not significantly affect steady-state HC emissions.

Combustion chamber deposits are known to increase steady-state as well as transient HC emissions. While deposit build-up on the cylinder and piston crown walls can take some 10,000 to 15,000 km to stabilize, their impact on HC emissions occurs much more rapidly. The engine-out HC rise most rapidly from clean-engine values over about 10 hours of operation, then more slowly over the next 20 hours, and stabilize over the next 10 to 20 hours even though deposit thicknesses continue to grow. However, the extent to which combustion chamber deposits in actual use build-up is not well quantified, and may well vary significantly with use patterns as well as the fuel used, oil consumption and oil type (the lubricant is the key component in forming deposits). SI engine deposits are porous in structure. The larger pores apparently act as “distributed crevices,” and store HC during the combustion process. The smaller pores with their large surface to volume ratio may absorb significant HC into any engine oil absorbed onto the deposits, which

reacts with gasoline combustion-generated radicals to form these deposits. Substantial levels of deposits in modern spark-ignition engines (e.g., levels which cause an increase in octane requirement of 5 to 10 octane numbers) can cause steady-state engine-out hydrocarbon emissions to increase by significant amounts. [51]

The above discussion is consistent with data on in-cylinder deposit build-up effects on HC emissions, which show that HC emissions can rise significantly over several thousand miles depending on how mileage was accumulated. Figure 11.30 shows the time-resolved HC concentrations measured with a fast-response FID HC analyzer through the exhaust process (130° ATC EVO to 15° ATC EVC: there are sampling time delays) and compares a clean engine (without combustion chamber deposits) with the same engine with stabilized deposit levels (with CCDs). The trace shows that total HCs are high during the first blowdown high mass flow phase, and later during the latter half of the exhaust stroke. This pattern of the blowdown flow entraining HC originating close to the cylinder head and resulting in significant HC exiting the cylinder in the first third of the exhaust process, and of the piston motion during the latter part displacing high HC gases from the bottom and middle of the cylinder during the displacement exhaust phase and out of the exhaust valve, has been known for many years. [36] With CCD, in Fig. 11.30, additional HCs exit early (likely from the fuel absorbed into the cylinder-head deposits) and exit later (likely from piston crown deposits). The time-averaged HC concentration difference due to deposits in this example of an engine with significant deposits is an increase of 20 to 25%. What is not yet adequately documented is the effect on engine-out HC emissions of combustion chamber deposit levels typically found in modern spark-ignition engines, with today's gasolines and lubricants.

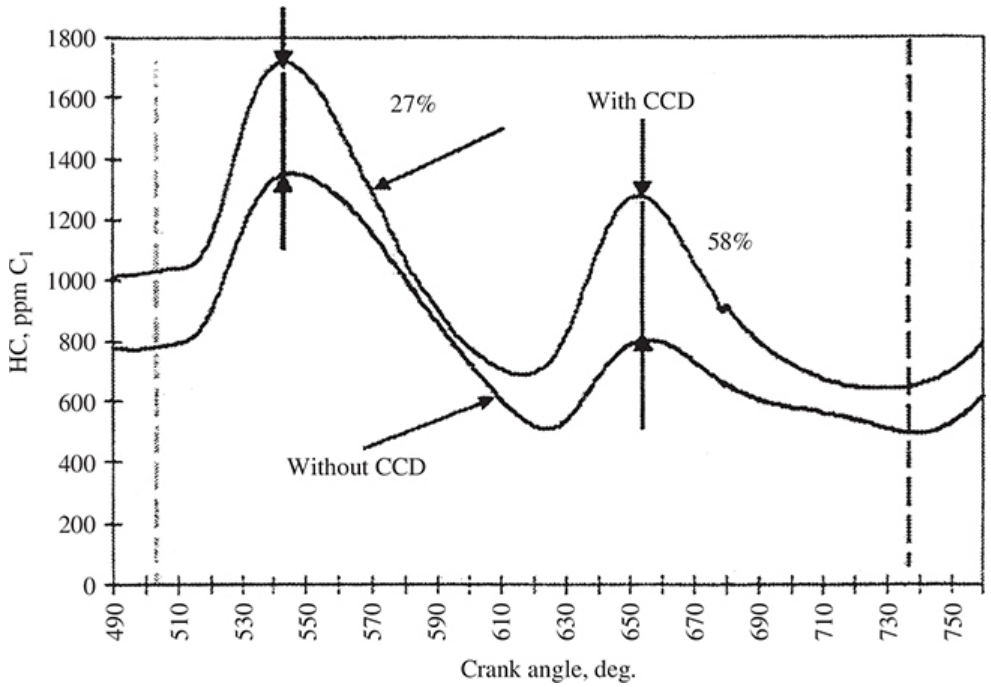


Figure 11.30 Time-resolved HC concentrations through the exhaust process of a warmed-up spark-ignition engine, with and without combustion chamber deposits (CCDs). Percentages are the HC increase at the two peaks above the clean engine (without CCDs) values. 2-liter four-cylinder naturally-aspirated gasoline-fueled engine operated at wide-open throttle, 2500 rev/min, and $\lambda = 0.9$. EVO and EVC at 503° and 730° ATC intake stroke, respectively. [50]

Flame Quenching

It has long been known that the spark-ignition engine flame extinguishes a short distance from the cooled combustion chamber walls leaving a thin layer of unburned (or partially reacted) fuel-air mixture adjacent to the wall. This is called a quench layer. While originally thought to be a major source of unburned HC emissions, it is now known that the mixture in these quench layers diffuses into the hot combustion products outside the layer and burns up during expansion, under most engine operating conditions. [37] It is not yet clear whether under extremely lean or high-EGR engine operation the thicker quench layers that then occur do fully burn up. However, even when the mixture in the quench layers does fully burn up before the engine cycle is completed, flame quenching at the combustion chamber walls still prevents a small fraction of the fuel (about 0.5%) from burning during the normal engine

combustion process. Quench layers thus constitute a small loss in engine performance and efficiency.

Flame extinguishment prematurely, before the above-described flame quenching at the chamber walls occurs, is potentially a more significant contributor to engine-out HC emissions. Such extinguishment can occur when an engine is operated close to its dilute operating limit (e.g., with high EGR), or its lean operating limit. Then, in slower burning engine cycles, the flame may approach the chamber walls sufficiently slowly, and late in the expansion stroke, that the pressure and temperature in the unburned mixture ahead of the flame is decreasing with time rather than increasing. The flame can extinguish under these circumstances resulting in a partial burning cycle, which could causes high HC emissions *for that cycle.* The effect on average HC emissions would depend on the relative frequency of occurrence of such partial burning cycles. It is also possible that misfiring cycles occur occasionally close to the EGR or lean limit when the flame initiated by the spark discharge extinguishes early in its development process. Then, close to a full cylinder charge of fuel and air will exit the cylinder unburned.

With a well-designed fast-burn combustion system, under normal steady-state engine operating conditions, flame-quenching phenomena are not expected to contribute significantly to HC emissions. However, during transients, with high levels of EGR, and with retarded timing, partial burning and even quench layers may contribute enough to total HC emissions to warrant attention. Poor combustion evident as partial burning and even misfiring in a small fraction of the engine's operating cycles are the explanations often given for part of the increase in engine-out HC emissions that occurs as EGR is increased at a fixed load and speed, as shown in Fig. 11.31. [34] The top figure shows how combustion slows down and deteriorates as EGR increases from moderate to high levels (COV_{imep} is the standard deviation, cycle-to-cycle, in imep divided by the mean imep. See Sec. 9.4.3 . A COV_{imep} below 2 to 5%, depending on operating conditions, is regarded as satisfactory for "stable" combustion. Part of the difference between the slower and faster burning combustion systems shown, at higher EGR levels, is due to the deteriorating combustion process. However, in a normally operating engine, bulk flame quench should not be occurring.

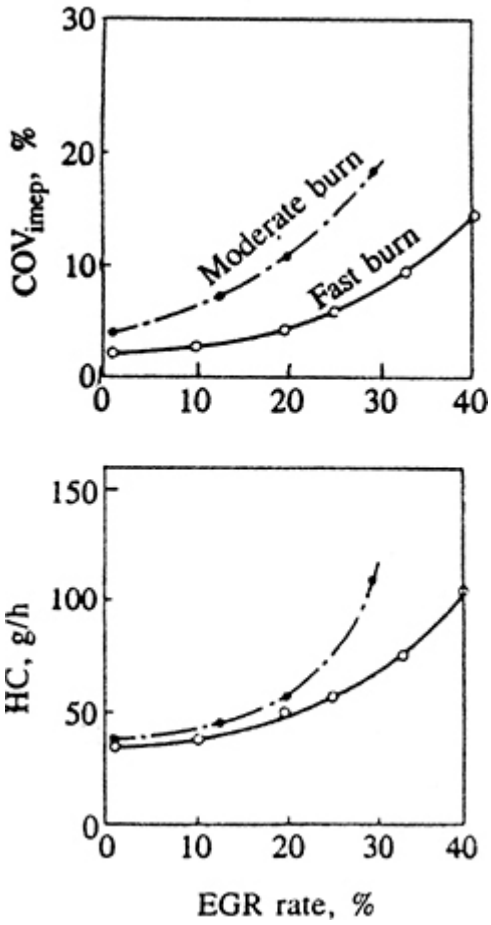


Figure 11.31 Effect of increasing percent exhaust gas recycle (EGR) on the coefficient of variation in indicated mean effective pressure and engine-out HC emissions at 1400 rev/min and 324 kPa imep, stoichiometric operation, MBT spark timing.[34]

Liquid Fuel on the Cylinder Walls

Both port-fuel-injected and direct-injected gasoline engines are in common use. In port-injected engines, the injector usually directs the liquid gasoline toward the back of the intake valve and surrounding intake port surface area. Most of the injected gasoline impinges on these surfaces, though a portion of the fuel will evaporate from the fuel droplets formed in the atomization process. The behavior of the liquid fuel after injection is strongly influenced by the timing of the injection pulse relative to the intake valve opening and closing times. With injection onto a closed valve, the rapid backflow of burned gases from the cylinder immediately following intake valve opening assists the fuel vaporization process off the valve and port surfaces

substantially; [52] under fully warmed-up part-load engine operating conditions, almost all the gasoline fuel enters the cylinder as vapor. With injection toward an open valve, however, more of the fuel will enter the cylinder as liquid, as droplets or ligaments entrained in the intake airflow. A major part of this liquid evaporates and mixes with the in-cylinder air, fuel vapor, and residual gas, to form the normal combustible mixture during the intake and compression process. The remaining liquid, especially the less volatile fuel components may form a liquid film, or may be stored in the deposits, oil layers or crevices in the combustion chamber, and vaporize into the bulk gases during the expansion and exhaust processes. The incomplete oxidation of this fuel would contribute to the exhaust hydrocarbon emissions. The HC emissions index, Eq. (2.44) , for liquid fuel entering the cylinder with port-injection is some five times that for fuel that enters as vapor, though this varies substantially with the valve seat location where the liquid flows in.[53]

The amount of not-yet evaporated fuel in the cylinder is especially significant during the engine starting process. Because the inlet valve and port, and the intake air, are cold, only a fraction of the injected fuel will evaporate: to achieve rapid and smooth starting, a much larger amount of fuel than would be required with complete vaporization must be injected, and a substantial amount of that fuel does not appear in the combustible charge mixture. This phenomenon is illustrated in Fig. 11.32. The data for this figure was obtained from single-cylinder engine running at a fixed speed (900 rev/min) and intake manifold pressure (0.5 bar). The in-cylinder hydrocarbon vapor concentrations before combustion were measured by a fast-response flame-ionization HC detector. From these HC concentrations values, the fuel mass, which had evaporated to form the combustible mixture, was estimated for each cycle. The remaining fuel mass was labeled as "unaccounted for." The cumulative amount of "unaccounted for" fuel is plotted as a function of the number of cycles from first injection in Fig. 11.32. The first firing cycles are also marked in the figure. Initially, only about 50% of the fuel enters the cylinder and vaporizes. The cumulative mass of unaccounted-for fuel for enrichments typical of engine startup could be some 10 or so injections. Part of the fuel fraction that enters the cylinder without vaporizing and mixing with the combustible charge mixture may find its way to the oil sump. The remaining part, however, may mix with the charge later

in the cycle and, if it is not oxidized, would contribute to the engine-out hydrocarbon emissions.

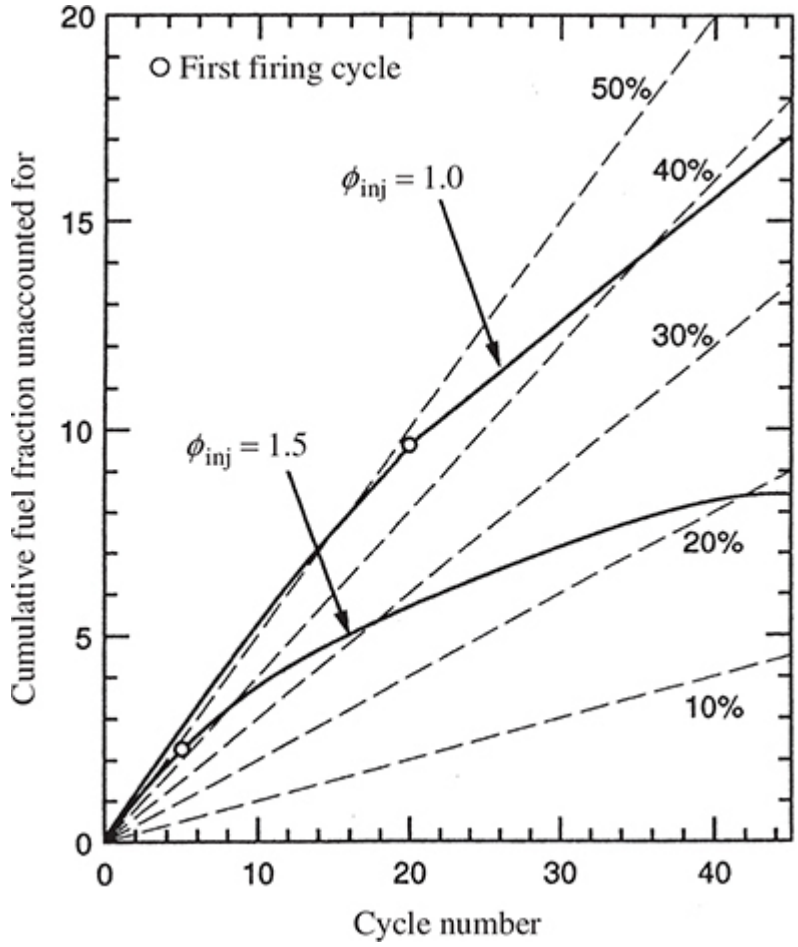


Figure 11.32. "Unaccounted for" fuel accumulation (normalized by the amount of one injection) as a function of cycle number during engine starting and warm-up: 900 rev/min, intake manifold pressure 0.5 bar, spark timing 20° BTC. The circle on each trace marks the first cycle with significant positive imep. Also shown are lines of constant fuel-air equivalence ratio, based on injected fuel, of 1.0 and 1.5. [34,54]

Comparisons have been made between the HC emissions levels with prevaporized gasoline fueling and with port injection. For port-fuel-injected gasoline (indolene), the steady-state HC emission levels are only about 10% higher than the prevaporized fuel case. During the warm-up process, however, the HC emissions are substantially higher (by about a factor of 2) than those of the prevaporized fuel case. The time scale to steady state is of

order 200 seconds, which is much slower than that for the prevaporized fuel. The higher HC emissions (compared to the prevaporized fuel case) during warm-up are attributed to the presence of the liquid fuel in the crevice/oil layer/deposit regions in the combustion chamber, and are presumably due to two effects: (1) The presence of liquid fuel would mean that a much larger amount of HC is stored in these regions and (2) the transport processes of the liquid fuel into and out-of these "storage" regions are different from those of the fuel vapor. Both effects would contribute to the higher level of HC emissions. That the steady state HC emissions are only about 10% higher than in the prevaporized case is probably due to the fact that the intake manifold and back of the inlet valve have reached high enough temperatures to achieve nearly complete vaporization: thus little gasoline enters the cylinder as liquid.

Gasoline Direct-Injection Engines

Direct-injection (DI) fuel systems are increasingly replacing (or supplementing) port-fuel-injection systems. With DI, fuel is injected directly into the combustion chamber through an injector located in each cylinder head. The behavior of the liquid fuel injected into the cylinder has several HC emissions impacts. First, the liquid fuel films formed on the intake port and valve walls with port injection are eliminated. However, with DI there will be much more liquid fuel within the cylinder, especially with a cold engine and during warm up, which radically changes the fuel metering calibration during starting and warm up, and transients. The fuel entering each cylinder in a DI engine is exactly what is injected. In port-injected systems, due to the buildup of liquid fuel films within the intake port, there are fuel transport delays and, during engine start up, significantly more fuel must be injected than enters the cylinder during the first several cycles. Thus DI fuel injection has the potential for reducing the cold-engine fuel enrichment required, and the more complex fuel metering calibration necessitated by port fuel injection. Thus, with DI, engine-out hydrocarbon emissions during these phases of engine operation can be reduced.

For homogeneous-charge DI spark-ignition engines under quasi-steady operating conditions with gasoline, the HC emissions mechanisms are essentially the same as with port-fuel-injection engines. The behavior of the DI fuel spray does need to be carefully managed to avoid any substantial liquid wetting of the cylinder liner and piston crown surfaces. Fuel wetting

the walls and forming liquid films does increase the HC emissions from an SI engine. The reason, as explained in the previous subsection, is that vaporizing that liquid fuel on the walls (largely due to heat transfer from the hot burned gases after the flame passes over the film), and then mixing that fuel vapor with sufficient oxygen in those burned gases to burn up that fuel, takes time and thus occurs well into the expansion stroke. [55]

DI gasoline engines that are operated stratified at light- and mid-load generally have higher HC emissions than homogeneous stoichiometric-operating DI engines. Flame quenching near the outer boundary of the close-to-stoichiometric fuel-containing cloud (see Sec. 9.3.5) occurs due to the very lean mixtures (due to overmixing of fuel with air) in these outer cloud regions. Then, since the temperatures of lean-mixture burned gases are lower than of stoichiometric burned gases, the potential for post-flame oxidation of these HC is significantly reduced. Also, with lean engine operation overall in this stratified DI mode, exhaust gas temperatures are lower (as a consequence of the lower expansion-stroke gas temperatures of lean gases, and of higher engine efficiency) so afterburning of HC in the exhaust port is reduced as also is the oxidation catalyst efficiency (see Sec. 11.6.3) due to lower catalyst temperatures. Higher HC emissions levels from lean stratified DI gasoline spark-ignition engines is one of their major development challenges, as is the parallel need for catalyst systems for NO_x removal from lean exhaust gases (such as the lean-NO_x trap, Sec. 11.6.3).[56]

Exhaust Valve Leakage

Experiments with fast-response flame-ionization HC detectors in the exhaust port of engines have shown that, periodically, a rise in HC concentrations can occur in the port while the valve is closed. It is believed that exhaust valve leakage can occur intermittently, even with normally functioning exhaust valve systems. A plausible cause is that combustion chamber deposits may break-up and the larger particles that result become embedded for short periods in the exhaust valve seat. If leakage occurs, then prior to combustion, unburned mixture can flow into the exhaust. Under most circumstances this source of HC emissions is expected to be small though not necessarily negligible. In poorly maintained older engines, it may be a more significant source: up to 7% of total engine-out HC emissions.

Oxidation and Retention in the Cylinder

As summarized in Table 11.6, after the end of the normal combustion process, the HC which through the above-described mechanisms did not burn, can oxidize within the cylinder, or may be retained within the cylinder with the residual burned gas and therefore be recycled, or may be oxidized within the exhaust port and manifold. Each of these three processes significantly reduces the amount of unburned HC emitted by the engine. This second phase of the total HC emissions mechanism, despite its importance, has received much less attention than the "sources" part of the mechanism summarized in Table 11.5 and described above. [h]

Figure 11.33 shows the variation in the average in-cylinder gas temperature for a typical part-load operating condition during the expansion and exhaust processes. If the unburned HC from the sources described above mix early enough with the burned gases, and provided enough oxygen is present, they will burn. Note that there are two different types of sources. Crevices and flame quenching produce unburned *fuel-air mixture*; here the oxygen required to burn the fuel is already mixed with the fuel vapor. In contrast, oil layers, deposits, and liquid fuel mechanisms produce unburned hydrocarbon vapor, *separated* from the oxygen it will need to burn.

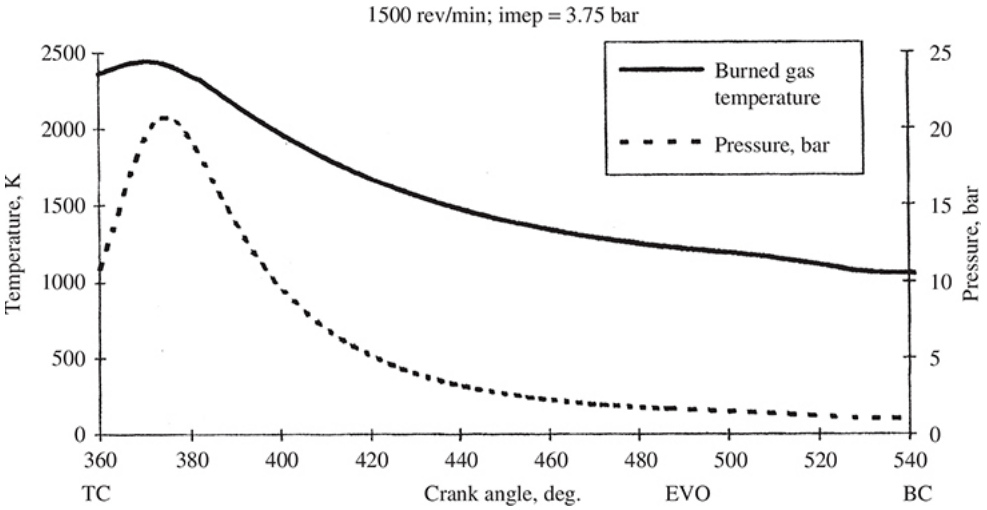


Figure 11.33 Cylinder pressure and average in-cylinder (burned) gas temperature during the expansion and exhaust strokes of the SI engine cycle, 1500 rev/min, 3.8 bar imep, stoichiometric operation. (Note that at peak pressure about 80% of the mixture has burned.) [34]

The studies of in-cylinder HC oxidation suggest that, locally, the HC in the quench layers on largely deposit-free combustion-chamber walls diffuse into the burned gases during the expansion stroke and oxidize almost completely. Engine cycle simulations suggest that the in-cylinder gas expansion and associated temperature decrease that occurs during the blowdown process that immediately follows exhaust valve opening slows down the in-cylinder oxidation process sufficiently to prevent much additional burn up occurring. At burned gas temperatures below about 1000 K, significant oxidation of HC in the time available in the cylinder is not likely to occur. An assessment of total in-cylinder HC oxidation by Daniel, [57] who fitted a physical mixing model and an overall chemical oxidation model to a matrix of engine-out HC emissions data at different operating conditions, estimated the amount of this post-flame oxidation. This model fitting process indicated that about one-third of the HC which escape the combustion process do fully oxidize within the cylinder. It is likely that the amount oxidized for each of the different HC sources depends on the composition of the source "efflux," the time of occurrence of that efflux during expansion and exhaust, and the location of the source within the cylinder. Various studies of these questions suggest that the HC from sources that produce fuel-air mixture after normal combustion (crevices, flame quenching) will oxidize significantly more than the HC from sources that produce fuel vapor (oil layers, deposits, liquid fuel films.)

The retention of the still unburned HC within the cylinder is also poorly understood. While the residual gas fraction can be estimated accurately, the HC are not uniformly distributed within the cylinder: since the sources are on the walls of the combustion chamber, the fraction of the in-cylinder HC retained is likely to be larger than the residual fraction. Previous analysis of the shape of the HC concentration and mass flow profiles through the exhaust process [36, 58] indicates that about half the exhausted HC exit the cylinder during the blowdown process and half toward the end of the exhaust-stroke displacement process. The blowdown presumably entrains much of the unburned HC adjacent to the wall at the top of the cylinder. The exhaust stroke is known to roll up the gas layer adjacent to the cylinder liner (which presumably contains much of the remaining unburned HC from the piston crevices and oil layers) into a vortex (see Sec. 8.9). Figure 11.24 illustrates our understanding of how the high HC-concentration gases from the various HC sources located around the outside of the combustion chamber move into

the bulk gases, and then exit the cylinder during the blowdown and piston displacement phases of the exhaust process.

Oxidation in the Exhaust

With a warmed-up engine, the gas temperature in the exhaust port, and even (under some conditions) in the exhaust manifold is hot enough for gas-phase HC oxidation to occur. Prior analyses and measurements confirm that for stoichiometric operation at part-load, with a warmed-up engine, about one-third of the engine-out HC will be burned up, as shown in Fig. 11.23. Under fuel-rich operating conditions, oxidation can be enhanced by air addition into the port region to ensure that adequate oxygen for burnup is available.

To oxidize the hydrocarbons in the gas phase, a residence time of order 50 ms at temperatures in excess of 600°C is required. To oxidize carbon monoxide, temperatures in excess of 700°C are required. Average exhaust gas temperatures at the cylinder exit (at the exhaust valve seat) are about 800°C; average gas temperatures at the exhaust port exit are about 600°C. [i] Figure 6.21 shows an example of measured cylinder pressure and the gas temperature at the exhaust port exit, and the estimated mass flow rate into the port, during the exhaust process at a part-throttle operating condition. Exhaust port residence time and gas temperatures vary significantly through the process. Precise values of these variables obviously depend on engine operating conditions. Normally, it is only in the exhaust port (and upstream end of the manifold) that any significant gas-phase HC oxidation can occur. Note that spark retard from MBT timing is an especially important variable. For cold-start engine-out HC emissions control, retarding the spark timing to after TC for 10 or more seconds is widely used to create a hotter exhaust flow which increases HC, CO, and H_2 burn up, and more quickly heats up and lights off the catalyst.

Secondary Air

Injection of secondary air into the exhaust ports of automobile spark-ignition engines to reduce the hydrocarbon emissions (and also CO emissions) has a long history. It is an effective strategy, but adds an extra system (an air pump, drive, and controls), and thus cost. It does provide significant reductions in cold start HC emissions entering the catalyst, and also speeds up the catalyst light-off process. Both of these benefits help achieve extremely low vehicle-out HC emissions levels. Also, secondary air can be used to compensate for

engine operation with a richer-than-stoichiometric mixture during starting and warm-up, which achieves smoother and more stable combustion but significantly increases engine-out HC emissions. An important part of this cold-start strategy is to retard the combustion phasing relative to piston motion, by sparking the fuel-air mixture in the cylinder (operating in the stoichiometric to some 20% rich range) after top center. Such retarded combustion increase the exhaust gas temperature significantly thus enabling much greater HC and CO burn up, with the two effects combining to light off the catalyst much more rapidly.

A study of the effects of secondary air injection on exhaust gas emissions, exhaust system heat transfer, and catalyst light off during cold start, using a variety of techniques to quantify emissions levels and temperatures at several locations from cylinder exit to the catalytic converter outlet, has identified the fundamentals of what occurs. The engine was operated at fast idle (i.e., cold engine) conditions, at fixed λ, with retarded spark timing (4 to 10° ATC), at fixed speed (1450 rev/min) and airflow (11 g/s). In-cylinder lambda was varied from 0.8 to 1.0. Secondary airflow rate ranged from 0 to 100% of the intake flow in the four-cylinder 2.2 liter naturally-aspirated gasoline engine. Operating 20% rich with a secondary air flow of 100% of the intake flow yielded the highest catalyst feed-gas temperature prior to catalyst light off (at 4.2 seconds). Substantial spark retard is required to initiate thermal oxidation of HC and CO in the exhaust gases. Holding this retarded spark for the duration of the catalyst light off period (~10 sec for an ambient-temperature cold start) is needed to keep the engine-out gases hot and enhance thermal oxidation in the exhaust port and manifold. To maximize this thermal oxidation, the secondary air should be injected as close to the exhaust valve as possible. The chemical energy released in the exhaust port outweighs the cooling effect of the secondary air. With $\lambda = 0.8$, secondary air could release up to 90$ of the cylinder-out exhaust gas chemical energy. Overall, the secondary air injection system reduced HC emissions by some 50 to 90% and CO emissions by 35 to 90% over the first 25 sec of operation at the conditions studied. [59]

An overall finding of this study was that post-catalyst HC emissions levels decreased steadily up to an exhaust relative air/fuel ratio of 1.3, with increasing secondary air, as shown in Fig. 11.34. The exhaust λ is defined by

$$\lambda_{exh} = \left(\frac{\dot{m}_{a,i} + \dot{m}_{a,s}}{\dot{m}_f} \right) \Big/ (A/F)_s \qquad (11.34)$$

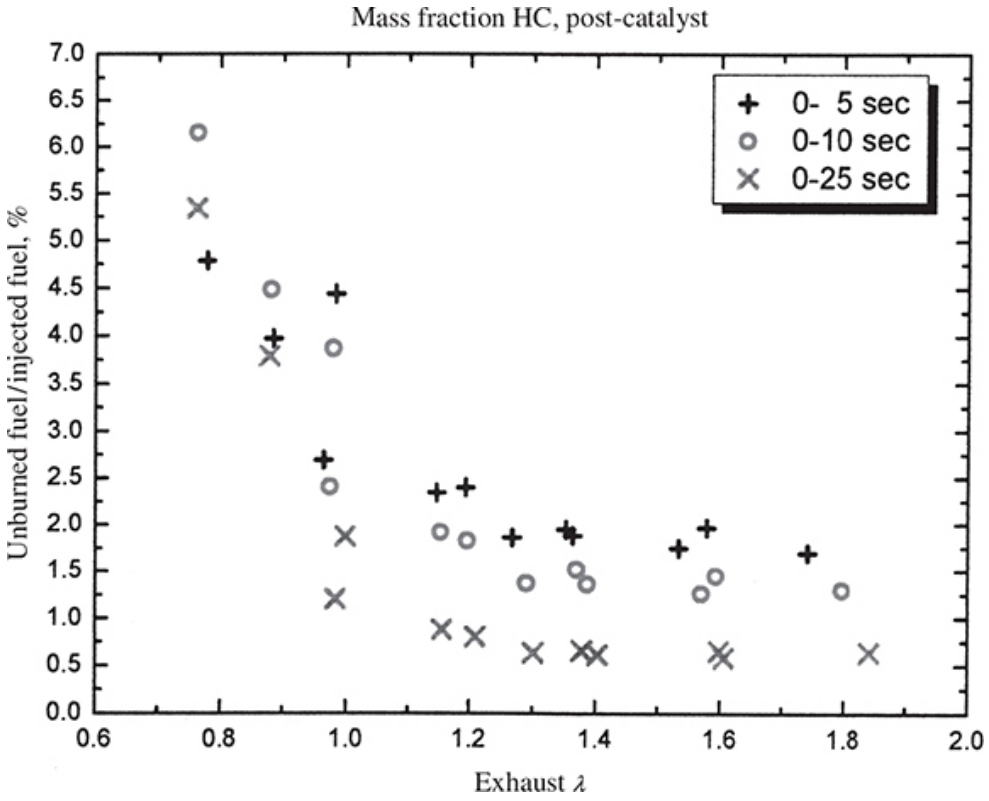


Figure 11.34 Cumulative mass HC emissions normalized by the amount of injected fuel at post-catalyst location as a function of exhaust lambda (relative air/fuel ratio, including secondary air), Eq. (11.34) , for time periods 0 to 5 sec, 0 to 10 sec, and 0 to 25 sec.[59]

where subscripts *i* and *s* denote intake and secondary air flows. Beyond this secondary air level there was no additional benefit from adding more air: the post-catalyst HC become essentially independent of the engine's relative air-fuel ratio.

Summary of the Total Process

It will be apparent from the above that the HC emissions formation process in spark-ignition engines is complex, and that there are several paths by which small but important amounts of the fuel escape combustion. It is appropriate here to summarize the overall structure of the spark-ignition

engine hydrocarbon emissions problem and identify the key factors and engine variables that influence the different parts of that problem. Table 11.8 provides such a summary. The total process is divided into four sequential steps: (1) the creation of unburned hydrocarbon sources; (2) the oxidation of a fraction of these HCs within the cylinder, following mixing with the hot bulk gases; (3) the flow of a fraction of the unoxidized HCs from the cylinder into the exhaust; and (4) the oxidation in the exhaust system of a fraction of the HCs that exit the cylinder. The detailed processes and the design and operating variables that influence each of these four steps in a significant way are listed.

TABLE 11.8 Critical steps and important engine variables in HC emissions mechanism

1. Sources of unburned HC
   (*a*) Crevices
      (1) Crevice volume
      (2) Crevice location (relative to spark plug)
      (3) Load
      (4) Crevice wall temperature
      (5) Mixture composition*
   (*b*) Oil layers
      (1) Oil consumption
      (2) Wall temperature
      (3) Speed
   (*c*) Incomplete combustion
      (1) Burn rate and variability
      (2) Mixture composition*
      (3) Load
      (4) Spark timing†
   (*d*) Combustion chamber walls
      (1) Deposits
      (2) Wall roughness
2. In-cylinder mixing and oxidation
   (*a*) Mixing rate with bulk gas
      (1) Speed
      (2) Swirl ratio
      (3) Combustion chamber shape
   (*b*) Bulk gas temperature during expansion and exhaust
      (1) Speed
      (2) Spark timing†
      (3) Mixture composition*
      (4) Compression ratio
      (5) Heat losses to walls
   (*c*) Bulk gas oxygen concentration
      (1) Relative (A/F) ratio
   (*d*) Wall temperature
      (1) Important if HC source near wall
      (2) For crevices: importance depends on geometry
3. Fraction HC flowing out of cylinder
   (*a*) In-cylinder flow during exhaust stroke
      (1) Valve overlap
      (2) Exhaust valve size and location
      (3) Combustion chamber shape
      (4) Compression ratio
      (5) Speed
   (*b*) Residual fraction
      (1) Load
      (2) Exhaust pressure
      (3) Valve overlap
      (4) Compression ratio
      (5) Speed
4. Oxidation in exhaust system
   (*a*) Exhaust gas temperature
      (1) Speed
      (2) Spark timing†
      (3) Mixture composition*
      (4) Compression ratio
      (5) Secondary air flow
      (6) Heat losses in cylinder and exhaust
   (*b*) Oxygen concentration
      (1) Relative (A/F) ratio
      (2) Secondary air flow and addition point
   (*c*) Residence time
      (1) Speed
      (2) Load
      (3) Exhaust component volume
   (*d*) Exhaust reactor‡
      (1) Oxidation catalyst
      (2) Three-way catalyst

* Relative air/fuel ratio and burned gas fraction (residual plus recycled exhaust gas).
† Relative to MBT spark timing.
‡ See Sec. 11.6.

The specific processes are listed in Table 11.8 in the most likely order of importance. Through each of these source mechanisms, fuel or fuel-air mixture escapes the primary combustion process. Those hydrocarbons must then survive the expansion and exhaust processes and pass through the exhaust system without oxidation if they are to end up in the atmosphere as HC emissions. The rate of mixing of these HCs with the hot bulk cylinder gases, the temperature and composition of the gases with which these HCs mix, and the subsequent temperature-time and composition-time histories of the mixture, will govern the amount of in-cylinder oxidation that occurs. The

distribution of these HCs around the combustion chamber is nonuniform (and changes with time); they are concentrated close to the walls of the chamber. The fraction of these HCs that will exit the chamber during the exhaust process will depend on the details of the in-cylinder flow patterns that take them through the exhaust valve. Overall, the magnitude of the residual fraction will be one major factor, but the residual gas is known to be much richer in HCs than the average exhaust gas. In particular, the flow patterns in the cylinder toward the end of the exhaust stroke, as gas is scraped off the cylinder wall as the piston moves toward the exhaust valve, will be important. Finally, a fraction of the unburned HCs which leave the cylinder through the exhaust valve will burn up within the exhaust system. Gas-phase oxidation in the exhaust ports (and possibly in the hotter parts of the exhaust manifold) is significant. The amount depends on the gas temperature, composition, and residence time. Then, the catalysts in the exhaust system provide a very substantial additional reduction in HC emission levels. Exhaust catalysts and their operating characteristics are described in Sec. 11.6.3 .

We now quantify the fuel fractions involved in the full HC emissions mechanism, based on our earlier discussion of each step in the process. The overall task is to quantify what fraction of the fuel remains unburned after each stage in the schematic shown in Fig. 11.23. This has been done for a typical part-load operating condition defined by the details in Table 11.9. [34]

TABLE 11.9 Part-load operating condition for fuel and HC flow chart

Fuel = gasoline
Stoichiometric *A/F*
Speed = 1500 rev/min
Inlet pressure = 0.46 atm
Imep = 3.8 bar, bmep = 2.7 bar
Spark timing = MBT (20° BTC)
Warmed-up coolant and oil

The complete flow chart for the gasoline which enters each cylinder each cycle is shown in Fig. 11.35. The fuel within each cylinder prior to any chemical reaction (100%) is at the top: through normal flame-propagation

combustion some 90% of it burns producing hot combustion products CO_2, CO, H_2O, H_2, N_2, … (the left-hand side of the diagram) which encompasses almost all the exhaust gases. The hydrocarbon emissions component of the exhaust prior to catalyst entry (lower right) is about 1.8% of the mass of the fuel, a well-defined number with a spread of about ±0.5 for typical engine operation. Note that normally about half these HC are compounds that are in the fuel, and half are partial reaction products of fuel molecules. The individual HC sources, and the amounts shown are *emissions indexes*, Eq. (2.44) , expressed as percent of the initial fuel in the cylinder. They are categorized as *fuel only* sources and *fuel-air mixture* sources (e.g., oil layers and crevices, respectively). This was done because evidence suggests that when the HCs from an individual source are fuel mixed with air, it is easier to oxidize them in the cylinder (post normal combustion) because the fuel vapor already has the oxygen needed to burn it up mixed with it. If the source outflow is HC alone (e.g., gasoline compounds desorbing from oil layers on the liner) then the HCs involved must mix with oxygen *and* with hot burned gases before they can oxidize. The in-cylinder oxidation ratios (one-third oxidized, or two-thirds oxidized for fuel only and fuel-air mixture, respectively) reflect that distinction.

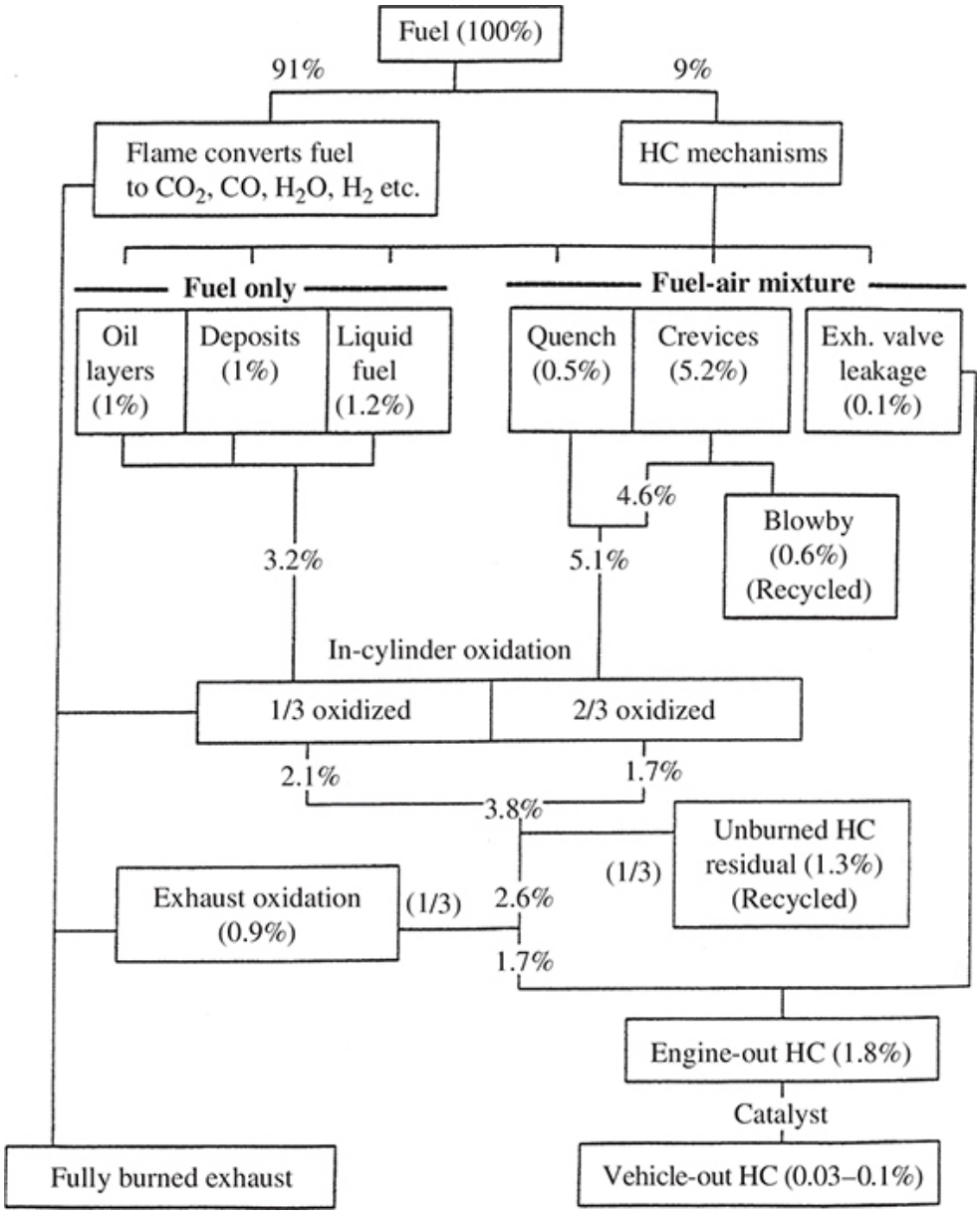


Figure 11.35 Flow chart for the gasoline fuel within each cylinder at the end of intake through both the normal combustion process and the total HC emissions mechanism. Sources or processes by which some of the fuel escapes burning during the normal combustion process are shown as boxes at the top of the diagram. In-cylinder oxidation, retention in the cylinder in the residual gas, and oxidation in the exhaust port and manifold, and catalyst are shown in sequence below. Numbers in parenthesis denote the HC emissions index (percent of gasoline in the cylinder at the start of each cycle) after each step in the total process. [34]

During the exhaust process, of the HCs *not oxidized* within the cylinder (3.8%) one-third is retained within the cylinder (at this part-load condition), and two-thirds are exhausted of which one-third oxidizes within the exhaust ports. Thus from the original 9% of the fuel that escaped burning during normal combustion, some 1.8% remains in the exhaust as engine-out HC emissions.

Note that through blowby to the engine crankcase (0.6%), which is returned to the engine's intake system and the 1.3% remaining in the cylinder with the residual gases, together, 1.9% of this 9% of fuel escaping normal combustion is recycled to the cylinder. Table 11.10 summarizes the percentage of fuel involved in each HC source, and the percent that source contributes to the total engine-out HC emissions. The importance of crevices is clear.

TABLE 11.10 HC sources: Magnitudes and percent of total engine-out HC emissions

Source	% of fuel escaping normal combustion	% of total HC engine-out emissions
Crevices	5.2	38*
Quench	0.5	5
Oil layers	1.0	16
Deposits	1.0	16
Liquid fuel	1.2	20
Exhaust valve leakage	0.1	5
TOTAL	9.0	100

*Blowby (0.6%) subtracted

The loss in imep is less than this 9% by the amount that in-cylinder HC oxidation contributes to work transferred to the piston ($\int pdV$) during expansion. This HC-oxidation expansion-stroke work contribution decreases from unity if it occurs during normal combustion at close to TC (which achieves close to the full expansion possible) to close to zero between EVO and BC. Post-normal-combustion in-cylinder oxidation of HCs is likely to contribute, on-average, about one-third of this. Thus, this additional imep contribution would be about $[(3.2 + 5.1) - (2.1 + 1.7)] \div 3 \approx 1.5\%$. So the overall imep loss is $(9.0 - 1.5) \approx 7.5\%$. The fuel conversion efficiency loss

benefits from the blowby and residual being “recycled”: that is (0.6 + 1.3) ≈ 1.9% for an efficiency loss of (7.5 - 1.9) ≈ 5.6%. It is often not realized that the engine-out HC emissions of about 2% actually represent a three to four times larger performance and efficiency loss.

11.4.4 Hydrocarbon Emission Mechanisms in Diesel Engine

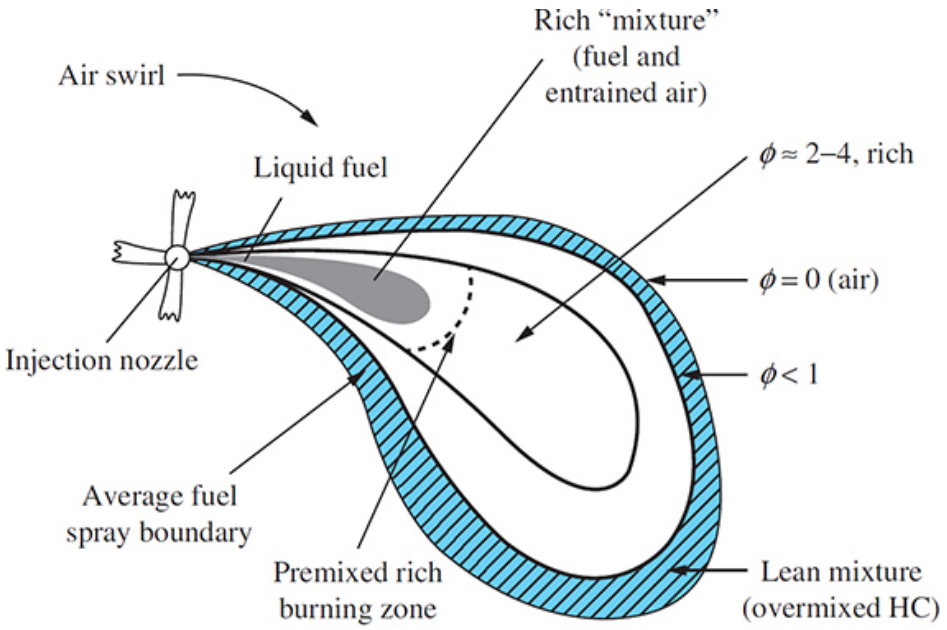
Background

Diesel fuel contains hydrocarbon compounds with higher molecular weights, and hence higher boiling points, than gasoline. Also, substantial pyrolysis of fuel compounds occurs within the fuel sprays during the diesel combustion process. Thus, the composition of the unburned and partially oxidized hydrocarbons in the diesel exhaust is complex, different from that in the spark-ignition engine exhaust, and extends over a larger molecular size range. Gaseous hydrocarbon emissions from diesels are measured using a hot particulate filter (at 190°C) followed by a heated flame-ionization detector. The HC constituents vary from methane to the heaviest hydrocarbons, which remain in the vapor phase in the heated sampling line (which is also maintained at about 190°C). Any hydrocarbons heavier than this are condensed and, with the solid-phase soot, are filtered from the exhaust gas stream upstream of the detector. The particulate emission measurement procedure includes a portion of total engine hydrocarbon emissions. Particulates are collected by filtering from a diluted exhaust gas stream at a temperature of 52°C or less. Those hydrocarbons that condense at or below this temperature are absorbed onto the soot. They are the *extractable fraction* of the particulate: that is, that fraction which can be removed by a powerful solvent, typically between about 15 and 45% of the total particulate mass. This section discusses gaseous hydrocarbon emissions; particulate emissions—soot and extractable material—are discussed in Sec. 11.5 .

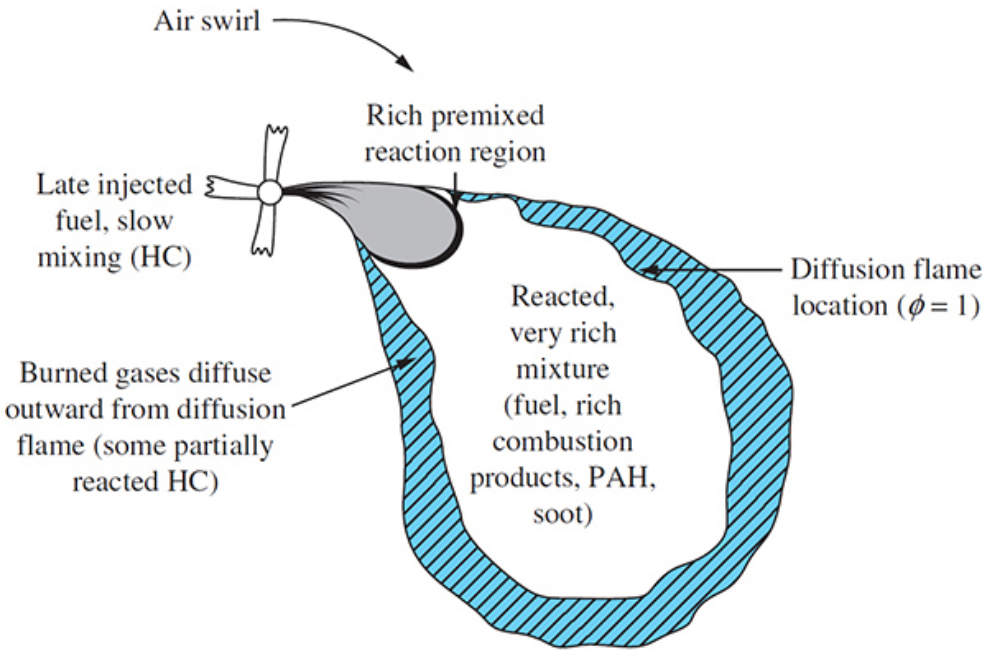
The complex heterogeneous nature of diesel combustion, where fuel evaporation, fuel vapor, air, and burned gas mixing, and rich premixed combustion followed by diffusion flame burn up, can occur in sequence and simultaneously, has been discussed extensively in Chap. 10 . As a result of this complexity, there are several processes that could contribute to diesel engine hydrocarbon emissions. In Chap. 10 the combustion process was

divided into four stages: (1) the ignition delay which is the time between the start of injection and autoignition; (2) the premixed or rapid combustion phase, during which much of the fuel that has mixed in each spray with entrained air during the delay period then partially burns; (3) the mixing-controlled diffusion-flame combustion phase, during which the rate of burning depends on the rate of fuel-air mixing; and (4) the late combustion phase where heat release continues at a low rate governed by the mixing of residual combustibles with excess oxygen and the kinetics of the oxidation process. There are two primary paths by which hydrocarbons can escape this diesel combustion process unburned: the fuel-air mixture can become too lean to autoignite or to support a propagating flame at the conditions prevailing inside the combustion chamber: or, during the primary combustion process, the fuel-air mixture may be too rich to ignite. This fuel can then be consumed only by slower thermal oxidation reactions later in the expansion process which, because the gas temperatures are lower, are significantly slower. Thus, hydrocarbons remain unconsumed due to quenching of the oxidation process.[j]

Figure 11.36 illustrates schematically how these processes can produce incomplete combustion products. Fuel injected during the ignition delay (Fig. 11.36*a*) will vaporize and mix within each fuel spray with air entrained into the spray, producing rich fuel vapor/air mixture ($\phi \approx 2\text{-}4$) in the bulk of the spray. Around the edges of these turbulent fuel sprays, more air is available and locally some fuel will "overmix" with air producing lean mixture that will be difficult to autoignite and burn at conditions prevailing inside the combustion chamber (though some of this mixture may burn later if it mixes with high-temperature burned products early enough in the expansion stroke). In the "premixed" burning phase which follows the delay, ignition occurs in the bulk of the spray where the rich mixture is now hot enough to autoignite and release a portion of its chemical energy. Complete combustion of this over-rich mixture depends on further mixing with air in the diffusion flame surrounding each spray, or later still after mixing with already-burned gases before further expansion and cooling occurs. Of all these possible mechanisms, the overlean mixture path is believed to be the most important.[21]



(*a*) Ignition delay and premixed-burn phase



(*b*) Mixing-controlled burning phase

Figure 11.36 Schematic representation of the diesel HC formation mechanism identifying its several components: (*a*) during the ignition delay and premixed burning phase; (*b*) during the mixing-controlled diffusion-flame burning phase.

For the fuel injected after the ignition delay period is over (Fig. 11.36*b*), rapid oxidation of the partly-burned fuel and products of fuel pyrolysis, as these mix with air in the diffusion flame that surrounds each spray, results in complete combustion. However, slow mixing of fuel and pyrolysis products with air, resulting in over-rich mixture or quenching of the combustion reactions, can also result in incomplete combustion products, pyrolysis products, and unburned fuel being present in the exhaust. [21]

Hydrocarbon emission levels from diesels vary widely with operating conditions, and the different HC formation mechanisms are likely to be more or less important at different operating modes. Engine idling and light-load operation produce significantly higher hydrocarbon emissions than full-load operation. However, when the engine is overfueled, HC emissions increase very substantially. As is explained more fully below, overmixing (overleaning) is an important source of HC, especially under light-load operation. Undermixing, resulting in over-rich mixture during the combustion period, is the mechanism by which some of the fuel remaining in the injector-nozzle sac volume escapes combustion, and is also the cause of the high HC emissions during overfueling. Wall temperatures affect HC emissions suggesting that wall quenching of the combustion chemistry may be important, also.

Overleaning

As soon as fuel injection into the cylinder commences, a distribution in fuel/air equivalence ratio across each fuel spray develops. The amount of fuel that is mixed leaner than the lean combustion limit ($\phi \approx 0.3$) increases rapidly with time, as illustrated in Fig. 11.36 *a*. In each fuel spray core, the mixture is significantly rich of stoichiometric. However, fuel close to the spray boundary has already mixed beyond the lean limit of combustion and will not autoignite. This mixture can only oxidize by relatively slow thermal-oxidation reactions which may not be complete. Within this region, unburned fuel, fuel decomposition products, and partial oxidation products (aldehydes and other oxygenates) will exist; some of these will escape the cylinder without being burned. The magnitude of the unburned HC from these overlean regions will depend on the amount of fuel injected during the ignition delay, the mixing rate with air during this period, and the extent to which prevailing cylinder conditions are conducive to autoignition. A correlation of unburned HC emissions with the length of the ignition delay would be expected. The

data in Fig. 11.37 from a direct-injection naturally-aspirated engine show that a good correlation between these variables exists. As the delay period increases beyond its minimum value (due to changes in engine operating conditions) the HC emissions increase at an increasing rate. [60] Thus, overleaning of fuel injected during the ignition delay period is a significant source of hydrocarbon emissions, especially under conditions where the ignition delay is long.

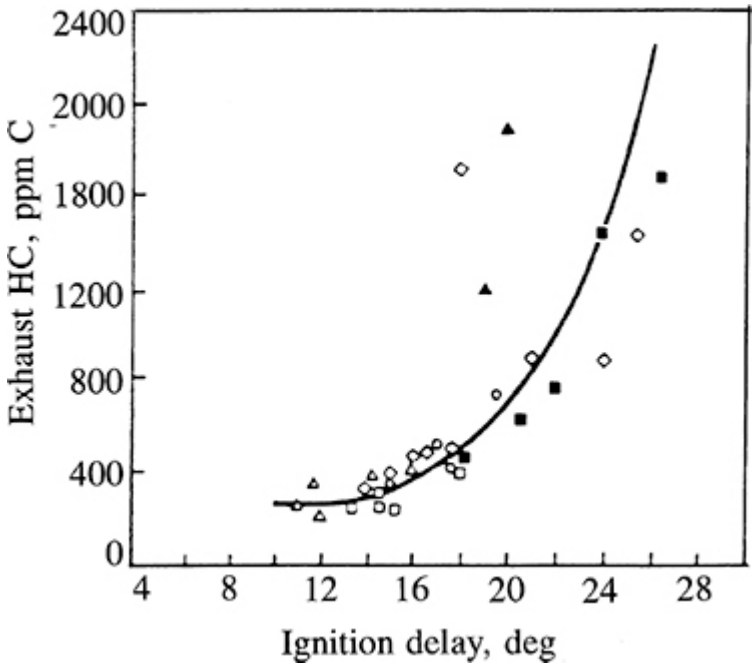


Figure 11.37 Correlation of exhaust HC concentration with duration of ignition delay for DI diesel engine. Various fuels, engine loads, injection timings, boost pressures, at 2800 rev/min. [60]

Undermixing

Two sources of fuel which enter the cylinder during combustion and which result in HC emissions due to slow or under mixing with air have been identified. One is fuel that leaves the injector nozzle at low velocity, often late in the combustion process. The most important source here is the nozzle sac volume, though secondary injections can increase HC emissions if the problem is severe. The second source is the excess fuel that enters the cylinder under overfueling conditions.

At the end of the fuel-injection process, the injector sac volume (the small volume left in the tip of the injector after the needle seats) is left filled with fuel. As the combustion and expansion processes proceed, this fuel is heated and vaporizes, and enters the cylinder at low velocity through the nozzle holes. This fuel vapor (and perhaps drops of fuel also) will mix relatively slowly with air and may escape the primary combustion process. Figure 11.38 shows HC emissions at the minimum ignition delay for a direct-injection diesel engine as a function of sac volume, along with drawings of some of the injector nozzles used. The correlation between HC emissions (under conditions when the overleaning mechanism is least significant) and sac volume is striking. Not all the fuel in the sac volume is exhausted as unburned hydrocarbons. For example, in Fig. 11.38, a volume of 1 mm^3 gives 350 ppm C_1, while 1 mm^3 of fuel would give 1700 ppm C_1. The sac volume may not be fully filled with fuel. Also, the higher-boiling-point fractions of the fuel may remain in the nozzle. Significant oxidation may also occur. The sac volume in current production nozzles helps to equalize the fuel pressures immediately upstream of the nozzle orifices. A small sac volume makes this equalization less complete, and exhaust smoke levels deteriorate.

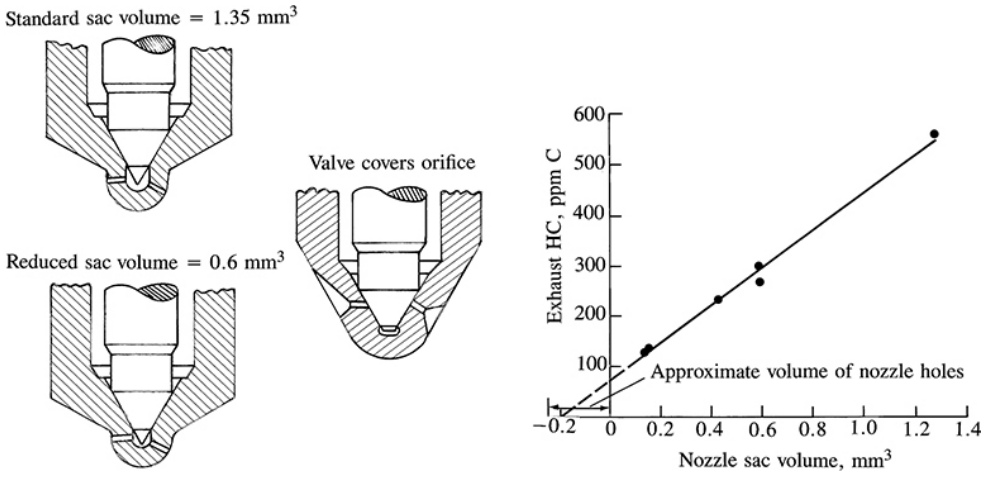


Figure 11.38 Effect of nozzle sac volume on exhaust HC concentration, DI diesel engine, at minimum ignition delay. V_d = 1 liter/cylinder, 1700 to 2800 rev/min.[60]

In DI engines, black exhaust smoke limits the full-load equivalence ratio to about 0.7. Under transient conditions as the engine goes through an acceleration process, overfueling can occur. Even though the overall equivalence ratio remains lean, locally over-rich conditions may persist through the expansion stroke and into the exhaust process. HC emissions are unaffected by an increasing equivalence ratio until a critical value of about 0.9 is reached when levels increase dramatically. This mechanism is not significant under normal operating conditions, but can contribute HC emissions under heavy acceleration conditions if overfueling occurs.

Quenching and Misfire

Hydrocarbon emissions have been shown to be sensitive to oil and coolant temperature. For example, when these temperatures were increased from 40 to 90°C in a DI diesel, HC emissions decreased by 30%. Since the ignition delay was maintained constant, overmixing phenomena should remain approximately the same. Thus, wall quenching of the flame may also be a significant source of HC, depending on the degree of spray impingement on the combustion chamber walls.

While cycle-by-cycle variation in the combustion process in diesel engines is generally much less than in spark-ignition engines, it can become significant under adverse conditions such as low compression temperatures and pressures, and retarded injection timings. Substantial variations, cycle-by-cycle, in HC emissions are thought to result. In the limit, if misfire (no combustion) occurs in a fraction of the operating cycles, then engine HC emissions rise as the percentage of misfires increases. However, complete misfires in a well-designed and adequately controlled diesel engine are unlikely to occur within its normal operating range. [21]

Engine-out hydrocarbon emissions from diesel engines are low (of order 0.1% of the fuel) over much of the engine's operating range. However, at very light loads (especially engine idle), HC emissions as a fraction of the fuel can be several times higher. The very lean in-cylinder mixture at these loads results in a much larger percentage of the injected fuel overmixing and, as a result, not burning. Table 11.11 provides two examples of HC and particulate emissions (and the volatile organic fraction of that particulate extracted with a solvent—heavier HC that condense in the exhausting and sampling systems) for a heavy-duty 12-liter and 1.9-liter light-duty diesel engine. Both engines were turbocharged and intercooled. The light-duty

passenger car engine was fitted with a diesel oxidation catalyst (DOC; see Sec. 11.6.3), which significantly reduces both the vehicle-out HC emissions and the fraction of the particulate that is absorbed volatile organic compounds. The heavy-duty diesel engine, at idle (mode 4) emits about five times the HC as a fraction of the fuel than at low to mid loads. But the fuel injected per cycle is significantly lower (about 2 kg/h for mode 4, compared to 15 to 25 for the other modes). The oxidation catalyst used with the light-duty diesel engine is effective at reducing the HC emissions above about 40% of the maximum torque, but much less effective at light load and idle (modes 4 and 5) when the catalyst temperature is significantly lower. The catalyst also reduces the soluble (volatile) organic fraction of the particulate emissions significantly. [61]

TABLE 11.11 HC, particulate, and soluble organic fraction emissions at different speeds and loads [61]

Mode	Speed, rev/min	Load, %	HC, g/kg fuel	TPM, g/kg fuel	SOF, % TPM
Heavy-duty diesel engine*					
1	1700	31	0.94	1.07	32
2	1400	17	1.44	0.87	43
3	1200	36	0.65	0.46	43
4	800	Min.	5.64	1.25	74
Light-duty diesel engine (with oxidation catalyst)†					
1	2620	36	0.17	0.50	12
2	2500	76	0.06	0.98	31
3	1900	50	0.07	0.60	6
4	1400	20	1.11	0.38	12
5	1200	Min.	2.44	0.53	22

HC = hydrocarbons; TPM = total particulate mass; SOF = soluble organic fraction of particulate.
*Six-cylinder 12-liter turbocharged aftercooled DI diesel engine rated at 312 kW at 2100 rev/min, peak torque 2237 Nm at 1200 rev/min.
†Four-cylinder 1.9-liter turbocharged aftercooled DI diesel engine rated at 66 kW. Exhaust includes a diesel oxidation catalyst.

While basic diesel engine-out HC emissions levels are low, multiple fuel-injection strategies are now being used which, with their several start and end-of-injection processes, will complicate and likely worsen HC emissions (see Sec. 10.7.1). Also, the advanced diesel combustion modes being explored and developed, where "premixing" of fuel and air well before the normal combustion process occurs, has significantly lower engine-out NO_x emissions but several times higher HC and CO emissions than a conventional diesel (see Sec. 10.7.2). Also, as exhaust particulate traps, NO_x reduction catalysts, and DOCs are now part of the total diesel engine's emission control system, the hydrocarbons in the exhaust gases play a role in the operation of these exhaust treatment devices. The HC can act as a source of thermal energy as they react in these devices: they also can act as a reducing agent as NO_x is reduced to N_2, or as a source of active species as they oxidize. Depending on the exhaust system devices and regeneration strategies utilized, the HC concentrations may be deliberately increased (e.g., by post-main and/or late fuel injections) to enhance emissions control.

11.5 PARTICULATE EMISSIONS

Particulate emissions have long been an issue with diesels. In such engines, as described in Chap. 10 , soot is formed within the evolving fuel sprays, and while much of the soot burns up inside the cylinder, a portion of it does not and is exhausted. Then, in the exhaust and during dispersion of the gases exiting the exhaust pipe, the gases cool, nucleate condense, and form small particles. As higher-molecular-weight hydrocarbons continue to condense, they are absorbed into the soot particles. Thus a size distribution of particulates builds up: the larger particles are categorized as *accumulation mode* (particles in the diameter range of 50 to 500 nm); the small particles below about 50 nm in diameter constitute the *nucleation mode.* With port-fuel-injection gasoline engines, the particulate emissions are much lower than in diesels: by a factor of about 70 on a mass basis. However, direct-injection gasoline engine particulate emissions are close to a factor of 10 higher than those from port-fuel-injection equivalent engines, again by mass.

Particulate traps located in the exhaust system of diesel and direct-injection gasoline engines have been developed, and are widely used in

diesels. These traps remove almost all the particulates from the exhaust gases using the thin walls of porous ceramic monoliths as a filter. Over time the pressure drop across the particulate trap increases, and the ceramic filter must be regenerated. Usually this requires that its temperature be increased so the particulate layer on the filter burns off. See Sec. 11.6.4 for a more detailed discussion of particulate traps.

11.5.1 Spark-Ignition Engine Particulates

Traditionally, three classes of spark-ignition engine particulate emissions have been defined: inorganic particles (of which lead compounds used to dominate), organic particulates (including soot), and sulfates.

Depending on the sulfur content in the fuel, significant sulfate emissions can occur with oxidation-catalyst equipped engines. Gasolines can contain from 10 up to some 500 ppm by weight sulfur (depending on the extent of sulfur removal in the refinery), which is oxidized within the engine cylinder to sulfur dioxide. This SO_2 can be oxidized by the exhaust catalyst to SO_3 which combines with water at ambient temperatures to form a sulfuric acid aerosol. Levels of sulfate emissions depend on the fuel sulfur content, the operating conditions of the engine, and the details of the catalyst system used. Low sulfur levels in gasolines are important to maintain high exhaust catalyst efficiencies over extended use.

For automobile engines operated with leaded gasolines (which contain about 0.15 g Pb/liter), the particulate emission rates are typically 100 to 150 mg/km. This particulate is dominated by lead compounds: 25 to 60% of the emitted mass is lead. [62] The particulate emission rates are considerably higher when the engine is cold, following start-up. The exhaust temperature has a significant effect on emission levels. The particle size distribution with leaded fuel is about 80% by mass below 2 μm diameter and about 40% below 0.2 μm diameter. Most of these particles are presumed to form and grow in the exhaust system due to vapor phase condensation enhanced by coagulation. Some of the particles are emitted directly, without settling. Some of the particles either form or are deposited on the exhaust system walls. Many of these are removed when the exhaust flow rate is suddenly increased and, together with rust and scale, account for the increase in mass of particles emitted during accelerations. Only a fraction of the lead

consumed in the fuel is exhausted, the remainder being deposited within the engine and exhaust system.

Current port-fuel-injected unleaded-gasoline SI engines have particulate emissions of a few mg/km in automobiles without catalysts. Over the next decade, particulate emissions requirements are likely to be reduced by up to an order of magnitude. The particulate is primarily condensed organic materials, though soot emissions (black smoke) can result from the burn up of overly rich mixtures. Direct-injection gasoline engines have higher particulate emissions than do port-injected engines by up to a factor of 10 on a mass basis, and are similarly higher based on particulate number density. Stratified charge direct-injection engines have much higher particulates (more nucleation particles) than do homogeneous stoichiometric DI engines. The following example illustrates the various ways that are used to characterize a vehicle's (and engine's) particulate emissions: for example, mass and number of particles emitted per unit distance traveled, particulate number density in the exhaust, particulate emissions index [Eq. (2.44)] g PM/kg fuel.

EXAMPLE 11.3

Particulate emissions are expressed in many ways: mg particulate matter (PM)/km, particle number density PN/cm ^{3}, g PM/kg fuel, #PN/km. The 2014–2017 European PN limit for passenger vehicles is 6×10^{11} #/km: convert this particulate emissions level to the other parameters listed above.

Needed inputs are:

(*a*) Vehicle fuel economy/consumption 35 miles per gallon gasoline is equivalent to $35 \times 0.42 = 15$ km/liter which corresponds to 6.7 liters/100 km. The fuel mass-consumption rate is 50 g/km or 80 g/mile. The density of gasoline is 0.75 kg /liter = 0.75 g/cm ^{3}.

(*b*) Stoichiometric gasoline/air mixture: thus the mass of air + fuel (equals the mass of exhaust), relative to mass of fuel, is

$$(m_a + m_f)/m_f = 15.6$$

The density of exhaust gas at standard pressure and temperature (≈density of air) is 1.2 kg/m ^{3}.

(*c*) Density of PM (close to the density of soot) is about 2 g /cm ^{3}. The average particle size is 80 nm diameter.

Starting with 6 × 10 #PN/km, find #PN/cm ^{3} of exhaust. The multiplication sequence is:

$$6\times10^{11}\left(\frac{\#}{\text{km}}\right)\times\frac{100}{6.7}\left(\frac{\text{km}}{\text{lit.fuel}}\right)\times\frac{1}{10^3}\left(\frac{\text{lit}}{\text{cm}^3}\right)\times\frac{1}{0.75}\left(\frac{\text{cm}^3\text{fuel}}{\text{g fuel}}\right)\times\frac{1}{15.6}\left(\frac{\text{g fuel}}{\text{g exh}}\right)\times1.2\times10^{-3}\left(\frac{\text{g exh}}{\text{cm}^3\text{exh}}\right)$$
$$=\underline{9.2\times10^5\,\#\text{PN/cm}^3\text{exh}}$$

To convert 6 × 10 ^{11} #PN/km to mg PM/km, the multiplication sequence is:

$$6\times10^{11}\left(\frac{\#}{\text{km}}\right)\times\frac{4}{3}\pi\left(\frac{80\times10^{-7}}{2}\right)^3(\text{cm}^3)\times2\left(\frac{\text{gPM}}{\text{cm}^3\text{PM}}\right)\times10^3\left(\frac{\text{mg}}{\text{g}}\right)=\underline{0.32\text{ mg PM/km}}$$

To convert 0.32 mg PM/km to mg PM/g fuel, the multiplication sequence is:

$$0.32\left(\frac{\text{mg PM}}{\text{km}}\right)\times\frac{100}{6.7}\left(\frac{\text{km}}{\text{lit fuel}}\right)\times\frac{1}{0.75}\left(\frac{\text{cm}^3\text{fuel}}{\text{g fuel}}\right)\times10^3\left(\frac{\text{lit}}{\text{cm}^3}\right)=\underline{0.0064\text{ mg PM/g fuel}}$$

With injection of liquid fuel directly into the cylinder, there are two key questions that impact particulate formation: (1) Is there significant liquid fuel wetting of the piston crown, cylinder head and liner? (2) Is there time for essentially complete evaporation of the fuel? The questions are coupled since slower evaporation results in greater penetration of drops in the fuel spray: with the resulting increase in wall wetting, the evaporation rate and timing become even more critical. The solid component of the particulate is largely soot which forms as gasoline vapor evaporates from the liquid-fuel wall films and mixes with the hot burned gases (which contain modest amounts of oxygen) thus forming hot enough very fuel-rich regions—the conditions under which soot forms. And, as the engine overall shifts to rich of stoichiometric, accumulation mode PM increases since, with more fuel and less air, these rich hydrocarbon-vapor/burned gas zones are more extensive and richer. As fuel injection timing shifts later from the intake stroke (homogeneous DI) to during the compression stroke (lean stratified DI operation), there is less time for evaporation and mixing, and more wall wetting, so accumulation mode PM increases. As ethanol is added to the gasoline fuel, additional oxygen is available in these rich vaporized-fuel

burned-gas regions where soot forms, so its formation rate is reduced. Also, the period during and after engine start up would be expected to be a period when particulate formation rates are higher.

The nucleation mode particulate mostly come from higher molecular weight hydrocarbon components in the gasoline fuel which condense during the exhaust process, and from soot precursor nuclei which are quenched before significant agglomeration into larger particles can occur. (See Sec. 11.5.4 on soot formation mechanisms.)

11.5.2 Characteristics of Diesel Particulates

Measurement Techniques

Diesel particulates consist principally of combustion-generated carbonaceous material (soot) on which organic compounds in the exhaust gases have been absorbed. Most particulate material results from incomplete combustion of fuel hydrocarbons; some is contributed by the lubricating oil. The engine-out emission rates are typically 0.03 g/km for light-duty diesels in an automobile. In larger heavy-duty direct-injection engines, particulate emission rates are 0.5 to 1.5 g/brake kW · h. The composition of the particulate material depends on the conditions in the engine exhaust and particulate collection system. At temperatures above 500°C, the individual particles are principally clusters of many small spheres or spherules of carbon (with a small amount of hydrogen) with individual spherule diameters between 15 nm and 30 nm. As temperatures decrease below 500°C, the particles become coated with adsorbed and condensed high molecular-weight organic compounds which include: unburned hydrocarbons, oxygenated hydrocarbons (ketones, esters, ethers, organic acids), and polynuclear aromatic hydrocarbons. The condensed material also includes (depending on the sulfur content in the fuel) species such as sulfur dioxide, and sulfuric acid (or sulfates).

The objective of most particulate measurement techniques is to determine the amount of particulate being emitted to the atmosphere. Techniques for particulate measurement and characterization range from simple smoke meter opacity readings to analyses using exhaust dilution tunnels. Most techniques require lengthy sample-collection periods because the particulate emission rate is usually low. The physical conditions under which particulate

measurements are made are critical because the emitted species are unstable and may be altered through loss to surfaces, change in size distribution (through agglomeration), and chemical interactions among other species in the exhaust, at any time during the measurement process (including sampling, storage, or examination). The most basic information is normally obtained on a gravimetric or mass basis: a known fraction of the exhaust flow is sampled from the diluted exhaust; the particulate matter is filtered and weighed (under carefully controlled conditions). Emissions are then expressed in grams per kilometer for a vehicle, grams per kilowatt-hour for an engine, grams per kilogram of fuel or milligrams per cubic meter of exhaust (at standard conditions). Smoke meters measure the relative quantity of light that passes through the exhaust or the relative reflectance of particulate collected on filter paper. They do not measure mass directly. They are used to determine visible smoke emissions and provide an approximate indication of mass emission levels. Visible smoke from heavy-duty diesels at high load is regulated. In the standard mass emission measurement procedure, dilution tunnels are used to simulate the physical and chemical processes the particulate emissions undergo as they enter the atmosphere. In the dilution tunnel, the raw exhaust gases are diluted with ambient air to a temperature of 52°C or less, and a sample stream from the diluted exhaust is filtered to remove the particulate material.

Particulate counting, as part of efforts to reduce the number of fine particles emitted from diesel engines can be done with several instruments. The Condensation Particulate Counter (CPC) has been chosen for determining the number of particles for compliance with regulation. Particle size and distribution can be obtained with several devices: for example, a Scanning Mobility Particle Sizer (SMPS), an Electrical Low Pressure Impactor (ELPI), and a Differential Mobility Spectrometer (DMS). The *Bosch Automotive Handbook* briefly summarizes these several particulate measurement approaches and devices. [63]

Particulate Composition and Structure

As mentioned above, the particle size distribution in diesel exhaust is bi-modal with a nucleation mode (less than about 50 nm in diameter) and an accumulation mode (from about 50 to 500 nm diameter). Figure 11.39 is a schematic of the particle and gas phase processes, and the factors affecting these processes, from the start of combustion to exhaust discharge into the

atmosphere. Dispersion into the atmosphere, and transformation, then continue. Note that in today's engines, this sequence includes the in-cylinder combustion process, what occurs in the exhaust system and the aftertreatment devices (particulate filter and diesel oxidation catalyst) it contains, dilution with air and cooling as the exhaust gases exit the exhaust pipe, and diffusion and continuing dilution in the atmosphere.

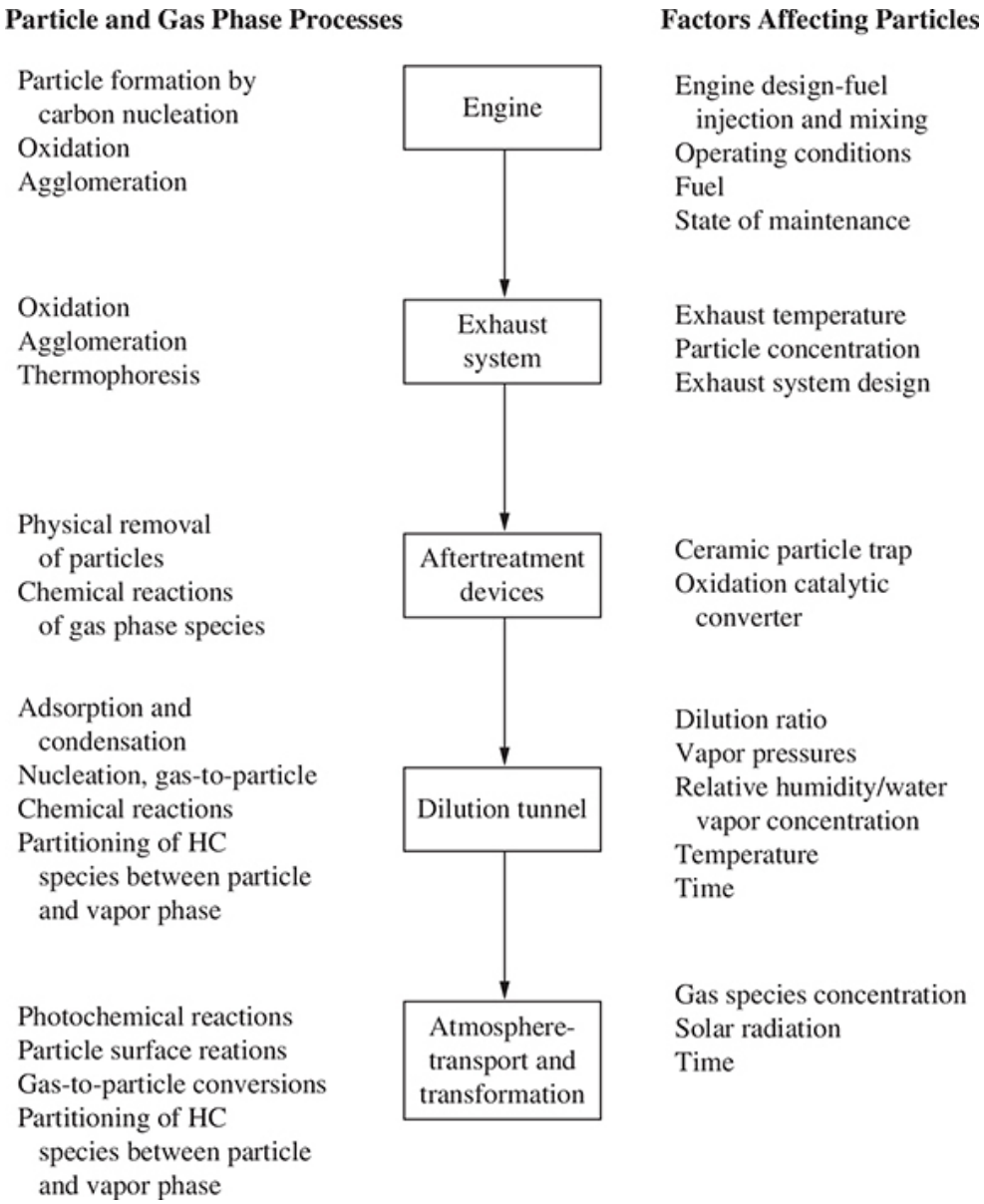


Figure 11.39 Schematic of the particle and gas phase processes and factors effecting the particle size distribution and chemical composition of diesel engine particulate matter emissions into the atmosphere.[64]

The use of a dilution tunnel to characterize the amount and composition of the particulates in the exhaust gas is meant to simulate in a controlled way the actual mixing, cooling and dispersion that occurs in practice as the engine's particulate laden exhaust gases enter the atmosphere. Examples of these particulates are shown in Fig. 11.40. In Fig. 11.40 *a* and *b*, the gases were sampled from the combustion chamber at 150° ATC when the exhaust valve is opening. The samples are seen to consist of collections of primary particles (spherules) of 10 to 25 nm diameter, agglomerated into aggregates (hereafter called particles). Individual particles range in size and appearance from clusters to chains of spherules. During combustion, the sampled aggregate particles are significantly larger: by EVO their overall size has been reduced yet the agglomerated spherule structure persists. At EVO clusters may contain some 20 to 100 or so spherules. The spherules are combustion-generated soot particles mostly in the 10- to 25-nm size range. [65] Figure 11.40*c* shows a typical accumulation-mode particle and several much smaller, spherical, nucleation-mode particles. These were obtained by sampling from a diesel engine exhaust, then diluted, and filtered. An accumulation particle, a chain-like collection of many soot spherules, and several small nucleate mode aerosol particles are illustrated. [66]

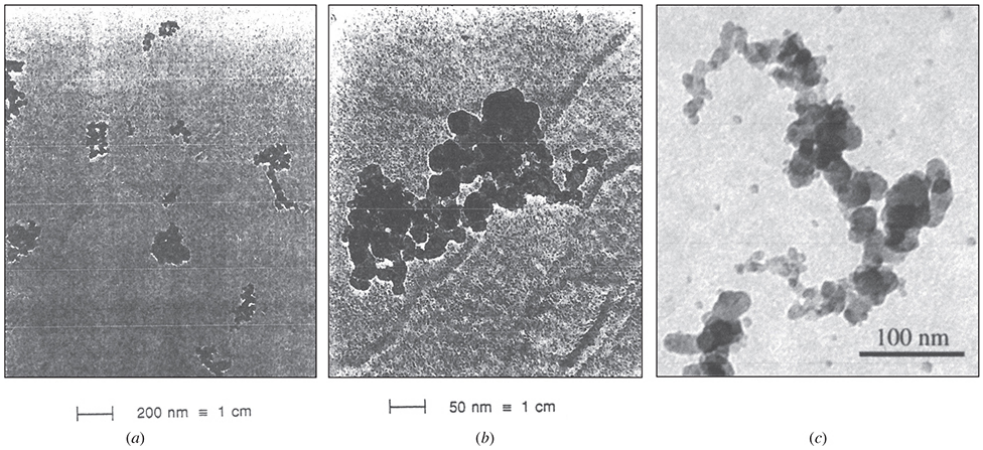


Figure 11.40 Photomicrographs of diesel particulates sampled from the in-cylinder gases, or the diluted exhaust. In-cylinder samples extracted at 150°CA ATC (near EVO) at 2000 rev/min, 5 bar bmep, 31% EGR. Magnification: (*a*) 50k and (*b*) 200k. [65] (*c*) Sample from engine exhaust, then diluted and collected on transmission electromicroscopy grid: shows chain-like accumulation-mode particle and several small spherical nucleation-mode particles. [66]

Determination of the particle size distribution from such photomicrographs involves assigning a representative dimension to such complex and irregular aggregates, and introduces uncertainties arising from only having two-dimensional images of particles available. Other approaches based on inertial impactors and electrical aerosol analyzers have been used. The data indicate that the particle size distribution is bimodal. The smaller-size nucleation mode is composed of liquid hydrocarbon drops, individual spherules, and sulfate (H_2O-H_2SO_4) nuclei. The larger-size accumulation mode is of agglomerated spherule particles characterized by number-mean diameters of 100 to 150 nm. Figure 11.41 illustrates this discussion of the different components of diesel exhaust particulates. An additional feature that it adds is that higher molecular weight hydrocarbons are absorbed onto individual spherule and agglomerated particle surfaces. These hydrocarbons range from about C_{20} to C_{30}. Condensed hydrocarbons and sulfates, and individual soot spherules, comprise the small, nucleation mode, particles. [65]

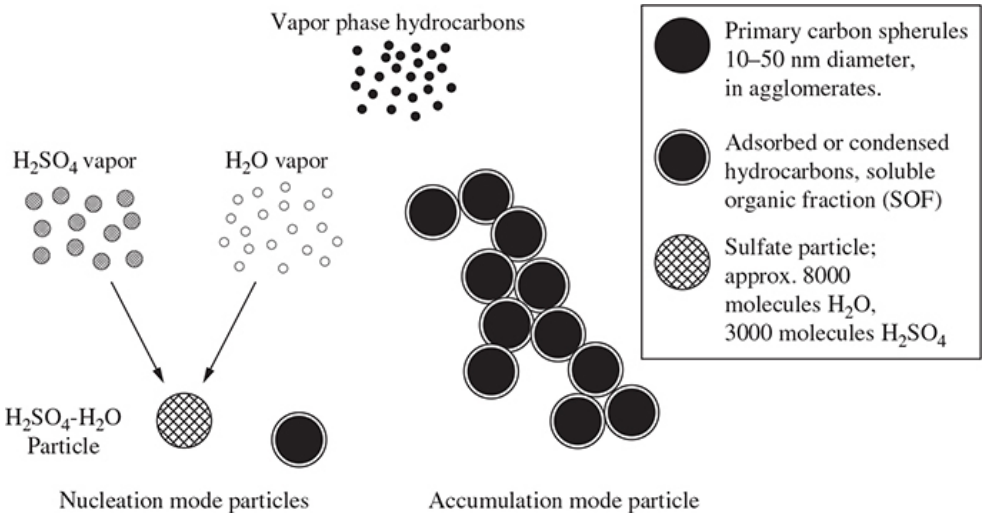


Figure 11.41 Schematic illustration of components of diesel particulate matter: accumulation mode; largely soot spherules, which agglomerate to form larger particles: nucleation mode; condensed hydrocarbon particles, individual soot spherules, and sulfate particles; these compounds are also absorbed onto the agglomerates.[65]

The exhaust particulate is usually partitioned with an extraction solvent into a soluble fraction and a dry-soot fraction. Two commonly used solvents are dichloromethane and a benzene-ethanol mixture. Typically 15 to 30 mass percent is extractable, though the range of observations is larger. Thermogravimetric analysis (weighing the sample as it is heated) produces comparable results. Typical average chemical compositions of the two particulate fractions are given in Table 11.12. Dry soot has a much lower H/C ratio than the extractable material. Although most of the particulate emissions are formed through incomplete combustion of fuel hydrocarbons, engine oil may also contribute significantly. The number-average molecular weight of the extractable material shown in Table 11.12 ranged from about 360 to 400 for a variety of engine conditions. This fell between the average molecular weight of the fuel (199) and that of the lubricating oil (443 when fresh and 489 when aged). [67] All of the oil contribution appeared in the extractable material. The contributions from the different individual compounds in the fuel have also been studied. All the compounds examined —paraffins, olefins, and aromatics—contributed to the particulate emissions; as a group, aromatics were the greatest contributors.

TABLE 11.12 Chemical composition of particular matter [67]

	Idle	48 km/h
Extractable composition	$C_{23}H_{29}O_{4.7}N_{0.21}$	$C_{24}H_{30}O_{2.6}N_{0.18}$
H/C	1.26	1.63
Dry-soot composition	$CH_{0.27}O_{0.22}N_{0.01}$	$CH_{0.21}O_{0.15}N_{0.01}$
H/C	0.27	0.21

In addition to the elements listed in Table 11.12, trace amounts of sulfur, zinc, phosphorus, calcium, iron, silicon, and chromium have been found in particulates. Sulfur and traces of calcium, iron, silicon, and chromium are found in diesel fuel: zinc, phosphorus, and calcium compounds are used in lubricating oil additives. [67] The sulfate (SO_4) associated with diesel particulate matter is collected during dilution tunnel sampling. It originates with the sulfur in the fuel, which is oxidized to SO_2, then to SO_3, which combines with H_2O to form H_2SO_4 vapor. Depending on local conditions (temperature and concentration/partial pressure) as the exhaust is diluted and cooled, multiple H_2O and H_2SO_4 vapor molecules collide and form molecular clusters which may then further grow. Alternatively, depending on local gas conditions, the molecular clusters of H_2O and H_2SO_4 may not have sufficient energy to be stable and eventually break up. Whether or not sulfate nucleation mode particles form depends on the sulfur level in the fuel and the thermodynamic state of the exhaust and the kinetics of the system. [64] In regions of the world where controlling vehicle emissions has made the greatest progress, fuel sulfur levels are low (below 15 ppm S by weight). Historically sulfur levels in distillate diesel fuels have been much higher (up to 5000 ppm and higher), especially in developing world regions. Fuel sulfur levels are, over time, being reduced. [k]

A lattice image of a portion of a diesel particle is shown in Fig. 11.42; it suggests that a concentric lamellate structure arranged around the center of each spherule. This arrangement of concentric lamellas is similar to the structure of carbon black. This is not surprising; the environment in which diesel soot is produced is similar to that in which oil furnace blacks are made. The carbon atoms are bonded together in hexagonal face-centered

arrays in planes, commonly referred to as platelets. As illustrated in Fig. 11.43, the mean layer spacing is 0.36 nm (only slightly larger than graphite). Platelets are arranged in layers to form crystallites. Typically, there are two to five platelets per crystallite, and on the order of 10^3 crystallites per spherule. The crystallites are arranged with their planes more or less parallel to the particle surface. The spherules, 10 to 25 nm in diameter, are fused together to form particles as shown in Fig. 11.41. A single spherule contains 10^5 to 10^6 carbon atoms. [68,69]

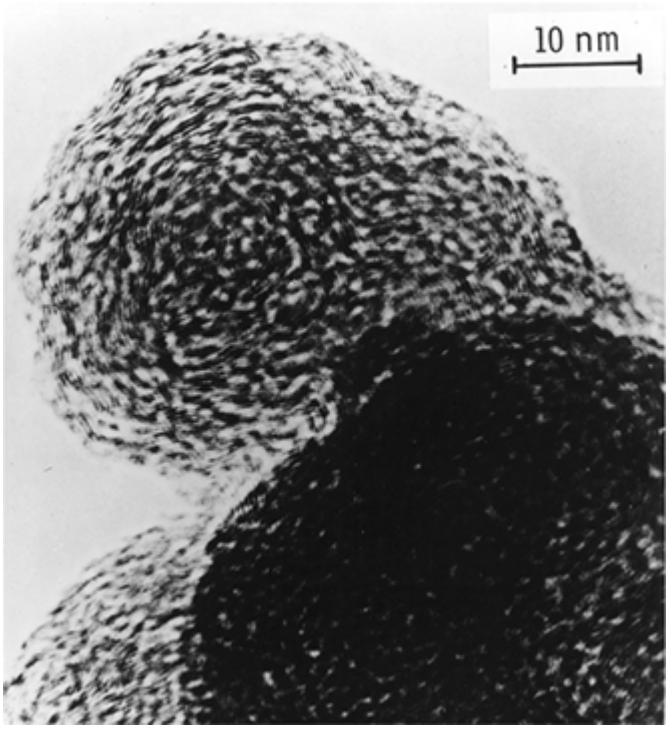


Figure 11.42 Lattice-imaging micrograph of spherules in an accumulation-mode diesel particle.[68]

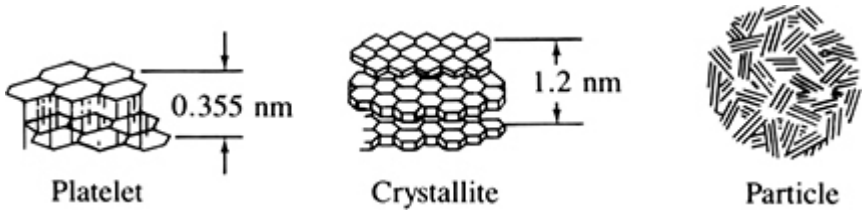


Figure 11.43 Substructure of carbon particles.[69]

A surface area of about 200 m^2/g has been measured for diesel soot. A smooth-surfaced 20-nm diameter sphere (a spherule with a density of 2 g/cm^3) would have a surface/mass ratio of 150 m^2/g, so the measured value is about 1.3 times the superficial area. Approximating a particle of agglomerated spherules by a single sphere of 200 nm diameter gives a surface/mass ratio of 15 m^2/g. [69] These data and estimates of superficial area per unit mass indicate that diesel soot has relatively low porosity.

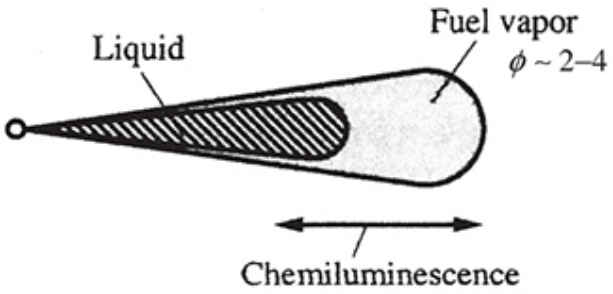
11.5.3 Particulate Distribution within the Cylinder

As the above discussion indicates, the solid component (soot) forms first within the fuel-rich core of the individual fuel sprays, once combustion has been initiated. The soluble organic fraction (SOF) of the particulate (the condensed hydrocarbon component) and nucleation mode particles form later during the exhaust process, and as dilution with air occurs as the exhaust gases leave the tailpipe. Many measurements have been made of the particulate distribution within the combustion chamber of operating diesel engines. The results provide valuable information on the particulate formation and oxidation processes and how these relate to the in-cylinder fuel distribution and heat-release development. Techniques used to obtain particulate concentration data include: use of rapid-acting poppet or needle valves which draw a small gas sample from the combustion chamber at a specific location and time for analysis, optical absorption techniques, and cylinder dumping where the cylinder contents are rapidly emptied into an evacuated tank at a preset time in the cycle. Concentration data taken at specific locations within the combustion chamber during the engine cycle are not necessarily representative of the cylinder contents in general: nor do they represent the time history of a given mass of gas. The fuel distribution, mixing, and heat-release patterns in the cylinder are highly nonuniform during the soot-formation process.

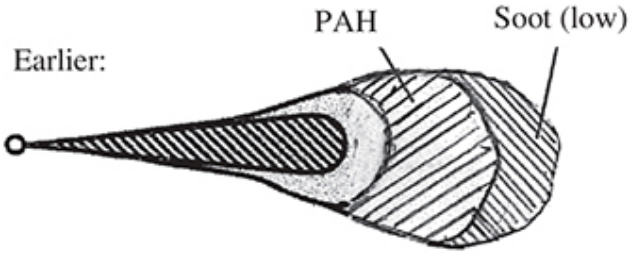
In direct-injection diesel engines, the soot component of the particulate forms within each fuel spray where local average equivalence ratios are very rich (see Secs. 10.3.4 and 10.7.2). Soot concentrations rise rapidly soon after combustion starts. A large fraction of the fuel molecules in each fuel-rich spray core react to form polycyclic aromatics, which grow to form soot particles. The fraction of local fuel carbon found as soot in these very fuel-

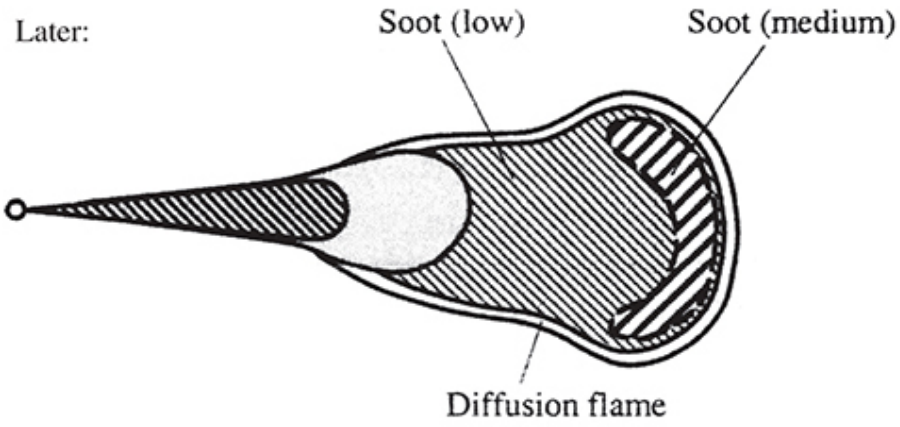
rich regions is high, some 50%. The sequences of physical and chemical processes that lead to soot formation are illustrated in Figs. 11.44 and 11.45. Figure 11.44 shows how soot develops in each individual fuel spray in a quiescent DI diesel combustion chamber. During the ignition delay period the spray penetrates into the combustion chamber, entraining hot end-of-compression air. The fuel droplets evaporate forming a fuel-rich ($\phi \approx 2\text{-}4$) mixture which, after a few crank-angle degrees, starts to react chemically, emit chemiluminescence, and release some 20% of its chemical energy. The autoignition chemistry breaks down the fuel molecules into polyacetylenes, which combine to form polycyclic aromatics. These molecules grow rapidly within the spray to ever larger molecules which form soot particle nuclei which grow into soot spherules and agglomerate into accumulation mode particles, as the mixture within the spray moves away from the nozzle. Thus, as shown, the soot concentration increases along the spray (now surrounded by a diffusion flame) from the rich premixed combustion zone some halfway along the spray toward the head. As fuel-rich, partly reacted, fuel-air mixture diffuses outward into the diffusion flame now surrounding each spray, the soot particles rise in temperature, and with the oxygen available from the air side of the flame, largely burn up.

Ignition delay:

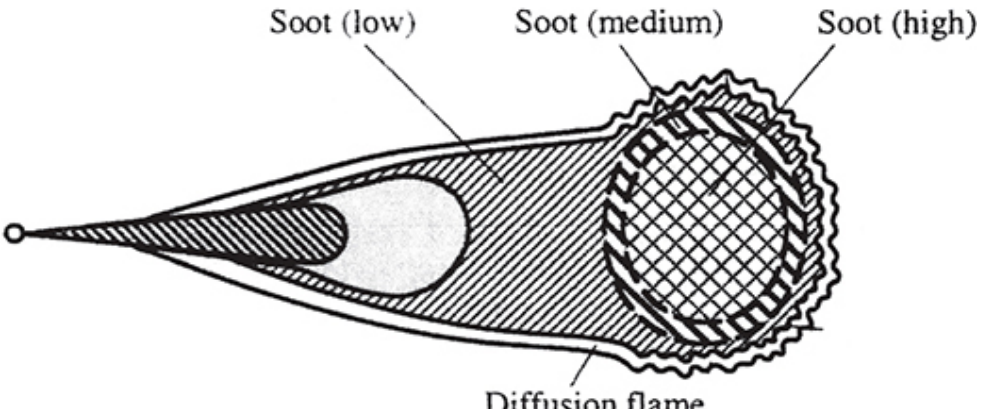


Rapid burning phase





Mixing conrolled phase



Diffusion flame

Figure 11.44 Schematics illustrating the critical soot forming processes, and the regions where PAH and soot form in an individual diesel fuel spray during the ignition delay, premixed rapid burning phase, and mixing-controlled diffusion flame phase of combustion.[70]

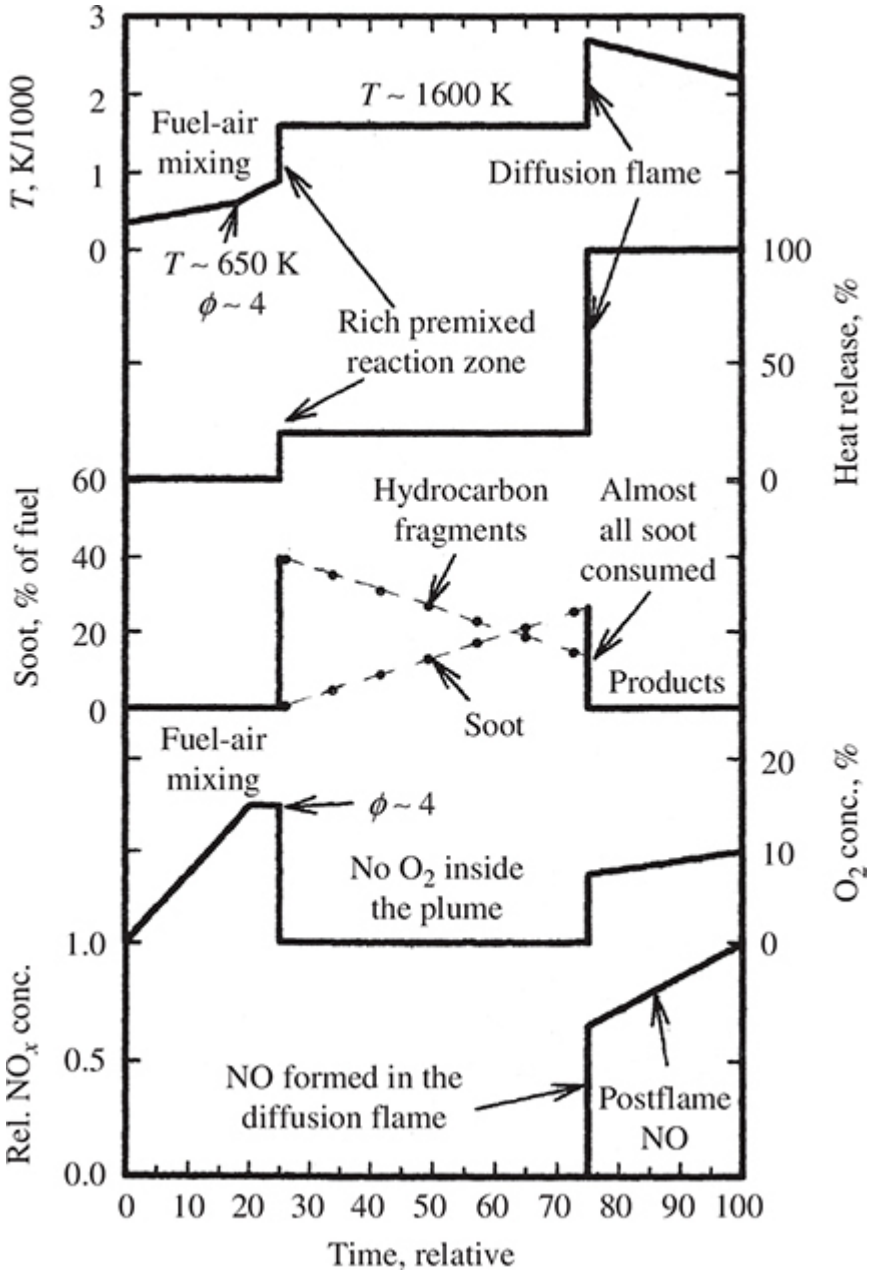


Figure 11.45 The sequence of chemical reactions that occur during fuel spray development, combustion, fuel reaction, and soot formation and

oxidation, in a DI diesel engine fuel spray.[71]

Figure 11.45 illustrates the chemical aspects of this sequence of processes occurring within each fuel spray. The premixed-reaction-zone fuel chemical-energy release (some 20% of the fuel's total energy) raises the gas temperature within the spray to about 1600 K as the fuel molecules fragment and (with many intermediate steps; see Sec. 11.5.4), form PAHs, followed by soot formation, in the fuel-rich within-the-spray mixture. This "rich-mixture" fuel (with its high soot content) then moves into the diffusion flame now surrounding each spray, is heated, mixes with oxygen, and releases almost all its as yet unreleased chemical energy. As this mixing-controlled diffusion-flame combustion process moves to completion, almost all the soot particles are burned up. As noted in Sec. 11.2.4 , this is also the primary NO formation region.

11.5.4 Soot Formation Fundamentals

The soot particles, whose characteristics have been described in the above two sections, form primarily from the carbon in the diesel fuel. The formation process starts with a fuel molecule containing 12 to 22 carbon atoms with an H/C ratio of about 2, and ends up with particles typically a hundred nanometers or so in diameter, composed of spherules 10 to 25 nm in diameter each containing some 10^5 carbon atoms, and having an H/C ratio of about 0.1. Much of the information available on the fundamentals of soot formation comes from studies in simple premixed and diffusion flames, stirred reactors, shock tubes, and constant-volume combustion vessels. Also, the production of carbon black requires a high yield of soot from pyrolysis of a hydrocarbon feedstock, and the literature from that field has much to contribute. However, the characteristics of diesel combustion which make it different from these more basic studies—the high gas temperatures and pressures, complex fuel composition, importance of turbulent mixing, the unsteady nature of the process, and the three-dimensional geometry of fuel sprays—make it challenging to interpret fundamental ideas regarding soot formation in the diesel context. There is much about the soot formation process in diesel engines that is incompletely understood.

Soot formation takes place within the combusting fuel spray at temperatures of about 1600 K, pressures of 100 bar, and with much less air locally than is needed to fully burn the fuel. The time available for the formation of soot particles from the fuel hydrocarbons is of the order of milliseconds. The resulting aerosol—dispersed solid-phase particles in a gas—can be characterized by the total amount of condensed phase (often expressed as the soot volume fraction, F_v, the volume of soot/total volume), the number of soot particles per unit volume (N) , and the size of the particles (e.g., average diameter d) . F_V, N, and d are mutually dependent [e.g., for spherical particles $F_V = (\pi/6)Nd^3$], and any two of these variables characterize the system. It is most convenient to consider N and F_V as the independent variables since they each relate to the "almost-separate" stages of soot particle generation (the source of N) and soot particle growth (the source of F_V) .

The formation process can be summarized as follows: [72]

1. Particle formation, where the first condensed-phase material arises from the fuel molecules via their partial oxidation and/or pyrolysis products. These products typically include various unsaturated hydrocarbons, particularly acetylene and its higher analogues ($C_{2n}H_2$), and polycyclic aromatic hydrocarbons (PAH). These two types of molecules are considered the most dominant precursors of soot in flames. The condensation reactions of gas-phase species such as these lead to the appearance of the first recognizable soot particles (often called nuclei). These first particles are very small ($d < 2$ nm) and the formation of large numbers of them involve minimal soot loading in the region of their formation.
2. Particle growth, which includes both surface growth, coagulation, and aggregation. Surface growth, by which the bulk of the solid-phase material is generated, involves the attachment of gas-phase species to the surface of particles and their incorporation into the particulate phase. Figure 11.46 , where the log of the molecular weight of a species is plotted against its hydrogen mole fraction $\tilde{x}_H$, illustrates some important points about this process. Starting with a fuel molecule where $\tilde{x}_H \geq 0.5$, it is apparent that neither polyacetylene

chain growth nor PAH growth would lead to soot particles which have $\tilde{x}_H$ in the range 0.1 to 0.2. What is required is condensation of species with the right hydrogen content, or condensation of species with higher hydrogen content followed by dehydrogenation, or a combination of both these processes. Obviously some polyacetylenes and some PAH can satisfy these requirements, as can saturated platelets (e.g., $C_{27}H_{27}$; see Sec. 11.5.2). Surface growth reactions lead to an increase in the amount of soot (F_V) but the number of particles (N) remains essentially unchanged. The opposite is true for growth by coagulation, where the particles collide and coalesce, which decreases N with F_V constant. Once surface growth stops, continued aggregation of particles into chains and clusters can occur.

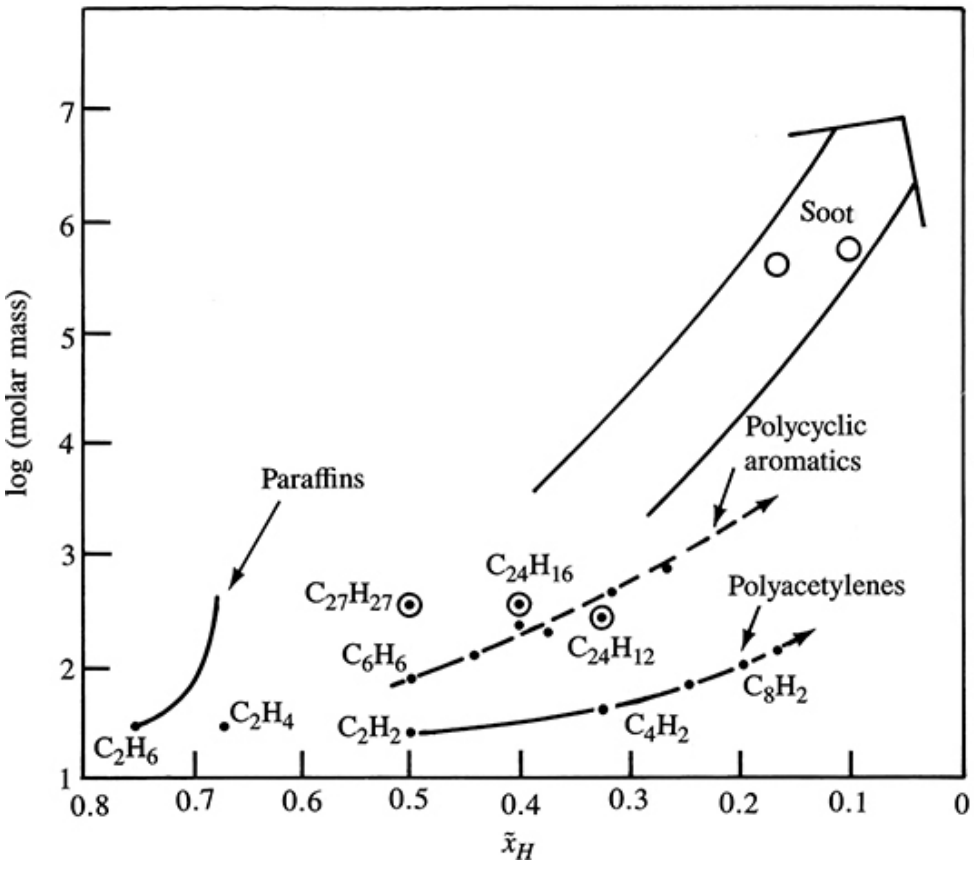


Figure 11.46 Paths to soot formation on plot of species molecular weight M versus hydrogen mole fraction $\tilde{x}_H$.[72]

These stages of particle generation and growth constitute the soot formation process. At each stage in the process oxidation can occur, where soot or soot precursors are burned in the presence of oxidizing species to form gaseous products such as CO and CO_2. The eventual emission of soot from the engine will depend on the balance between these processes of formation and burnout. The emitted soot is then subject to a further mass addition process as the exhaust gases cool and are diluted with air. Adsorption into the soot particle surface and condensation to form new particles of hydrocarbon species in the exhaust gases occurs in the exhaust system and in the dilution tunnel which simulates what happens in the atmosphere. Figure 11.47 illustrates the relationship between these processes.[67] Although they are illustrated as discrete processes, there is overlap, and they may occur concurrently in a given elemental mixture region within the diesel combustion chamber. Of course, also due to the non-homogeneous nature of the mixture and the duration of fuel injection and its overlap with combustion, at any given time different processes are in progress in different regions or packets of fluid. The fundamentals of each of these processes will now be reviewed.

Soot Particle Formation

Empirically, it has been found useful to define the composition of the fuel-oxidizer mixture at the onset of soot formation in flames by the carbon/oxygen ratio. Equilibrium considerations indicate that soot (C_S) should form when, in the overall reaction

$$C_mH_n + yO_2 \rightarrow 2yCO + \frac{n}{2}H_2 + (m-2y)C_S \qquad (11.35)$$

m becomes larger than $2y$: i.e., when the C/O ratio exceeds unity. The corresponding fuel/air equivalence ratio is given by

$$\phi = 2\left(\frac{C}{O}\right)(1+\delta) \qquad (11.36)$$

where $\delta = n/(4\,m)$: ϕ is 3 for (C/O) = 1, with $n/m = 2$. The experimentally observed critical C/O ratios are less than unity, however, varying with fuel composition and details of the experimental setup from about 0.5 to 0.8. The

critical C/O ratio for soot formation increases with increasing temperature but is only weakly dependent on pressure. Beyond the carbon formation limit, the yield of soot increases rapidly with increasing C/O ratio and is strongly enhanced by increasing pressure. [72]

It is obvious that soot formation is a nonequilibrium process. Several different theories have been advanced to explain the pyrolysis process—the extensive decomposition and atomic rearrangement of the fuel molecules that culminates in nucleation that occurs in very fuel-rich mixtures. Often-cited mechanisms are thermal cracking that results in fragmentation of fuel molecules into smaller ones, condensation reactions and polymerization that result in larger molecules, and dehydrogenation that lowers the H/C ratio of the hydrocarbons destined to become soot. Three different paths to the production of soot appear to exist, depending on the formation temperature. At the lowest temperatures (≤1700 K), only aromatics or highly unsaturated aliphatic compounds of high molecular weight form substantial solid carbon through pyrolysis. At intermediate temperatures (>1800 K), all normally used hydrocarbon fuels produce soot if burned at sufficiently rich stoichiometry, but appear to do so by following a different path.

A simple mechanistic model for nucleation in the low and intermediate temperature ranges which has considerable experimental support is illustrated in Fig. 11.48. [67] At low temperatures, an aromatic hydrocarbon can produce soot via a relatively fast direct route that involves condensation of the aromatic rings into a graphite-like structure. Above about 1800 K, however, a slower, less-direct route is favored that entails ring breakup into smaller hydrocarbon fragments. These fragments then polymerize to form larger unsaturated molecules that ultimately produce soot nuclei. Aliphatic molecules can only follow this latter less-direct route. Experimental measurements in flames suggest that polyunsaturated hydrocarbon compounds are involved in nucleation, and acetylenes and polyacetylenes have been detected that decrease in concentration as the mass of carbon formed increases. Such observations fit the indirect path in Fig. 11.48.

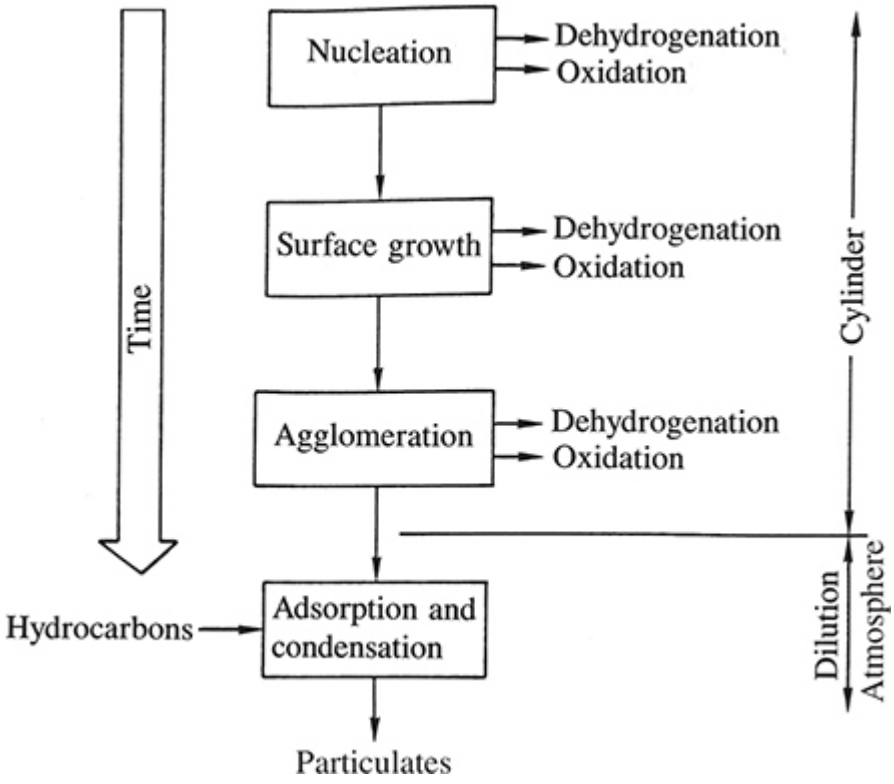


Figure 11.47 Processes leading to net production of diesel particulates.[67]

Soot Particle Growth

Nucleation produces a large number of very small particles with a modest soot loading. The bulk of the solid-phase material is generated by surface growth, which involves the gas-phase deposition of hydrocarbon intermediates on the surfaces of the spherules that develop from the nuclei. A qualitative description of the changes that occur as a function of time in a rich premixed flame during nucleation and surface growth is shown in Fig. 11.49. The soot fraction, F_V, in units of soot volume per unit volume of gas, is related to the number density N and the volume-mean diameter of the soot particles by

$$F_V = \frac{\pi}{6} N d^3 \tag{11.37}$$

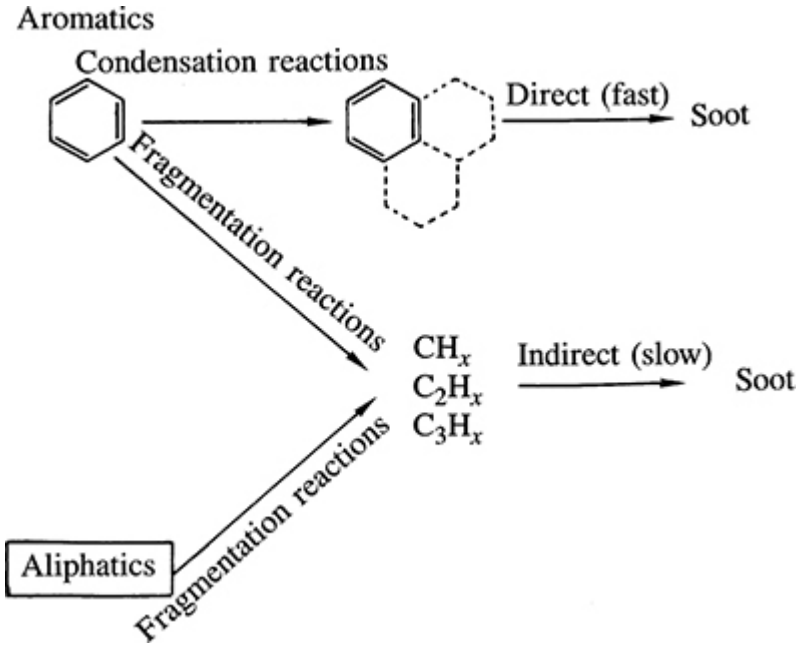


Figure 11.48 Mechanistic model for formation of soot from aromatic and aliphatic hydrocarbon compounds.[67]

d is the actual diameter of the spherules, or the diameter of a sphere of equivalent volume to an agglomerated particle. The rate of change of particle number density with time t can be written

$$\frac{dN}{dt} = \dot{N}_n - \dot{N}_a \tag{11.38}$$

where $\dot{N}_n$ is the rate at which fresh nuclei appear and $\dot{N}_a$ is the rate of agglomeration of spherules or particles that collide and stick. At the peak of the N curve, $\dot{N}_n = \dot{N}_a$. To the left of the peak, $\dot{N}_a > \dot{N}_n$ the particle diameter remains essentially constant at the minimum detectable diameter and the (small) rise in soot volume is dominated by nucleation. To the right of the peak in the N curve, $\dot{N}_a > \dot{N}_n$. The number of agglomerating collisions is high because of the high number density; at the same time nucleation ends because there is enough dispersed surface area for gaseous deposition of hydrocarbon intermediates so the probability of generating new nuclei falls to zero. With nucleation halted slightly to the right of the N-curve peak, all the subsequent increase in soot volume fraction (the majority) stems from surface growth. To the right of the N curve peak, in the case illustrated, the number density falls

by three orders of magnitude. This is the result of agglomeration, which is responsible for a portion of the increase in particle diameter. Agglomeration does not contribute to the rise in soot volume fraction, F_v. Surface growth that takes place on nuclei and on spherules is responsible for forming the concentric shells (somewhat distorted and warped) that constitute the outer portions of spherules and which are distinct from the less-organized spherule center (see Figs. 11.42 and 11.43). Surface growth on agglomerated particles may partly fill in the crevices at the junctures of adjoining spherules to provide the nodular structure evident in Fig. 11.42. [67]

Once particles have formed, inter-particle collisions can lead to agglomeration, thereby decreasing the number of particles and increasing their size. Three types of agglomeration have been identified in soot formation. During the early stages of particle growth, collision of two spherical particles may result in their *coagulation* into a single spheroid. This is easy to visualize in hydrocarbon pyrolysis where the beginnings of a soot particle may have the viscosity of a tarry liquid. Also, when the individual particles are small, rapid surface growth will quickly restore the original spherical shape. This process occurs up to diameters of about 10 nm. On the other hand, if spherules have solidified before collision and surface growth rates have diminished, the resulting particles resemble a cluster in which the original spherules retain much of their individual identity. After surface growth essentially ceases, continued coalescence of the soot particles results in the formation of chainlike structures of discrete spherules. This suggests electrostatic forces are significant. Positive charge measured on these particle chains is claimed to be the cause of their chainlike structure. [67, 69] This latter coalescence once surface growth ceases is termed *aggregation*.

It has been shown experimentally that during coagulation, the rate of decrease of particle number density was proportional to the product of a coagulation coefficient and the square of the number density:

$$-\frac{dN}{dt} = KN^2 \tag{11.39}$$

This is the Smoluchowski equation for coagulation of a liquid colloid. Based on Brownian motion, this equation is applicable when the Knudsen number (ratio of mean free path to particle diameter) exceeds 10. K depends on such

factors as particle size and shape, size distribution, and the temperature, pressure, and density of the gas. Equation (11.39) has been used to predict coagulation rates in low-pressure sooting flames.[69,72] These studies show that under conditions approximating those in engine flames, the fraction of the initial number density N_0 remaining at time t is given approximately by

$$\frac{N}{N_0} \approx (KN_0t)^{-1} \tag{11.40}$$

Thus as t increases, N/N_0 decreases rapidly. Although these coagulation calculations are simplistic (in that many of the assumptions made are not strictly valid since soot particles are not initially distributed homogeneously within the combusting fuel-rich sprays, they are not monodisperse, and surface growth and oxidation may be taking place during agglomeration), an overall conclusion is that the rate of coagulation of spherules and particles to larger particles is very sensitive to number density. Thus the number of particles decreases rapidly with advancing crank angle in the diesel engine during the early part of the expansion process and agglomeration is essentially complete well before the exhaust valve opens.

Throughout the soot formation process, the H/C ratio of the hydrocarbons formed in the pyrolysis and nucleation process, and of the soot particles, continually decreases. The H/C ratio decreases from a value of about 2, typical of engine fuels, to of order 1 in the youngest soot particles that can be sampled, and then to 0.2 to 0.3 once surface growth has ceased in the fully agglomerated soot. [72] The latter stages of this process are indicated in Fig. 11.49. The addition of mass to the soot particles occurs by reaction with gas-phase molecules. The reacting gas-phase hydrocarbons appear to be principally acetylenes, with larger polymers adding faster than the smaller. Small polyacetylenes undergo further polymerization in the gas phase, presumably by the same mechanism leading to nucleation. As a result of preferential addition of the larger polymers, the H/C ratio of the particles decreases toward its steady-state value. Thus most of the polyacetylenes added must be of very high molecular weight, or dehydrogenation must also take place. [69, 72]

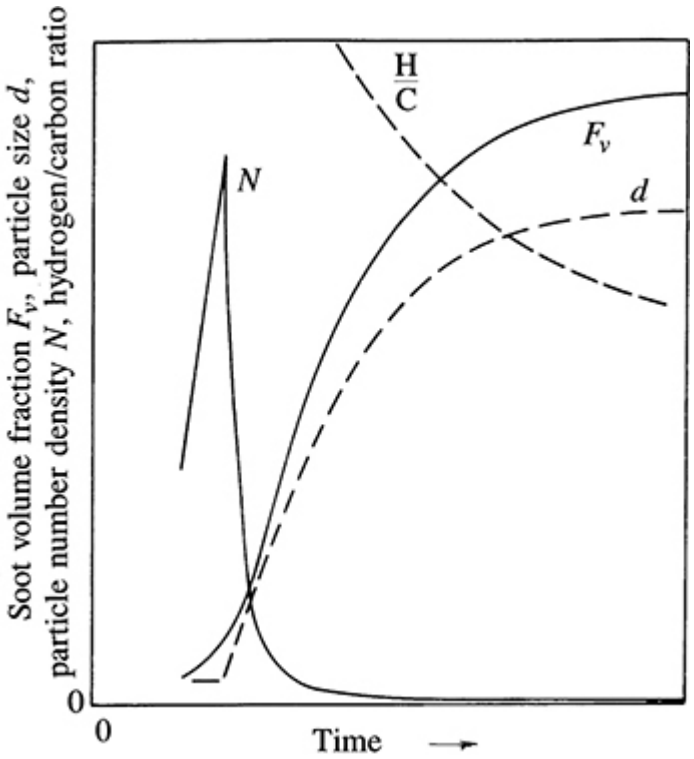


Figure 11.49 Variation in soot volume fraction F_V, particle number density N, particle size d, and soot hydrogen/carbon ratio with time in a flame.[67]

11.5.5 Soot Oxidation

In the overall soot formation process, shown schematically in Fig. 11.47, oxidation of soot at the precursor, nuclei, and particle stages can occur. The engine in-cylinder soot-concentration data reviewed in Sec. 11.5.3 indicate that almost all the soot formed within the fuel sprays is oxidized before the exhaust process commences. In the discussion of diesel combustion movies in Sec. 10.3.1 , dark (brown) regions were observed in the original color photographs (see Fig. 10.4); these were interpreted as soot particle clouds, and were seen to be surrounded by a diffusion flame which appeared white from the luminosity of the high-temperature soot particles consumed in this flame. As air mixed with this soot-rich region, the diffusion flame eradicated the dark soot clouds as the particles were burned up.

In general, the rate of heterogeneous reactions such as the oxidation of soot depends on the diffusion of reactants to and products from the diffusion flame region as well as the kinetics of the oxidation reactions. For particles

less than 1 μm diameter, diffusional resistance is minimal. The soot oxidation process in the diesel cylinder is kinetically controlled, therefore, since particle sizes are smaller than this limit. There are many species in or near the diffusion flame that could oxidize soot: examples are O_2, O, OH, CO_2, and H_2O. Earlier reviews of soot oxidation in flames postulated that at high oxygen partial pressures, soot oxidation could be correlated with a semi-empirical formula based on close-to-stoichiometric combustion products; however, oxidation by OH radicals (which are plentiful within the flame) may also be important under high temperature, high pressure, stoichiometric and slightly-lean within-the-flame conditions. Figure 11.50 shows the carbon particle surface recession rate that this semi-empirical formula predicts as a function of temperature and oxygen partial pressure.[67]

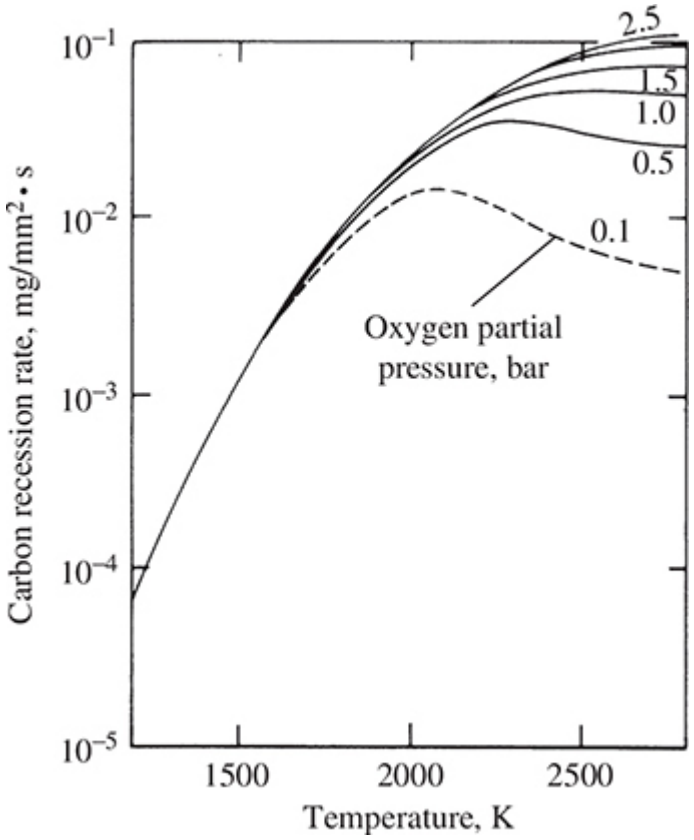


Figure 11.50 Specific soot oxidation rate predictions as a function of temperature and oxygen partial pressure, from the Nagle and Strickland-Constable, O_2 based, correlation.[67]

It has been argued on the basis of structural similarities that the rates of oxidation of soot and graphite should be similar. While this is a significant simplification, the semi-empirical formula of Nagle and Strickland-Constable has been shown [73] to correlate pyrolytic-graphite oxidation for oxygen partial pressures $p_{02} < 1$ atm and temperatures between 1100 K and 2500 K. This formula is based on the concept that there are two types of sites on the carbon surface available for O_2 attack. For the more reactive type A sites, the oxidation rate is controlled by the fraction of sites not covered by surface oxides (and therefore is of mixed order, between 0 and 1 in p_{O_2}). Type B sites are less reactive, and react at a rate which is first order in p_{O_2}. A thermal rearrangement of A sites into B sites is also allowed (with rate constant k_T. A steady-state analysis of this mechanism gives a surface mass oxidation rate w (gC/cm$^2 \cdot$ s):

$$\frac{w}{12} = \left(\frac{k_A p_{O_2}}{1 + k_Z p_{O_2}} \right) x + k_B p_{O_2} (1 - x) \tag{11.41}$$

where x is the fraction of the surface occupied by type A sites and is given by

$$x = \left(1 + \frac{k_T}{p_{O_2}} \right)^{-1} \tag{11.42}$$

The empirical rate constants determined by Nagle and Strickland-Constable for this model are listed in Table 11.13. According to this mechanism, the reaction is first order at low oxygen partial pressures, but approaches zero order at higher pressures. At a given oxygen pressure, the rate initially increases exponentially with temperature (equivalent activation energy is k_A/k_Z or 34,100 cal/mol). Beyond a certain temperature the rate decreases as the thermal rearrangement favors formation of less reactive B sites. When, at sufficiently high temperature, the surface is completely covered with B sites, the rate is first order in oxygen partial pressure and increases again with temperature. [72] Park and Appleton [73] have compared this formula with oxidation rate data obtained from pyrolytic graphite samples, carbon black particles, and with the available flame soot oxidation data. The formula

correlates the data shown to within a factor of 2. Under diesel engine conditions, the O_2 partial pressure can be high (~several atmospheres), as can the temperature of close-to-stoichiometric burned gases (≤2800 K). Figure 11.50 shows the carbon particle surface recession rate that this formula predicts as a function of temperature and oxygen partial pressure.[67]

TABLE 11.13 Rate constants for Nagle and Strickland-Constable soot oxidation mechanism [74]

Rate constant	Units
$k_A = 20 \exp(-15{,}100/T)$	$g/cm^2 \cdot s \cdot atm$
$k_B = 4.46 \times 10^{-3} \exp(-7640/T)$	$g/cm^2 \cdot s \cdot atm$
$k_T = 1.51 \times 10^5 \exp(-48{,}800/T)$	$g/cm^2 \cdot s$
$k_Z = 21.3 \exp(2060/T)$	atm^{-1}

This Nagle and Strickland-Constable relationship has been used to estimate the amount of soot that can be oxidized in a typical diesel engine. It was assumed that soot was present in stoichiometric combustion products within the diffusion flames, at selected times in the cycle, and that mixing with air leaned out the burned gas mixture at different rates until the overall fuel/air equivalence ratio was reached. The surface recession rate during this process was computed. Figure 11.51 shows sample results at an engine speed of 1600 rev/min and an overall cylinder equivalence ratio of 0.58. Fast, intermediate, and slow mixing occurred in 30, 70, and 140 crank-angle degrees, respectively. The surface recession rate rises to a maximum as p_{o_2} increases, and then decreases as the falling gas temperature more than offsets the increasing oxygen concentration. While the shape of the recession rate versus time curves depends on the mixing rate, the total amount of carbon burned (the area under each curve in Fig. 11.51(*b*) is about the same (0.1 $\mu g/mm^2$). However, the point in the cycle at which the soot-containing burned gas mixture passes through stoichiometric is much more important. For the late-mixing element shown (mixing lean of stoichiometric at 40° ATC), the total carbon mass oxidized is only 40% of that for the early mixing

calculation. This is primarily due to the decreasing gas temperatures as the expansion stroke proceeds, and not the longer time available for burnup. [74]

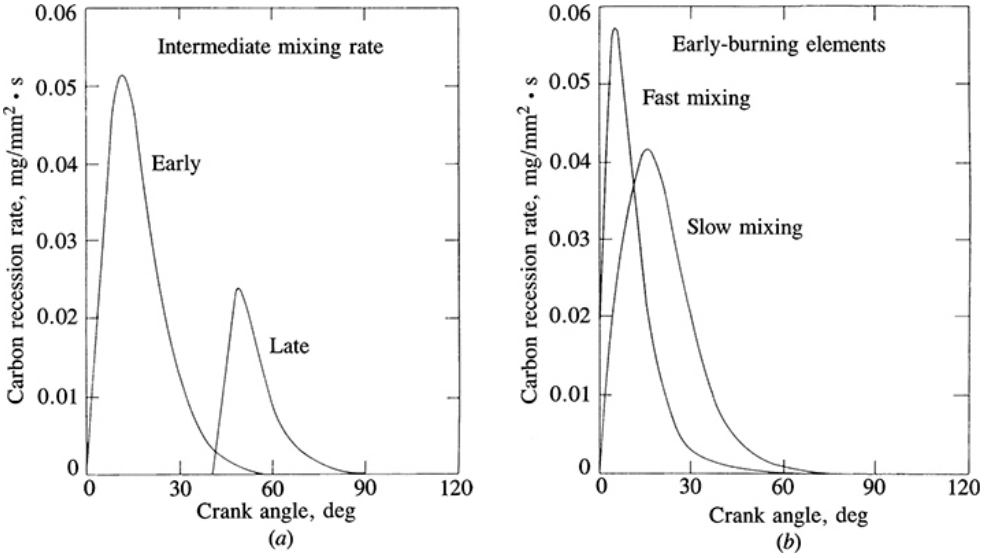


Figure 11.51 Soot particle burnup rate in diesel combustion environment: (*a*) in early- and late-burned fuel-air elements with intermediate mixing rate; (*b*) for fast and slow mixing for early-burning elements.[67]

For a spherical soot particle, the mass burning rate w (g/cm^2 · s) can be converted to a surface recession rate using

$$\frac{dr}{dt} = \frac{-w}{\rho}$$

where ρ is the density (≈2 g/cm^3). The integrated values of $w(t)$ when divided by ρ then give the maximum radius of a soot particle that can be burned up. Integrated values of 0.1 μg/mm^2 (estimated for TC start of burn up) correspond to a radius of about 50 nm or diameter of 100 nm. Individual spherule diameters are less than about 30 nm, so soot which mixes with oxygen containing burned gases early in the expansion stroke is likely to be fully burned. Thus the soot present in the exhaust would be expected to come from regions which mix with oxygen too late for the oxidation rate to be sufficient for complete particle burn-up.

Agglomeration will have an indirect influence on the amount of soot oxidized through its effect on surface area. In the limiting case of a spherical cluster, n monodisperse spherules ($10 \leq n \leq 100$) can be imagined as compacted into a single solid sphere of equal volume. Alternatively, the same n spherules can be imagined compacted into a cylinder of diameter equal to that of the original spherules. Since oxidative attack is essentially an exterior surface phenomenon, the surface/volume ratio is the appropriate measure of the effect of particle shape on soot mass burn-up rate. It can be shown that the surface/volume ratios for the single sphere, single cylinder, and individual spherule, are in the ratio $n^{-1/3}$, $\frac{2}{3}$, and 1, respectively. Thus agglomeration will decrease the relative oxidation rate. In the limit, spherical clusters are less desirable than a chain; the larger the cluster the bigger the relative reduction in surface area. However, the densely packed spherule limit does not appear to be approached in practice. A specific surface area, of about 200 m^2/g for diesel soot, has been measured. [75] A smooth-surfaced 25-nm diameter spherule with a 2-g/cm^3 density has a surface/mass ratio of 150 m^2/g: the measured value is about one-third higher value, indicating low porosity and an agglomerate structure which is loosely rather than densely packed. [74]

The research literature on soot oxidation (e.g. Refs. 76 and 77) suggest an alternative mechanism: oxidation by OH radical attack. In an atmospheric-pressure flame environment with rich and close-to-stoichiometric combustion products where O_2 mole fractions are low, oxidation by OH radical attack is more significant than oxidation by O or O_2. Thus, OH radicals could be important in oxidizing soot in the diesel's diffusion flame under close-to-stoichiometric conditions. The maximum soot-particle surface-recession rates in the diesel combustion context occur in the diffusion flame that surrounds each fuel spray. The soot particles formed within each spray (where the local fuel-air equivalence ratio is significantly rich of stoichiometric, $\phi \approx$ 2-4) enter the diffusion flame from the "fuel side"; oxygen enters from the air surrounding each spray. The soot particles are heated to close to the stoichiometric flame temperature (≈2700 K) and burn. Maximum oxidation rates most likely occur at an equivalence ratio of 0.9 (10% lean), where there is excess oxygen, as the combustion products containing these high-temperature particles diffuse outward from the close-to-stoichiometric diffusion-flame reaction zone and mix with air. As this

outward diffusion and mixing with ever more air continues, particles that have not burned up will cool so soot oxidation slows down significantly. These particles become a key component in the engine-out particulate emissions.

11.5.6 Adsorption and Condensation

The final process in the particulate formation sequence illustrated in Fig. 11.47 is adsorption and condensation of hydrocarbons. This occurs after the cylinder gases have been exhausted from the engine, as these exhaust gases are diluted and cooled by mixing with air. In the standard particulate mass emission measurement process this occurs in a dilution tunnel which simulates approximately the actual atmospheric dilution process. A diluted exhaust gas sample is then filtered to remove the particulate. After equilibrating the collection filter at controlled conditions to remove water vapor, the particulate mass is obtained by weighing. In the prescribed EPA procedure, the filter temperature must not exceed 52°C. For a given exhaust gas temperature, the filter (and sample) temperature depends on the dilution ratio, as shown in Fig. 11.52.

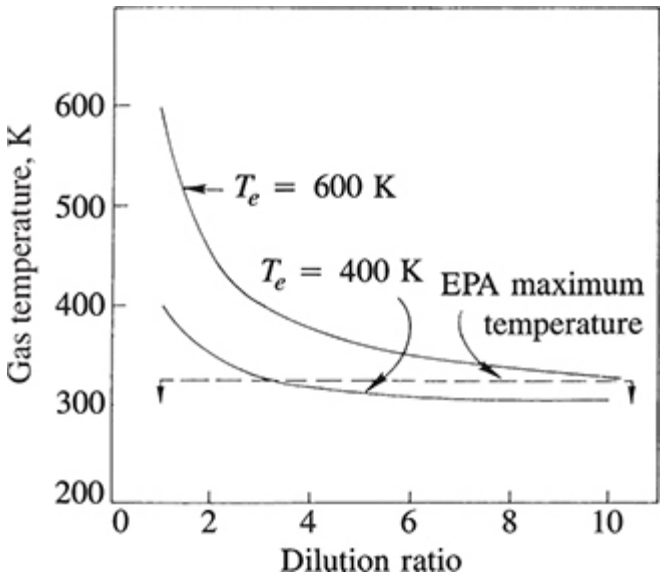


Figure 11.52 Effect of exhaust gas dilution ratio on the temperature of the collected particulate sample as a function of engine exhaust temperature T_e.[67]

The effect of the dilution ratio (and the dependent sample temperature) on collected particulate mass, for a standard dilution tunnel where the total sample is partitioned into extractable and nonextractable fractions, is as follows. The nonextractable fraction, the carbonaceous soot generated during combustion, is not affected by the dilution process. With no dilution (a dilution ratio of unity) the difference between the total and nonextractable mass is small; the bulk of the extractable fraction is acquired after the exhaust gas is mixed with dilution air. Extensive studies of this dilution process have shown that both adsorption and condensation occur. Adsorption involves the adherence of molecules of unburned hydrocarbons to the surfaces of the soot particles by chemical or physical (van der Waals) forces. This depends on the fraction of the available particle surface area occupied by hydrocarbons and on the partial pressure of the gaseous hydrocarbons that drives the adsorption process. As the dilution ratio increases above unity, the effect of decreasing temperature on the number of active absorption sites dominates and the extractable fraction increases. At high dilution ratios, the sample temperature becomes insensitive to the dilution ratio (see Fig. 11.52) but the decreasing hydrocarbon partial pressure causes the extractable mass to fall further. At its maximum, the extractable fraction is about one-third of the total particulate. Condensation will occur whenever the vapor pressure of the gaseous hydrocarbon exceeds its saturated vapor pressure. Increasing dilution decreases hydrocarbon concentrations and hence vapor-pressure. However, the associated reduction in temperature does reduce the saturation pressure. High exhaust concentrations of hydrocarbons are the conditions where condensation is likely to be most significant, and the hydrocarbons most likely to condense are those of low volatility. Sources of low-volatility hydrocarbons are the high-boiling-point end of the fuel, unburned hydrocarbons that have been pyrolyzed but not consumed in the combustion process, and the lubricating oil. [67]

Experiments with a passenger car diesel where the oil was tagged with a radioactive tracer have shown that the oil can contribute between 2 and 25% of the total particulate mass, with the greatest contribution occurring at high speed. About half of the extractable mass was traceable to the oil. All the material traceable to the oil was found in the extractable fraction, indicating that the oil did not participate in the combustion process. However, the oil is not always a significant contributor: in other diesel engines, fuel may be the dominant source of extractable material. [67]

11.6 EXHAUST GAS TREATMENT

11.6.1 Available Options

Our discussion so far has focused on *engine-out* emissions. Substantial further reductions in emissions can be obtained by removing pollutants from the exhaust gases in the engine exhaust system. Effective devices developed to achieve this result include catalytic converters (oxidizing catalysts for HC and CO, reducing catalysts for NO_x, and three-way catalysts for all three pollutants), and traps or filters for particulates. The stringency of current and anticipated vehicle air-pollutant regulations requires major reductions in engine-out emissions, so effective aftertreatment catalyst and filter technologies in the exhaust system are now essential.

The temperature of exhaust gas in a spark-ignition engine can vary from 300 to 400°C during idle to about 900°C at high-power operation. The most common range is 400 to 600°C. Spark-ignition engines operate most of the time at the stoichiometric air/fuel ratio to enable the three-way catalyst technology. Historically, such engines have operated fuel-rich at high torque levels to prevent knock onset and reduce exhaust gas temperatures, but that option is being used much less since it significantly increases HC and CO emissions. Lean operation of SI engines improves the engine's efficiency at part-load, but achieving stable combustion and tight emission's control are challenges. The exhaust gas may therefore contain modest amounts of oxygen (when slightly lean) or more substantial amounts of CO (when rich). In contrast, diesel engines, where load is controlled by the amount of fuel injected, always operate well lean of stoichiometric. The diesel's exhaust gas therefore contains substantial oxygen and is at a lower temperature (200 to 500°C). Removal of gaseous pollutants from the exhaust gases after they leave the engine cylinder can be either thermal or catalytic. In order to oxidize the hydrocarbons in the gas phase without a catalyst, a residence time of order 50 ms and exhaust temperatures in excess of 600°C are required. To oxidize CO, temperatures in excess of 700°C are required. Temperatures high enough for homogeneous thermal oxidation can be obtained by spark retard (with some loss in efficiency) and minimizing heat losses in the exhaust ports and manifold. The residence time can be increased by

increasing the exhaust manifold volume to form a *thermal reactor*: however, this approach has not proved successful.

The task of catalytic aftertreatment devices is to oxidize the CO and HC in the exhaust gas to CO_2 and H_2O at temperature levels the exhaust gases can readily provide. With NO_x (predominantly nitric oxide, NO) the task is reduction to N_2, again at or below the available exhaust gas temperature. The temperatures required for gas- phase chemistry to effect these changes, since the *activation energy* (or energy barrier) for breaking these air pollutant molecule bonds is above the thermal energy available at average exhaust gas temperatures. So only modest pollutant reduction in the exhaust system occurs. The catalyst (e.g., a collection of noble metal atoms on a high surface-area substrate) facilitates the desired chemistry by reducing this energy barrier so reactions can occur at significantly lower (and attainable) temperatures. For CO and HC oxidation on the catalyst surface, sufficient oxygen must be available. With a stoichiometric mixture, the air (oxygen) associated with the CO and unburned HC is (on average) available. For NO reduction in spark-ignition engines, reducing agents in the exhaust gas (CO, H_2, and HC) are present. For diesels, with their lean operation, reducing agents such as ammonia NH_3 are added to effect this reduction. Two added features often embodied are that catalysts can be formulated to target and enhance specific (selective) chemical reaction rates. Also, storage capacity for a specific pollutant can be built into the catalyst, with release and removal performed periodically, triggered by changing the exhaust gas composition: for example, in lean NO_x storage-reduction catalysts for SI engines where the engine-out NO stored when the engine operates lean, is released and reduced when the engine is switched for a short period to fuel-rich operation. Catalytic oxidation of CO and hydrocarbons and reduction of NO_x in the exhaust can be achieved at temperatures as low as 250 to 300°C. Thus effective removal of these pollutants occurs over a much wider range of temperatures than can be achieved with gas-phase thermal oxidation of HC and CO.

In spark-ignition engine exhausts, catalysts for CO, HC, and NO removal simultaneously are widely used, and are highly effective. However, these *three-way catalysts* are only effective if the exhaust gas is stoichiometric. Precisely at this condition, there is oxygen available for oxidation and reducing agents (CO, H_2, and HC) are available for NO reduction. In

diesels, which operate lean, NO_x and particulates are the primary pollutants emitted. Diesel oxidation catalysts are frequently used to oxidize hydrocarbons since, if not removed, higher molecular weight HC can be absorbed later, onto soot particles. Selective catalytic reduction (SCR) of NO_x (in an otherwise oxidizing environment) is effected using NH_3 as the primary reducing agent which is added in the form of urea to the exhaust gases. Also, NO_x storage catalysts for lean operating engines are available where NO is converted to NO_2, and stored as a nitrate. When the storage capacity is full, the engine is operated rich for a short period, the nitrate is decomposed, released as NO_2, and reduced to N_2 by the HC, CO, and H_2 in the rich exhaust.

For effective operation, these exhaust catalysts must operate at an elevated temperature (a few hundred °C) and with an exhaust gas flow rate (which defines the residence time) and composition matched to the catalyst's size and chemical conversion task. Thus insertion of catalysts into the engine's exhaust requires that the catalyst and engine be designed as two interacting systems (often with sensors and feedback loops). The warm up of the catalyst following a cold engine start is one key area. The catalyst surface must reach a temperature of about 300°C to become effective, and the exhaust gas flow and temperature are the available sources of thermal energy. Figure 11.53 illustrates the warm up of an oxidation catalyst and its evolving impact on engine-out HC emissions. During the starting and warm up of a spark-ignition engine, the engine-out HC emissions are higher than the warmed-up engine HC levels. But the catalyst takes time to warm up. It takes time for the catalyst to *light off*: that is, attain a temperature at which is catalytic efficiency is defined as the percentage of a given pollutant removed by the catalyst). Thus the HCs exhausted from the exhaust system are initially the cold engine-out levels: only after catalyst light off do the engine-out levels drop rapidly to a warmed-up engine and catalyst (usually a reduction of over 90%).

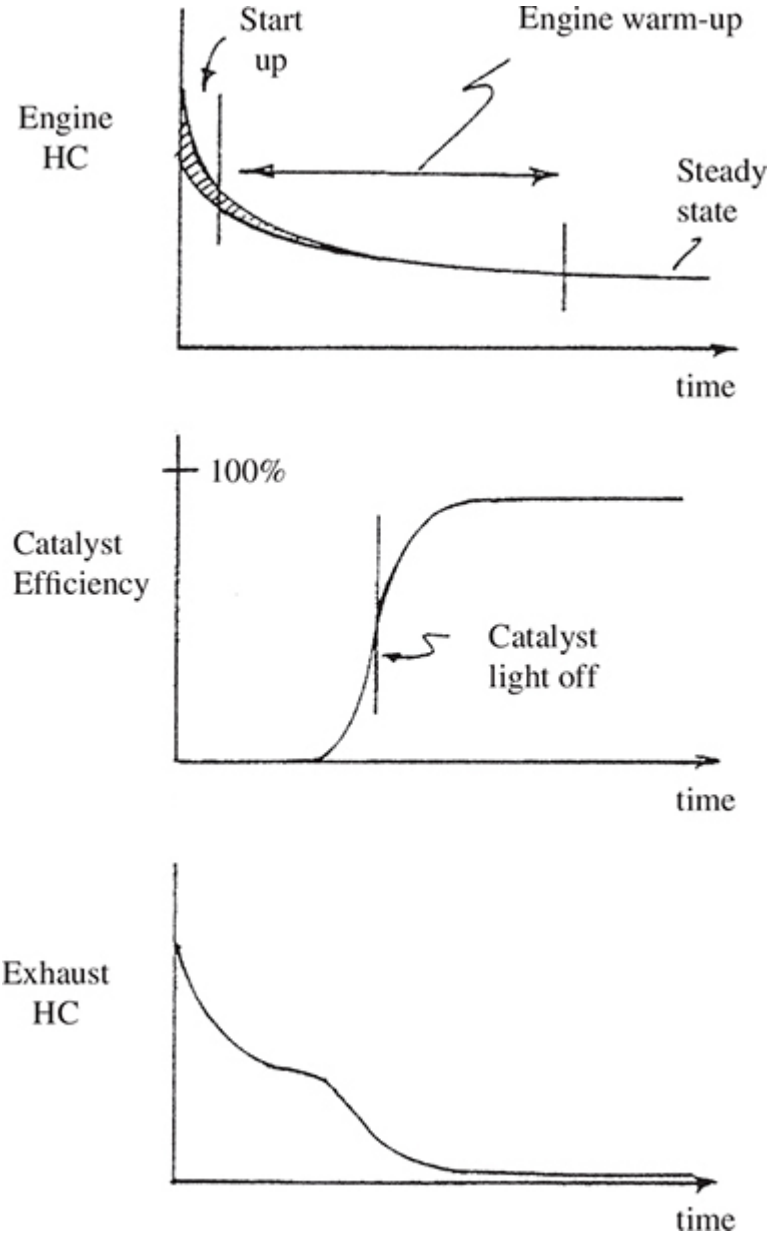


Figure 11.53 Illustration of the warm-up behavior of a spark-ignition engine/catalyst system. Catalyst light off takes some 10 to 30s, depending on conditions.

Particulates in the exhaust gas stream can be removed by a trap or filter. Due to the small particle sizes involved, some type of filter is the most effective trapping method. Porous ceramic-monolith catalyst-support technology has proved a good approach to exhaust gas filtration. Half of the wall-flow monolith channels are plugged at one end, the other half are

plugged at the other end. When used as a particulate trap, the porosity of the monolith channel walls has higher and carefully controlled porosity. The channel or cell density (about 200 cells/in ^{2}, or 30 per cm ^{2}) is a factor of 2 to 4 lower than that of catalyst substrates. A filter wall-flow monolith for a diesel automobile would be about 150 mm in diameter, 250 mm in length, and thus have about 5000 flow channels (into only half of which the exhaust gas flows). Such a particulate filter is usually preceded by a diesel oxidation catalyst (DOC), which removes the hydrocarbons in the exhaust gas prior to where they might condense or be absorbed onto the soot particles. Mass accumulates within the trap, and the exhaust manifold pressure thus increases during trap operation. The particulates, once trapped and accumulated, are burned up either by initiating oxidation within the trap by raising the trap temperature by adding to the exhaust's thermal energy by burning extra fuel in the cylinder or in the manifold. Or the trap may contain catalytically active material to initiate burn up of the particulate matter at lower temperature. The operation of particulate traps is reviewed in Sec. 11.6.4 .

11.6.2 Catalyst Fundamentals

Following this introduction to automotive catalysts, we now develop several relevant fundamental concepts. Figure 11.54 illustrates the role of the catalyst in enabling reactions in more detail (e.g., between a given hydrocarbon molecule and oxygen in the exhaust gases at the catalyst surface) at temperatures of order 300 to 600°C. Such diagrams indicate the state (enthalpy) of the reactants as they transition to end-products (CO_2 and H_2O in the example of HCs above) as the reaction coordinate, reactants to products, progresses. The heat of reaction (ΔH; see Sec. 3.5.1 and Fig. 3.5), the chemical energy released by the reaction, is not changed by the presence of the catalyst. However, the presence of an effective catalyst (e.g., the noble metal platinum) significantly reduces the energy (E_A) required to break the molecular bonds in the reactant molecules. Thus the desired chemistry occurs at the active site on the catalyst surface at a significantly lower temperature than would be needed to obtain sufficiently energetic molecular collisions in the gas phase. The catalyst provides a lower energy barrier (E_A) as reactants are transformed to products, which increases the rates of the reactions involved. The changes in enthalpy ΔH and free energy ΔG, initial to final

state, are neither affected by the catalyst nor is the final equilibrium. The rate of reaction varies inversely to the activation energy. The activation energy for the entire reaction E_A, is primarily governed by the slowest of all the reaction steps involved in the overall reactants to products chemical process. This slowest step is the rate-limiting step in the entire process.

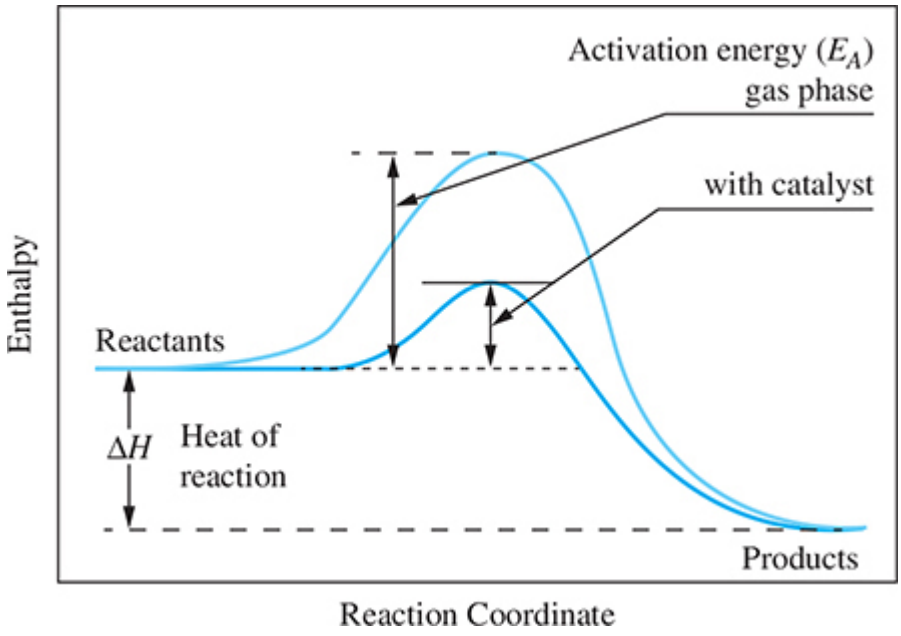


Figure 11.54 Reaction energy paths for reactants transitioning to products illustrating the role of the activation energy barrier. For an exothermic reaction (e.g., CO oxidizing to CO_2) where, overall, thermal energy is released (ΔH is positive). The presence of the catalyst significantly reduces the energy barrier to the reaction.

Catalysts are characterized by two basic properties: their *activity* and their *selectivity*. Activity relates to the degree to which the catalyst increases the rate of the relevant chemical reactions. Normally, a higher activity—faster reaction rate—in a given context is more desirable. Reaction rates are proportional to the concentrations of the reacting species, each raised to the power of its stoichiometric coefficient in the relevant chemical reaction (see Sec. 3.7.2). The expressions for and values of the rate constants for the slower reactions are critical. The catalyst enhances the transition to chemical equilibrium: it does not change that equilibrium (see Sec. 3.7.1).

A catalyst can be selective: that is, accelerate certain specific reactions much more than others. A catalyst's selectivity, which is usually designed into a catalyst through the elements incorporated in its formulation, is also usually dependent on its temperature: it enhances specific reactions within a temperature window. Diesel NO_x catalysts are "selective" in this way.

Automotive catalysts are heterogeneous catalysts: that is, the active noble metal catalyst materials are dispersed over a solid surface. Their structure and mode of operation are illustrated in Figs. 11.55 and 11.56. Monoliths (honeycomb like structures) are widely used in automotive applications as supports. As shown in Fig. 11.55, the exhaust is directed through square cross-section channels [e.g., 400 to 600 channels/in ^{2}, 60 to 90 channels/cm ^{2}; some 15 cm (6 in.) long], which expose the exhaust gas to a large surface area for sufficient time. Such a monolith support is extruded. Synthetic cordierite, a combination of magnesia, alumina, and silica, is the material used ($2MgO.2Al_2O_3.5SiO_2$). High melting point (over 1300°C), low thermal expansion coefficient (hence good resistance to thermal shock), low heat capacity (for rapid warm-up), and adequate strength are its key properties. The monolith is surrounded by a flexible seal and support inside a steel can as shown.

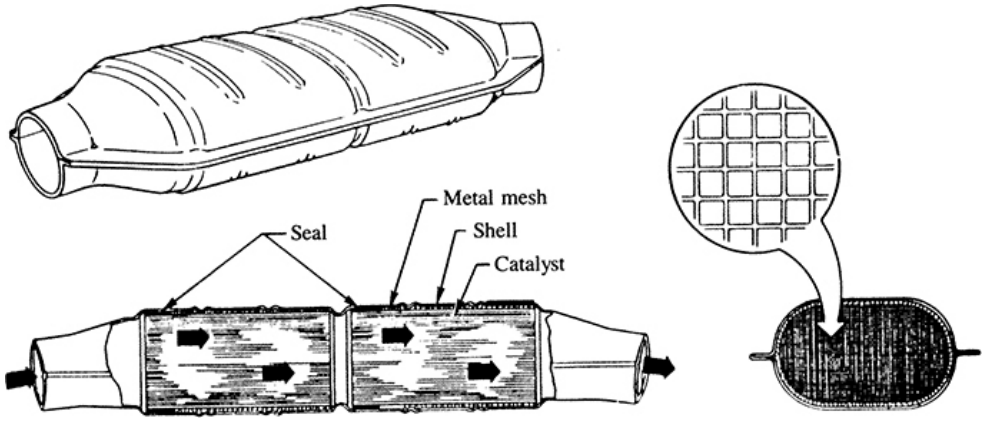


Figure 11.55 Monolith design of catalytic converter for spark-ignition engine emissions control.

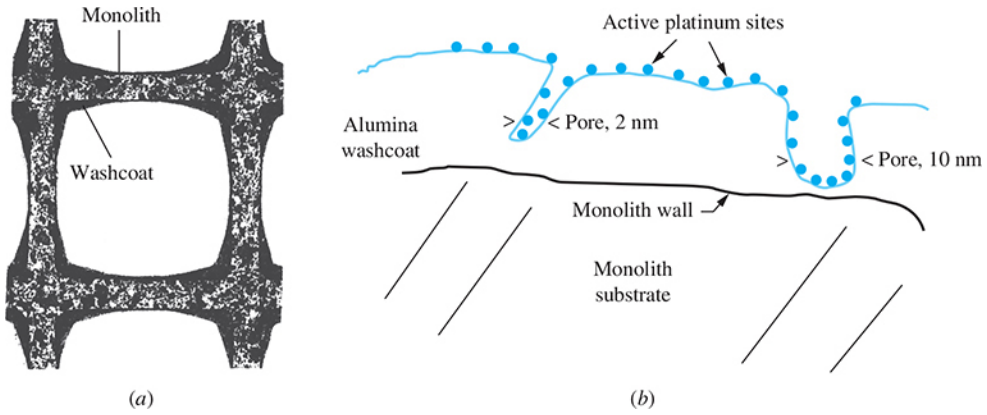


Figure 11.56 (*a*) Cross-section showing a monolith channel and washcoat (about 1 mm^2 flow area); (*b*) schematic of catalytically active (Pt) sites dispersed over the washcoat surface which is deposited on the monolith substrate walls.[78]

The critical catalyst component is the *washcoat*. Commercial ceramic monoliths have large pores and relatively low surface area (e.g., 0.3 m^2/g). The washcoat, deposited on the substrate, through its high porosity provides a large surface area (100 m^2/g) on which to disperse the noble metal catalyst material. The washcoat material can also enhance the activity of the catalyst through catalyst component/support interactions, and stabilize the precious metals against deactivating mechanisms such as sintering. Figure 11.56(*a*) illustrates the layout and role of the washcoat. The base material is γ–alumina often with several of these oxides—cerium, lanthanum, iron, nickel, barium, and neodymium—added. The monolith may be dipped in a washcoat slurry (with the catalytic metals included); or a two-step process may be used where the monolith is first coated with an non-catalytic layer followed by dipping it into a solution containing the catalytic salts which are absorbed onto the washcoat surface, which is then dried and calcined by heating in the air.

The noble metal catalyst atoms are dispersed over this highly porous washcoat surface as illustrated in Fig. 11.56 *b*. The surface of the alumina is rich in surface OH^- groups, which are sites at which the noble metal catalytic material can bond. The smaller the size of the tiny latinum (and palladium, and rhodium) particles (or crystallites), the more dispersed these active sites will be, and the higher the catalytic activity.

The conversion of reactants (e.g., hydrocarbons and oxygen) to products (CO_2 and H_2O) at the catalyst surface involves the following physical and chemical steps. First, bulk diffusion of the reactants within each of the channels toward the walls occurs. Diffusion into the pores of the washcoat toward the internally dispersed catalyst material then follows. Next, the reactant molecules must chemisorb at the surface onto an active (catalytic) site. If the surface temperature is high enough (so the reactant molecules are at or above the peak of the activation energy barrier shown in Fig. 11.54), the reactants undergo the chemical reactions that convert them to products which then desorb from the catalytic sites. Products then diffuse back through the pore network to the outer washcoat surface. Finally, the product species diffuse into the bulk gases flowing through the monolith channels.

These steps—reactant bulk mass transfer, pore diffusion, chemisorption and reaction to products—followed by the reverse sequence for the products, occur at different rates which are dependent on temperature. Chemical reaction rates follow an exponential dependence on temperature, $\exp(-E_A/RT)$, where E_A is the activation energy (see Fig. 11.54). In automotive catalysts, E_A is typically 40 kJ/mol or greater. Pore diffusion is largely dependent on pore size and geometry: the rate at which reactants and products diffuse into the pores varies approximately with the square root of the rate constant for the chemical reaction, thus giving it an effective activation energy of about half that of the chemical reaction, 25 to 40 kJ/mol. The bulk diffusion/mass transfer rates vary approximately as $T^{3/2}$, which corresponds to some 8 to 16 kJ/mol. At low temperatures, the chemical reaction rates are slow relative to diffusion rates. As temperature increases, the steps with the highest activation emerges (and exponential dependence) increase the fastest and pore diffusion becomes limiting. At high temperatures both chemical and pore diffusion rates become sufficiently fast that bulk diffusion with its lessor dependence on temperature becomes rate limiting. Equations that describe the reaction chemistry and species diffusion processes can be used to quantify this sequence of individual processes and thus obtain estimates of species conversion rates in these monolith high-surface-area catalysts. [79]

Automotive catalysts have been developed to the point where they are extremely effective at removing air pollutants from engine exhaust gases. Catalyst effectiveness is quantified through its catalyst efficiency which is

defined as the ratio of the amount of a particular air pollutant (e.g., HC) removed by the catalyst to the amount of that pollutant entering the catalyst:

$$\eta_{cat,HC} = \frac{\dot{m}_{HC,in} - \dot{m}_{HC,out}}{\dot{m}_{HC,in}} = 1 - \frac{\dot{m}_{HC,out}}{\dot{m}_{HC,in}} \tag{11.43}$$

Figure 11.57 illustrates the layout of a gasoline engine plus three-way catalyst system with its relative air/fuel ratio sensor to ensure stoichiometric operation, and provides the context for this catalyst efficiency definition. Catalyst efficiencies are now well-above 90%, and with clean fuels can maintain these high efficiencies over extended mileage. Note that $\dot{m}_{HC,out}/\dot{m}_{HC,in}$ is the catalyst *inefficiency*, and is 5% for a 95% catalyst efficiency corresponding to a factor of 100 ÷ 5 = 20 reduction in engine-out emissions. Catalyst efficiency increases as temperature increases in an "S-shaped" curve. At the low temperature end of the curve there is little catalyst activity. At some point, the catalyst "lights off" and catalytic chemical activity becomes significant (with HC and CO oxidizing automotive catalysts, significant chemical energy is released within the catalyst which further raises its temperature. A commonly used *light-off temperature* definition is the temperature (T_{50}) at which the catalyst efficiency reaches 50%.

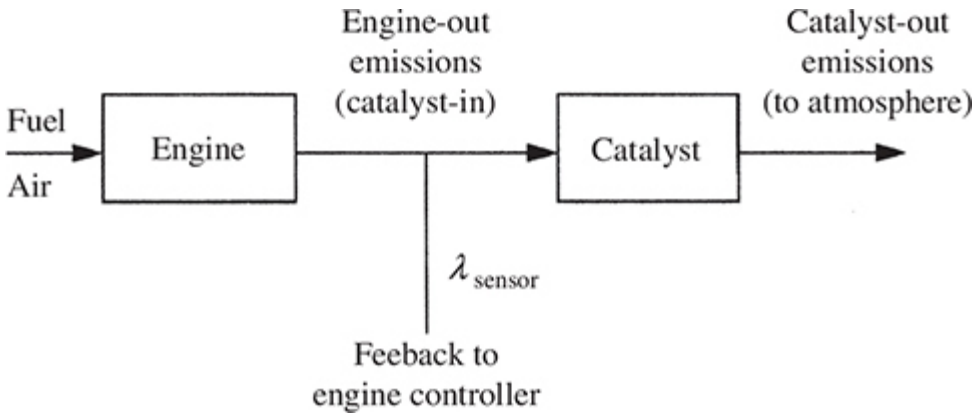


Figure 11.57 Layout of spark-ignition engine plus three-way catalyst system, with its λ-sensor (and feedback to the engine control system), to maintain stoichiometric operation.

While catalysts are not consumed during catalytic reactions, they steadily (though slowly) deactivate with use. The important deactivation modes are poisoning, fouling or masking, and thermal deactivation or sintering. Figure 11.58 illustrates these catalyst deactivation processes. Poisoning is a major cause of deactivation. Contaminants (usually in the fuel or oil consumed) such as lead, phosphorus, sulfur, and other inorganic metals, deactivate the noble metal (e.g., Pt) active sites by forming catalytically inactive alloys, or chemi-absorb onto active sites and block those sites from further catalytic action. The former mode (e.g., with phosphorus) results in permanent deactivation. The latter site blocking (e.g., with sulfur) can be reversed by desorbing the poison from the catalyst surface. Figure 11.58 *a* illustrates this catalyst poisoning mode: the number of active sites is reduced so activity decreases. Masking or fouling of the catalyst washcoat, illustrated in Fig. 11.58 *b*, covers portions of the washcoat surface by physically depositing high molecular weight HCs from the oil, components of the oil additive package, scale from corrosion, dusts, etc., as a glaze over portions of the washcoat surface. These "deposits" prevent exhaust gas species from reaching active catalytic sites and also narrow and block entrances to the smaller washcoat pores. The third mode is thermal (high temperature) deactivation or sintering, as shown in Figs. 11.58*c* and *d*. Here, under higher than intended surface temperatures, the active sites (fine catalyst crystals) can migrate over the surface, coalesce and form larger crystals at fewer active sites, thus decreasing catalytic activity. At higher temperatures still, the porosity of the alumina-based washcoat may be reduced through "carrier sintering" (Fig. 11.58*d*). Note that loss of active sites through selective poisoning and catalyst sintering primarily affects the reaction kinetics: thus higher temperatures are needed to achieve a given activity level but the maximum conversion efficiency may well be unaffected. Catalyst deactivation resulting from increased pore diffusion resistance manifests itself as lower conversion efficiency, and thus changes the efficiency versus temperature curve slope. Masking causes both loss of active (surface) sites and pore blocking. Thus the conversion efficiency curve is shifted toward higher temperatures with a significant reduction in slope. Note that with clean fuels, low oil consumption, and adequate control of catalyst temperatures, the deterioration in catalyst efficiency with extensive use (mileage in a vehicle) can be held within tolerable bounds. [78,79]

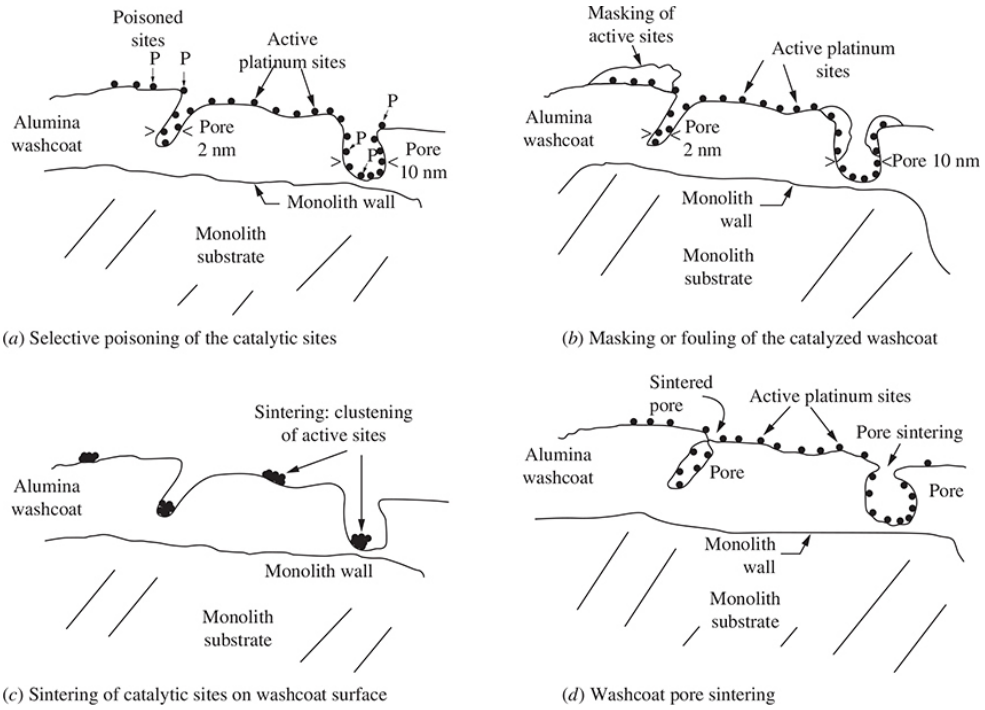


Figure 11.58 Catalyst deactivation processes: (*a*) selective poisoning (P) of active platinum (Pt) sites; (*b*) masking or fouling of sections of the washcoat surface; (*c*) migrating and coalescence of Pt active sites on the washcoat surface due to higher than intended temperatures; (*d*) washcoat pore sintering due to higher still temperatures, which reduces washcoat porosity.[78]

The above discussion implies that the air pollutants to be removed from the exhaust gases must spend sufficient time within the catalyst to complete the sequence of processes that essentially complete removal requires. The catalyst must be sized so that over the operating range of the engine, in a given application, this constraint is satisfied. The residence time within the catalyst is defined in terms of the volumetric flow of exhaust gases and the catalyst volume through the space velocity, SV:

$$\text{SV} = \text{volume flow rate of exhaust/catalyst volume} \tag{11.44}$$

The volume flow rate is at standard atmospheric pressure and temperature. The catalyst volume is the volume of the honeycomb monolith. [1] The normal units are reciprocal hours, h^{-1}. In automotive applications, the maximum

space velocity is limited to about 50,000 to 100,000 h^{-1} as indicated in Fig. 11.59. The inverse of the space velocity is the exhaust gas residence time within the catalyst. 50,000 h^{-1} corresponds to a 70-ms residence time, which at 2000 rev/min is about two crank revolutions. Too high a flow rate (too short a residence time) results in a reduction in catalyst activity.

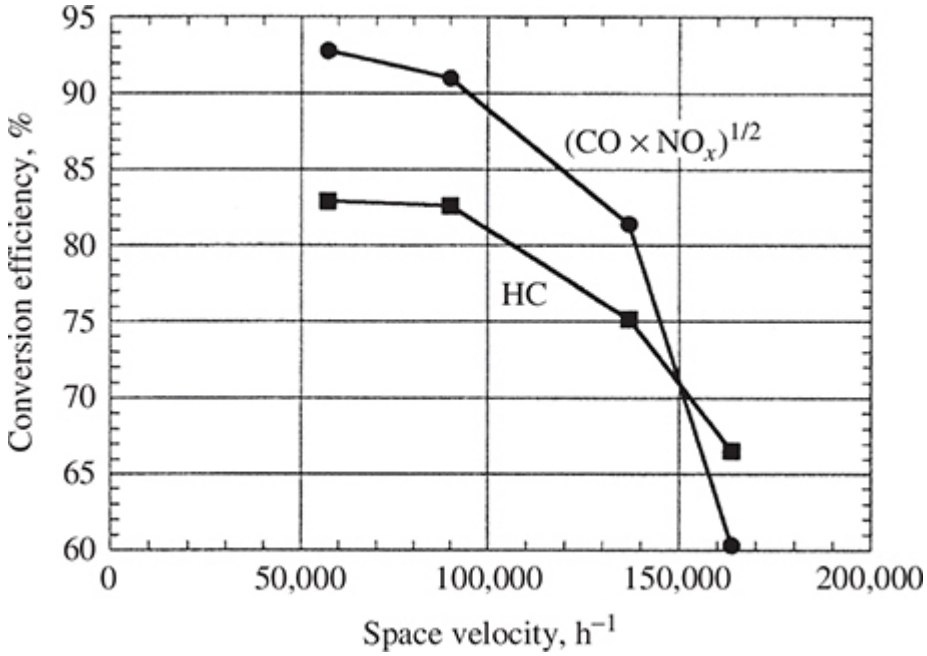


Figure 11.59 Effect of space velocity [SV, defined in Eq. (11.44)] on catalyst conversion efficiency. Maximum conversion efficiency achieved at below about 50,000 (h^{-1}). [80]

Note that a spark-ignition engine operating under conditions typical of light-duty vehicle urban driving (about one-quarter the maximum torque, and at low engine speeds), the exhaust gas flow is about one-tenth that at maximum rated-power conditions (WOT and 5000 to 6000 rev/min). Under normal driving conditions, space velocities are much below the full power value. Thus (with appropriate catalyst temperatures) only the front axial portion of the catalyst monolith and washcoat is needed to achieve high catalyst efficiencies. At high loads and speeds, that is at high space velocities, then close to the full catalyst length is needed to remove almost all the engine-out emissions.

The size and geometry of the catalyst substrate also affect the pressure drop across the catalytic converter. The typical monolith substrate converter (see Fig. 11.55) has an entrance section, the monolith substrate section (with some 10,000 channels), and an exit section. The flow losses in the entry and exit header sections depend on exhaust pipe diameter and other geometric details, and are small. The flow blockage the monolith structure causes, creates contraction and expansion losses as the exhaust flows into and out of the monolith. Thus the overall monolith pressure drop, Δp_c, is:

$$\Delta p_c = \Delta p_m + \Delta p_{\text{ch}} + \Delta p_{\text{out}} \tag{11.45}$$

The channel pressure drop Δp_{ch} is about 90% of the total: Δp_{in} and Δp_{out} are relatively small.

The flow in the monolith channels is laminar, and the pressure drop is given by the Darcy equation:

$$\Delta p_{\text{ch}} = 4f(L/d_{\text{ch}})(\rho v^2/2) \tag{11.46}$$

where f is the friction factor, L and d_{ch} are the length and hydraulic diameter of each channel, r is the exhaust gas density, and v the gas velocity in the channels. (The gas velocity is obtained from the engine mass flow rate divided by the gas density and the substrate open area.)

The flow in catalyst monolith channels is always laminar, so the friction factor f is then given by

$$f = K/(\text{Re}) \tag{11.47}$$

where K for round channels is 16, and for square cross-section channels is 14.2 for Reynold's numbers (Re) < 2100, the laminar flow regime. The contraction and expansion flows into and out of the monolith channels can be transitional or turbulent. Friction coefficients for these flows can be estimated from the following correlations:

$$K_{\text{in}} = 1.08 - 0.415(A_F/A) \tag{11.48}$$

and

$$K_{out} = (1 - A_F/A)^2 \tag{11.49}$$

where A_F / A is the ratio of the substrate (open) flow area to the total substrate cross-sectional area. Combining Eqs. (11.45) to (11.49) gives the total catalyst substrate pressure drop:

$$\Delta p_c = [4f(L/d_{ch}) + K_{in} + K_{out}](\rho v^2/2) \tag{11.50}$$

This pressure drop is comparable in magnitude to the exhaust pipe pressure drop, and to the muffler pressure drop.

11.6.3 Catalytic Converters

The catalytic converters used to clean up the engine's exhaust gases, as explained in the previous section, consist of active catalytic material on the surface of a ceramic monolith in a specially designed metal casing which directs the exhaust gas flow through the catalyst bed. The active material employed for CO and HC oxidation or NO reduction (normally noble metals, though base metals oxides can be used) must be distributed uniformly over a large surface area so that the mass-transfer characteristics between the gas phase and the active catalyst surface are sufficiently fast to enable close to 100% conversion efficiency. The converter configuration commonly used is shown in Fig. 11.55. The active noble metal catalyst material is impregnated into a highly porous alumina washcoat about 20 μm thick that is applied to the passageway walls. The typical monolith has square-cross-section passageways with inside dimensions of ~1 mm separated by thin (0.15 to 0.3 mm) porous walls. The number of passageways per square centimeter varies between about 60 and 100 (400 to 700 passageways/in ^{2}). The washcoat, 5 to 15% of the weight of the monolith, has a highly porous surface area of area 100 to 200 m ^{2}/g. The gas flow is directed down through the bed as shown to provide a large flow area and catalytic surface area, and low-pressure drop. While the exhaust gas flow is turbulent, in the monolith catalyst passageways, it is laminar.

Oxidation Catalysts

Oxidation catalysts were the first type of catalytic converter widely used to reduce engine-out HC and CO emissions with gasoline engines in the 1970s.

Their use declined in the 1980s as strict control of NO_x emissions became mandatory and three-way catalysts which, with stoichiometric engine operation removed all three pollutants, became the dominant SI engine catalyst technology (see below). The function of an oxidation catalyst engine is to oxidize CO to CO_2, and hydrocarbons to CO_2 and water, in the exhaust gas stream which typically contains about 10% CO_2, 12% H_2O, 100 to 2000 ppm NO, SO_2 at the ppm level, 1 to 2% O_2, 0.2 to 2% CO, and 1000 to 5000 ppm C_1 HC. About half the hydrocarbons emitted by the SI engine are unburned fuel compounds. The saturated hydrocarbons (which comprise some 20 to 30%) are the most difficult to oxidize. The ease of oxidation increases with increasing molecular weight. Sufficient oxygen must be present to oxidize the CO and HC. This may be supplied by the engine running slightly lean of stoichiometric or with a secondary air pump that introduces air into the exhaust ports just downstream of the valve (see secondary air discussion on p. 648).

Because of their high intrinsic activity, noble metals are the most effective catalytic materials. They show higher specific activity for HC oxidation, are more thermally resistant to loss of low-temperature activity, and are much less deactivated by the sulfur in the fuel than base metal oxides. A mixture of platinum (Pt) and palladium (Pd) is commonly used. For the oxidation of CO, olefins, and methane, the specific activity of Pd is higher than that of Pt. For the oxidation of aromatic compounds, Pt and Pd have similar activity. For oxidation of paraffin hydrocarbons (with molecular size greater than C_3), Pt is more active than Pd. Pure noble metals sinter rapidly in the 500 to 900°C temperature range experienced by exhaust catalysts. Since catalytic behavior is manifested exclusively by surface atoms, the noble metals are dispersed as finely as possible on an inert support such as γ-A_2O_3 which prevents particle-to-particle metal contact and suppresses sintering. The size of the noble metal particles in a fresh catalyst is less than 50 nm. This can increase to ~100 nm when the catalyst is exposed to the high temperatures of the exhaust in vehicle operation. Typical noble metal concentrations in a commercial monolith catalyst are between 1 g/liter and 2 g/liter of monolith volume, with Pt/Pd ≈ 2 on a weight basis. As a rough rule of thumb, the ceramic honeycomb volume required is about half the engine displaced volume. This gives a space velocity through the converter (volume flow rate

of exhaust divided by converter volume, Eq. 11.44) over the normal engine operating range of 5 to 20 per second (20,000 to 70,000 h^{-1}).

The *conversion efficiency* of a catalyst is the ratio of the rate of mass removal in the catalyst of the particular pollutant of interest to the mass flow rate of that pollutant into the catalyst, Eq. (11.43) . The variation of conversion efficiency of a typical oxidizing catalytic converter with temperature is shown in Fig. 11.60. At high enough temperatures, the steady-state conversion efficiency of a new oxidation catalyst is typically 98 to 99% for CO and above 95% for HC. However, the catalyst is ineffective until its temperature has risen above the *light-off temperature* (about 250°C) at which point the catalyst becomes more than 50% effective.

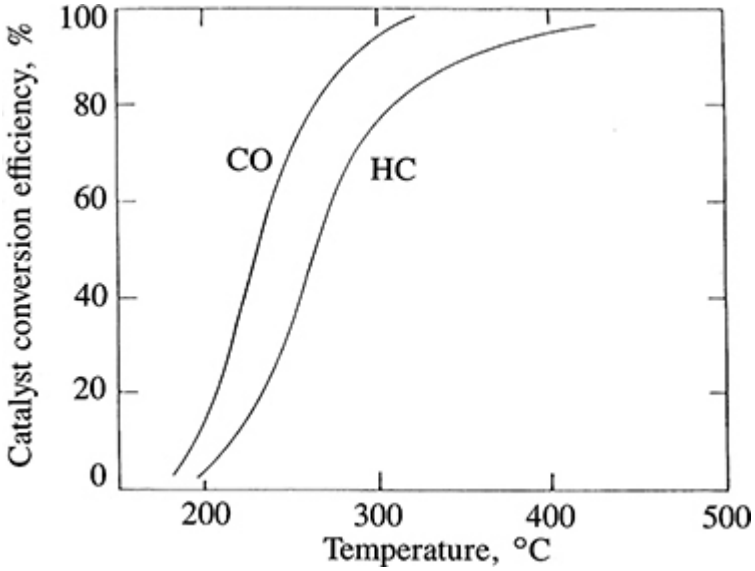


Figure 11.60 Conversion efficiency for CO and HC as a function of temperature for typical oxidizing catalytic converter.[35]

The above numbers apply to new noble-metal oxidation catalysts: as catalysts spend time in service their effectiveness deteriorates. Catalysis involves the adsorption of the reactants onto surface sites of high activity, followed by chemical reaction, then desorption of the products. As explained in Sec. 11.6.2 , catalyst degradation involves both the deactivation of these sites by catalyst poisons and a reduction in the effective number of these sites through sintering. Poisoning affects both the warm-up and steady-state performance of the catalyst. When poisoning occurs, catalytic activity is

impeded through prolonged contact with interfering elements that either physically block the active sites or interact chemically with the active material.

A material balance across the catalyst monolith, assuming steady-state, one-dimensional plug flow operation gives

$$\frac{d(vC)}{dz} = -R \tag{11.51}$$

where v is the exhaust flow velocity, C the concentration of the reactant of interest (g-moles/cm ^{3}), z is the axial distance, and R the rate of reaction (g-moles/cm $^3 \cdot$ s). If the reactant concentration is low, the chemical energy release is modest, so the flow can be treated as isothermal and Eq. (11.51) simplifies to

$$v\frac{dC}{dz} = -R \tag{11.52}$$

Since $C_{HC} \ll C_{O_2}$, there is an excess of O_2 so the reaction rate depends to first order on the hydrocarbon concentration: for example, for ethane oxidation

$$C_2H_6 + \frac{7}{2}O_2 \rightarrow 2CO_2 + 3H_2O$$

the reaction rate equation is

$$v\frac{dC_{C_2H_6}}{dz} = -k'C_{C_2H_6}$$

which integrates to

$$\ln\frac{C_o}{C_i} = -\frac{k'z}{v} = -k't \tag{11.53}$$

where subscripts o and i denote catalyst-out and -in concentrations, and t is the residence time of C_2H_6 in the catalyst bed (the reciprocal of the space

velocity, Eq. 11.44). Rate constants, such as k', are determined experimentally. [79] Similar relationships can be written for the oxidation rates of the more reactive olefinic and aromatic hydrocarbon.

The oxidation kinetics of CO over Pt and Pd noble metal catalysts can be described by

$$\frac{d[CO]}{dt} = \frac{K_1 p_{CO} p_{O_2}}{(1 + K_2 p_{CO} + K_3 p_{HC})^2 (1 + K_4 p_{NO}^n)} \tag{11.54}$$

where K_1 to K_4 and n are constants at any given temperature, and p_{CO}, p_{O_2}, p_{HC}, and p_{NO} are the partial pressures of carbon monoxide, oxygen, hydrocarbons, and nitric oxide, respectively. These relationships incorporate the fact that the rates of HC and CO oxidation are inhibited by high CO and reactive HC concentrations, and that NO concentrations in the range of 0 to 1000 ppm inhibit HC and CO oxidation also. [81]

Oxidation-catalyst equipped vehicles may emit sulfuric acid aerosol. Gasolines contain between about 10 and several hundred ppm by weight S (depending on fuel specifications), which leaves the combustion chamber as SO_2. This SO_2 can be oxidized by the catalyst to SO_3, which combines with water at ambient conditions to form an H_2SO_4 aerosol. The SO_3 can be chemisorbed on the alumina catalyst surface. SO_3 production can be controlled by lowering or raising the catalyst temperature. Figure 11.61 shows that at low temperatures, SO_3 production is kinetically limited; at high temperatures SO_3 production is thermodynamically limited. Palladium and rhodium catalysts produce less SO_3 than Pt and have comparable HC and CO catalytic activity. By decreasing oxygen concentrations leaving the catalyst to ~1%, SO_3 production can be substantially reduced. [81]

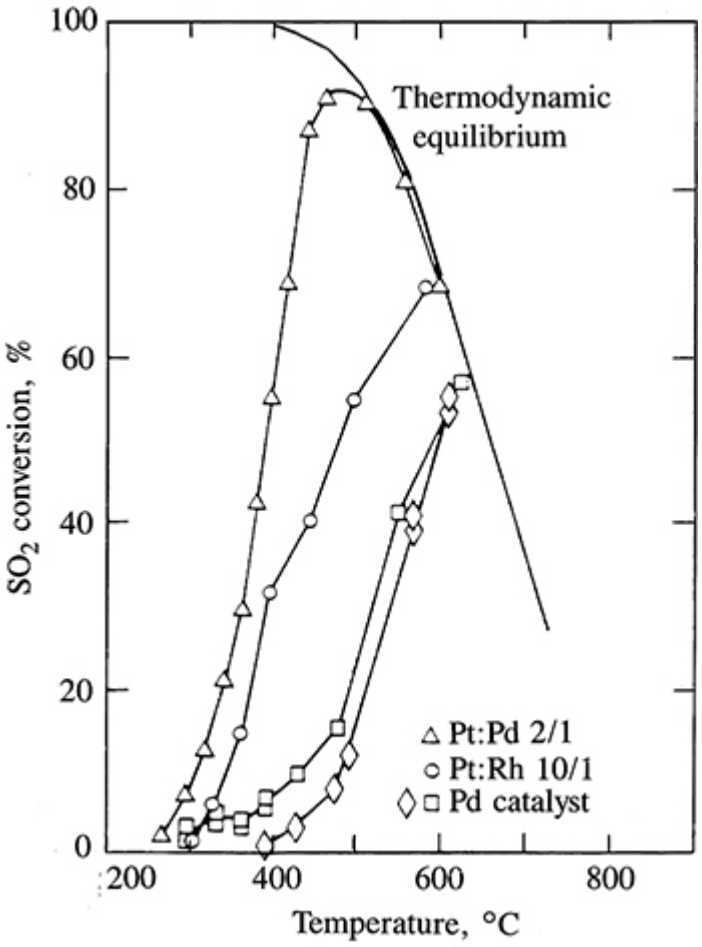


Figure 11.61 SO_2 conversion to SO_3 within an oxidizing catalytic converter, as a function of temperature, with 5% O_2 concentration and no reducing gases present. Space velocity (volume flow of exhaust per unit catalyst volume) ~10 s^{-1}. Results for Pt–Pd, Pt–Rh, and Pd catalysts. [82]

Oxidation catalysts, no longer widely used in gasoline SI engines, have come back as an important component of diesel aftertreatment systems. Oxidation of diesel HC emissions became important from a regulatory point of view, and because high-molecular-weight hydrocarbons condense and contribute to particulate emissions (termed the soluble organic fraction, SOF, of the total particulate matter). Data suggest that this SOF of the diesel particulate matter varies significantly with engine design details and operating conditions, but can be some one-third of the total particulate matter. [83] An oxidation catalyst is able to remove much of this SOF, but exhaust gas temperatures are usually too low to oxidize the insoluble soot fraction of PM.

Diesel exhaust gas is inherently lean of stoichiometric so there is always excess oxygen. While that lowers the effective HC and CO oxidation catalyst light-off temperatures, the diesel exhaust gases are significantly cooler than in stoichiometric-operating gasoline engines so overall catalyst efficiencies are lower than is achieved in gasoline-engine oxidation catalysts, rising to about 70% for HC and approached 90% for CO. However, DOCs do remove a significant fraction of the aldehydes and ketones responsible for diesel exhaust odor, and the polycyclic aromatic hydrocarbons, an especially toxic component of diesel exhaust. [79,83]

Three-Way Catalysts

When a gasoline engine is operated at the stoichiometric air/fuel ratio, then both NO reduction and CO and HC oxidation can be effected in a single catalyst bed. The catalyst brings the exhaust gas composition to a near-equilibrium state at these exhaust conditions; i.e., a composition of CO_2, H_2O, and N_2. Enough reducing gases will be present to reduce NO and enough O_2 to oxidize the CO and hydrocarbons. Such a catalyst is called a *three-way catalyst* since it removes all three pollutants simultaneously. Figure 11.62 shows the conversion efficiency for NO, CO, and HC as a function of the actual and relative air/fuel ratio. There is a narrow range of air/fuel ratios close to stoichiometric in which high conversion efficiencies for all three pollutants are achieved. The width of this window is narrow, about 0.1 air/fuel ratios (7×10^{-3} in relative air/fuel or equivalence ratio units), and depends on catalyst formulation details and engine operating conditions.

The overall chemical conversions that remove the three pollutants under stoichiometric exhaust conditions are these:

Oxidation:

$$\text{CO} + \text{oxygen} \rightarrow \text{CO}_2$$

$$\text{HC} + \text{oxygen} \rightarrow \text{CO}_2 \text{ and } \text{H}_2\text{O}$$

Reduction:

$$\text{NO} + \text{CO, HC, or } \text{H}_2 \rightarrow \text{N}_2 + \text{CO}_2, \text{H}_2\text{O}$$

Usually the catalyzed CO reactions begin first, followed by the HC and NO reactions. With hot exhaust gases in a warmed-up gasoline engine, the chemical reaction rates are fast, and the overall conversion rates are controlled by pore diffusion and/or bulk mass transfer. [78]

The relative air/fuel ratio window for achieving both oxidation and reduction simultaneously is too narrow to be realized in an operating engine without *closed-loop* relative air/fuel ratio control. An oxygen sensor in the exhaust is used to indicate whether the engine is operating on the rich or lean side of stoichiometric, and provide a signal for adjusting the fuel system to achieve the desired air-fuel mixture (see Sec. 7.4). Holding the equivalence ratio precisely on the chosen close-to-stoichiometric value is not a practical expectation of such a feedback system, so oxygen storage capacity (OSC) is built into the washcoat. Ceria (CeO_2) is the most commonly used oxygen storage component. It can release oxygen (forming Ce_2O_3) when the exhaust gas at the surface of the catalyst is slightly rich: then Ce_2O_3 can absorb and store oxygen from the exhaust gas on the catalyst surface when the gas is lean, by transitioning to CeO_2: that is,

$$2CeO_2 \rightleftarrows Ce_2O_3 + O$$

Ceria constitutes some 10 to 20% of the γ-Al_2O_3 washcoat material. [83] It has been found that addition of zirconia to the ceria (in roughly comparable amounts) in the washcoat improves the stability of the ceria and promotes oxygen ion mobility.

To maintain this OSC in a state where oxygen can always be released or absorbed in response to random relative *A/F* fluctuations in the engine's exhaust gases, the engineair/fuel ratio control system is oscillated about stoichiometric, rich and lean at a given amplitude and frequency, to maintain the desired storage level. Amplitudes of ± about 0.5 *A/F* ratio at between 0.5 Hz and 1 Hz frequency provide optimum performance. Thus the actual exhaust gas relative air/fuel ratio window within which all three pollutants are effectively removed, is widened substantially. Note that while the engine-out exhaust gas composition varies around the stoichiometric composition (due to both this deliberate oscillation and through more random engine variations), the oxygen storage capability ensures that the gas composition at

the three-way catalyst surface remains within the narrow high-efficiency close-to-stoichiometric band shown in Fig. 11.62.

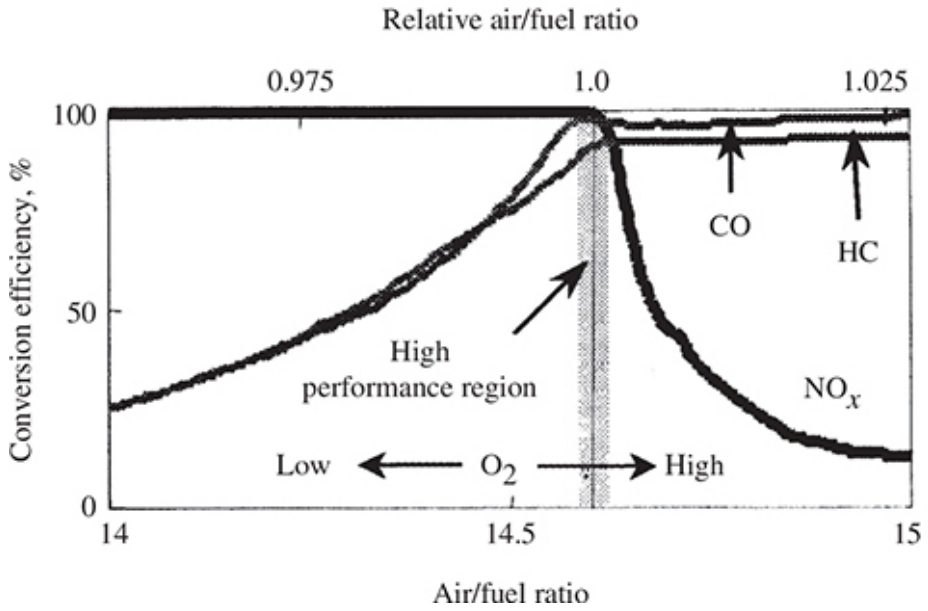


Figure 11.62 Conversion efficiency for NO, CO, and HC for a three-way catalyst as a function of exhaust gas air/fuel ratio.[84]

The noble metal catalysts utilized in TWCs are combinations of platinum, rhodium, and palladium. Rhodium is the principal ingredient used in commercial Pt-Rh catalysts to remove NO. It is very active for NO reduction, is much less inhibited by CO and sulfur compounds, and produces less NH_3 than Pt. To remove NO under slightly lean-of-stoichiometric conditions, the catalyst must react the CO, H_2, or HC with NO rather than with O_2, as the exhaust gas passes through the catalyst bed. Rhodium shows some NO reduction activity slightly lean of stoichiometric. On the rich side, the three-way catalyst window is determined by hydrocarbon and CO removal. Platinum is commonly used for HC and CO oxidation; it has good activity under stoichiometric and slightly lean conditions. When sufficient rhodium is present, the participation of Pt in NO removal is minimal. In the rich regime, the three-way catalyst consumes all the oxygen that is present in the exhaust, and as a consequence removes an equivalent amount of CO, H_2, and hydrocarbons; it is thought that the H_2 is removed first. In addition, the water-gas shift reaction

$$CO + H_2O \rightleftarrows H_2 + CO_2$$

and the steam-reforming reaction

$$\text{Hydrocarbon} + H_2O \rightarrow CO, CO_2, H_2$$

can consume CO and HC. (Ceria also promotes these reactions.) The exhaust contains an H_2/CO molar ratio of about one-third (see Sec. 4.9.1): the equilibrium ratio at 500°C is about 4. Considerable CO removal can be expected if the water-gas shift equilibrium is approached. Platinum is active in promoting this equilibrium. For large molecular weight paraffin hydrocarbons, and for olefins and aromatic hydrocarbons, the equilibrium for the steam-reforming reactions lies to the right. This reaction can therefore lead to considerable hydrocarbon removal. Rhodium is particularly active in the steam-reforming reaction; platinum is also active. Commercial three-way catalysts contain platinum and rhodium (the ratio Pt/Rh being around 5). The amounts used are a few grams of Pt-Rh per liter. Alumina is the preferred support material. Ceria is the oxygen storage component. [80] Palladium is also used in three-way catalysts, alone or with rhodium. It is used in greater amounts (several times that of Pt-Rh—some 10g/liter). It has a lower light-off temperature than Pt-Rh catalysts. Its performance is comparable to that of Pt-Rh catalysts. [83]

The effectiveness of the catalyst depends on its operating temperature, the availability of excess oxygen mixed throughout the reacting gases, and the monolith surface area. The operating temperature depends on the inlet gas temperature, heat losses, and the amount of HC, CO, and H_2 burned up in the reactor. This latter factor is important: 1.5% CO removal results in a 220 K temperature rise. As a consequence, catalytic converters with fuel-rich cylinder-exhaust gas and secondary air give greater fractional reductions in HC and CO emissions than reactors with fuel-lean cylinder exhaust (which do not require any secondary air).

A practical limitation to catalyst effectiveness with fuel-rich engine operation is adequate mixing of secondary air and engine exhaust gases in the exhaust ports and manifold. The secondary airflow with a conventional air pump is effectively shut off by the exhaust blowdown process, and little oxidation occurs in the exhaust port because the air and exhaust gases are

effectively segregated. Mixing can be promoted by suitably arranging the catalyst inlet and exit ports and by using baffles.

DeNO $_x$ Catalysis

NO is removed by reduction using the CO, hydrocarbons, and H $_2$ in the exhaust. No catalyst is available for the decomposition of NO to O $_2$ and N $_2$ (thermodynamically favored at exhaust gas temperatures) that is sufficiently active for use in engines. NO reduction can be carried out under rich conditions where there is an excess of reducing species over oxidizing species. The catalyst is then referred to as an *NO reduction catalyst.* Such an approach is used to regenerate a lean NO $_x$ trap (LNT; see below) when its NO $_x$ storage capacity reaches its limit. However, the rich operation necessary for NO reduction results in a fuel consumption penalty and constrains the performance of the NO catalyst since a fraction of the NO removed can be converted to ammonia NH $_3$ rather than N $_2$. NH $_3$ formation under rich operation must be small because the ammonia can be oxidized back to NO. Reduction of NO by CO, HC, or H $_2$ can be accomplished by base metal catalysts (e.g., CuO, NiO) in an appropriate temperature range (350 to 600°C). This "passive deNO $_x$" approach has not proved that attractive.

However, catalytic reduction of NO in lean operating engines, whether lean-at-light-load gasoline engines (e.g., stratified charge DI spark-ignition engines) or diesel engines, is an import emissions control need. Two approaches have been extensively explored and developed over the past several years, and are now in production. One is the lean NO $_x$ trap (LNT), which stores the NO $_x$ as nitrate when the engine is operating lean and then operates fuel-rich for a short period when the nitrogen oxides are released and then reduced to N $_2$. The second approach (used with light- and heavy-duty diesel engines) is selective catalytic reduction (SCR) of NO $_x$. In the predominant version of this approach, urea (stored on the vehicle) is titrated into the exhaust gases, converted to the reducing agent (ammonia), which then effects the NO $_x$ reduction in a quasi-steady manner. A second SCR strategy is to add and use hydrocarbons as the "titrated" reducing agent. These approaches will now be reviewed.

Lean NO_x Trap

This catalyst system absorbs and stores NO_x when an oxidizing exhaust gas flows through it and, since the storage capacity is limited, is periodically regenerated by a reducing purge. During this purge the engine is operated fuel-rich so exhaust gas constitutes can reduce the stored NO_x to N_2.

Figure 11.63 illustrates this operation. Under lean operating conditions, the platinum catalyst oxidizes the NO in the exhaust gases to NO_2. This is then absorbed and stored as an alkaline-earth (base metal, M) nitrate on the catalyst surface. Barium is usually the absorbant: Ca, Sr, and Mg are other alkaline earths that are also effective.

$$Ba + 2NO_2 + \frac{1}{2}O_2 = Ba(NO_3)_2 \qquad (11.55)$$

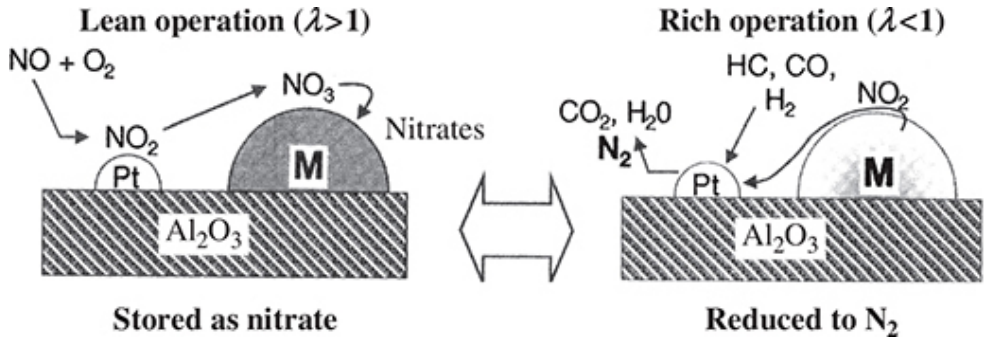


Figure 11.63 Schematic of NO_x storage-reduction mechanism of lean NO_x trap (LNT).[85]

The noble metal catalyst sites (Pt and Rh) must be in close proximity to the base nitrate-storage material. Alkali metal absorbers (Cs, K, Na) are added to the barium to improve catalyst efficiency, nitrate stability, and resistance to sulfur poisoning: see below.

When the nitrate storage capacity is essentially full, the trap is regenerated by rapidly transitioning the engine to rich operation (*A/F* 11.5 to 12.5) so the barium nitrate decomposes and the NO_2 released is reduced by the CO, HC, and H_2 in the now-rich exhaust gases. The decomposition of the barium nitrate occurs through the reverse of reaction (11.55) or its equivalent

$$Ba(NO_3)_2 = BaO + 2NO + 1\frac{1}{2}O_2 \tag{11.56}$$

followed by NO (and NO_2) reduction: for example,

$$NO + CO = \frac{1}{2}N_2 + CO_2 \tag{11.57}$$

Under these fuel-rich conditions at elevated temperatures, the nitrate species become thermodynamically unstable and decompose to NO and NO_2 [Eqs. (11.55) and (11.56)], and the HC, CO, and H_2 reduce these oxides of nitrogen on the noble metal catalyst to N_2, CO_2, and H_2O. Once the storage capacity has been regenerated, the engine returns to its (normal) lean operating mode.

Figure 11.64 illustrates how this regeneration process is controlled. The stoichiometry of the engine, when the regeneration process is initiated, transitions from its normal light-load lean operation, in stages, to well rich of stoichiometric (*A/F* about 12). To maintain constant torque, the throttle opening is reduced and the quantity of fuel injected per cycle is increased. [m] The rich-operation regeneration period is short (1 to 3 seconds). Once regeneration is complete, the return transient takes the engine back to its normal lean operating conditions. With these LNTs, regeneration occurs roughly every 60 s. There is therefore a fuel economy penalty of a few percent due to this frequent periodic regeneration process.

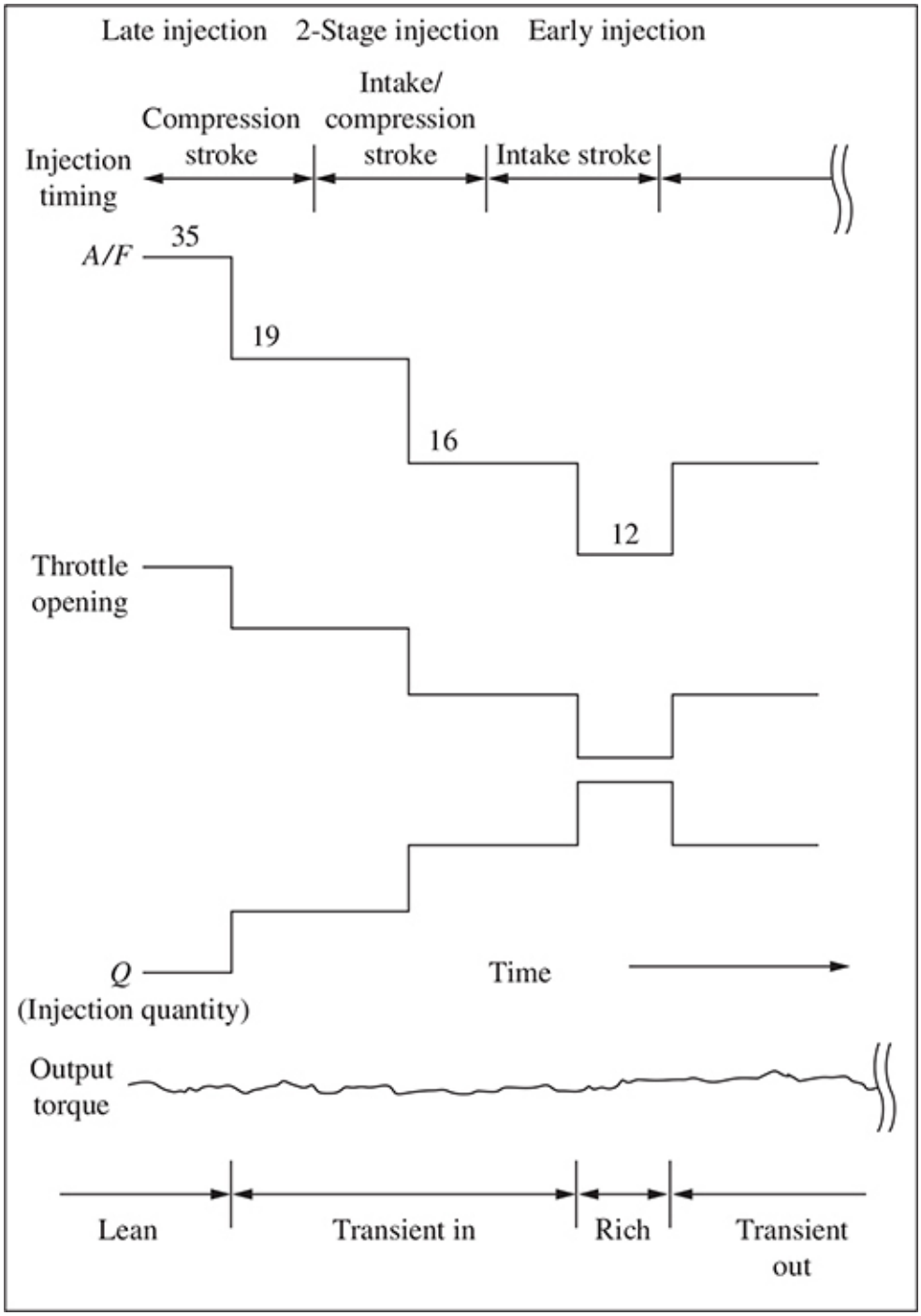


Figure 11.64 Regeneration control strategy of lean NO_x trap: steady-state lean operation (NO_x storage); transition to fuel-rich operation; fuel-rich regeneration and NO_x reduction; transition back to lean operation.[85]

An issue with LNTs is their susceptibility to sulfur poisoning. Just as barium and other alkaline bases store NO_x as nitrates, they can store sulfur oxides as sulfates. To maintain high catalyst efficiency over long periods of operation, the sulfur content of the fuel must be very low (a few ppm S). Some regions of the world have gasoline's and diesel fuels approaching these low levels. Many areas do not. In general sulfur poisoning of LNTs is reversible and catalytic site activity can be restored by a desulfation process which involves decomposition of the sulfate. This requires temperatures between about 500°C and 700°C accompanied by mixture enrichment. However, desulfation of LNTs does not quite restore the full nitrate storage capacity, so there is a requirement for extremely low S fuels to maintain high catalyst efficiency over extended operation. [79]

Various desulfation approaches are possible. For diesel engine LNT use, exhaust gas temperatures can be increased by additional late injections of fuel. With lean-operating gasoline engines, rapid large amplitude modulation of the (*A/F*) to enable exothermic reactions increases catalyst temperature (and minimizes the production of H_2S rather than SO_x). Another possibility is a sulfur trap ahead of the LNT.

The catalyst efficiencies achievable in LNTs, and the impact of sulfur and desulfation, are illustrated in Fig. 11.65. NO_x conversion efficiencies of more than 90% are realizable over the 300 to 400°C range with new catalyst systems (top set of curves). Additional developments are extending this high efficiency temperature range to 600°C. [86] When aged for 250 hours with 3 ppm S fuel, about a 20% loss in conversion efficiency occurs (next two lower curves). Higher levels of fuel sulfur (16 ppm S for 200 hours), and higher levels still (30 ppm S for 150 hours), further degrade efficiency as indicated in the third and fourth pairs of curves. Desulfurization of all these catalysts restores their NO_x conversion efficiency to close to their efficiencies before aging and exposure to fuel sulfur. Successive desulfurizations show modest loss in efficiency from the clean catalyst levels. Note 200 hours of "aging" corresponds to about 13,000 km (8,000 miles) of driving in an automobile.

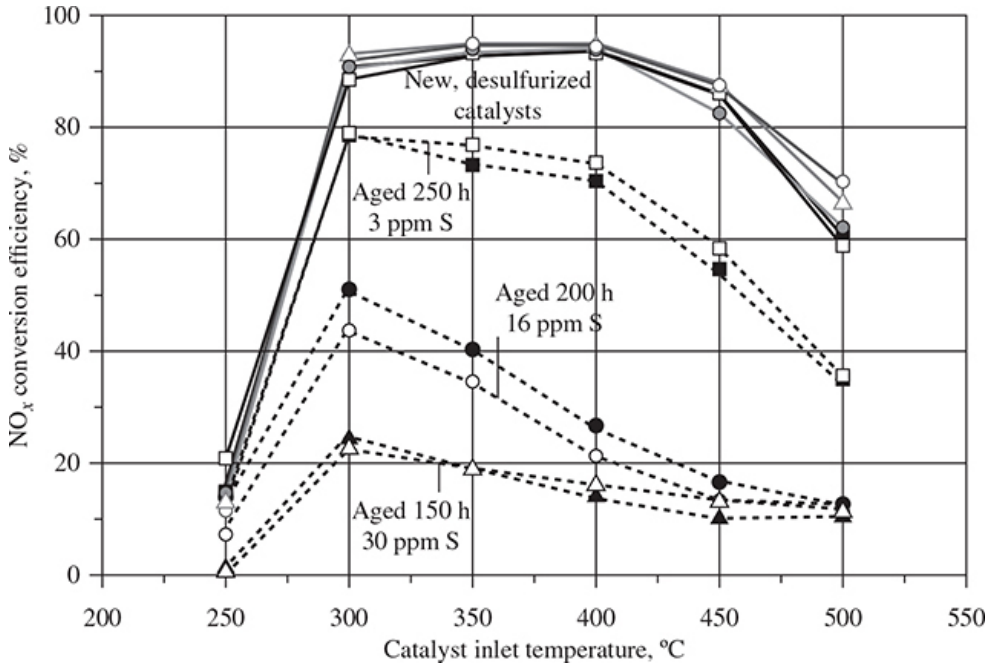


Figure 11.65 Conversion efficiencies for two lean NO_x storage catalysts as a function of temperature: curves are for new catalysts, aged catalysts (with 3 to 30 ppm S fuels), and after desulfurization.[86] Two hundred hours of aging corresponds to about 13,000 km of driving.

Selective Catalytic NO_x Reduction

The established NO_x catalytic reduction approach for diesels in passenger cars and for most heavy-duty and off-road applications is selective catalytic reduction (SCR). Urea is generally the source of the ammonia (NH_3) used as the reducing agent in this catalyst system. SCR catalysts have a higher efficiency than LNT technology, though ongoing developments may change that ranking.

The overall system is complex and a typical layout is shown in Fig. 11.66.[87] The exhaust first passes through an oxidation catalyst (V) where the NO is largely oxidized to NO_2. A high NO_2/NO_x ratio entering the SCR catalyst (S) is important. The HC and CO in the exhaust as are also oxidized to CO_2 and H_2O. Urea $(NH_2)_2CO$ is injected into and mixes with the exhaust gases, and that mixture then passes through a hydrolysis catalyst (H)—the first part of the SCR system—where water in the exhaust reacts with the urea releasing ammonia (NH_3). The figure shows the overall chemical reactions that occur

in each stage of the total process. The ammonia is the reducing agent in the SCR catalyst (S), which converts the NO_2 and NO in the mixture exiting the oxidation catalyst to N_2. The amount of NH_3 supplied via control of the urea flow is adjusted to provide a modest excess (up to 20%) above the amount required for the NO_2 and NO to be reduced to N_2. A final oxidation catalyst (O) then removes any excess NH_3 by oxidizing it to N_2.

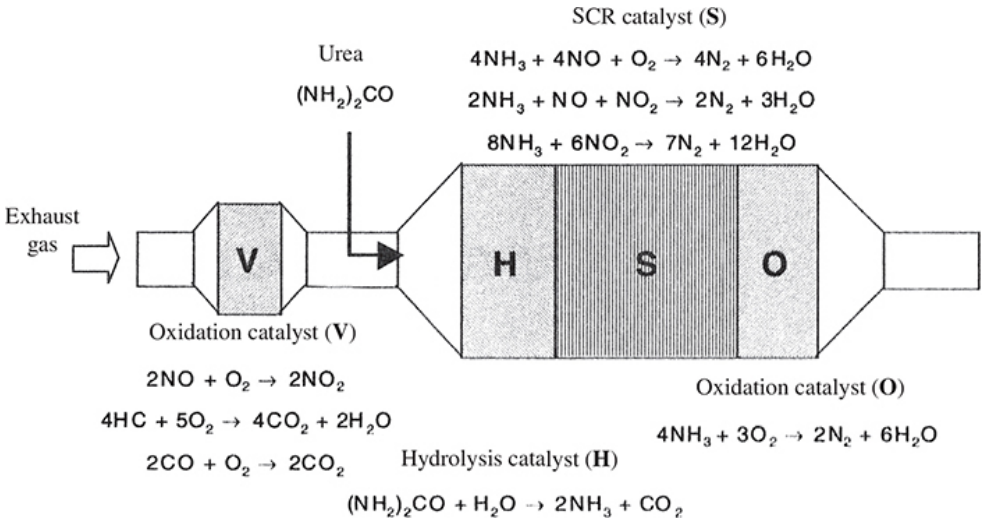


Figure 11.66 Layout of multi-component selective catalyst reduction system for NO_x reduction in a lean-operating diesel engine: oxidation catalyst (V); urea hydrolysis catalyst (H); SCR catalyst (S); oxidation catalyst (O). The overall chemical reactions are shown. [87]

The overall NO_x conversion efficiency is high over a specific operating temperature window that depends on the catalyst used, as is illustrated in Fig. 11.67. Platinum catalyst technology can only be used at temperatures below about 250°C due to its poor selectivity for NO_x reduction at higher temperatures. A vanadium catalyst with a titanium oxide carrier shows good efficiency up to above 500°C. These catalyst materials are resistant to sulfur and HC poisoning. Copper zeolite catalysts have been developed to the point where they show high efficiency (>95%) over a wide temperature range up to about 650°C. Iron zeolites are also used as an SCR catalyst, as are copper/iron combination zeolites. The results in Fig. 11.67 are illustrative, and development of this SCR catalyst technology is expanding their high-efficiency temperature windows.

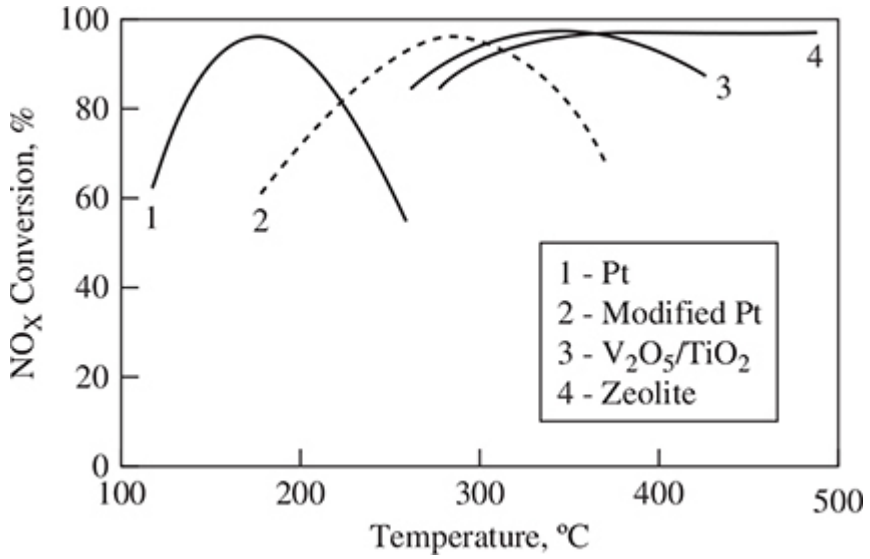


Figure 11.67 Operating temperature windows for different SCR catalysts: NO_x conversion efficiencies as a function of catalyst temperature.[79]

Other factors of importance are the impact of the NH_3: NO_x ratio, the fraction of the NO_x that is NO_2, ammonia "slip" through the SCR system, the production of N_2O (a greenhouse gas of significance) and poisoning by sulfur and hydrocarbons. At temperatures above 300°C the NO_x conversion efficiency increases linearly with increasing NH_3: NO_x molar ratio, as expected, for the stoichiometric reaction between NO and NH_3. At lower temperatures, above a critical value of this ratio, the conversion efficiency increases little, and ammonia slip then occurs. Also, to achieve a high SCR catalyst deNO_x efficiency, it is desirable to over-inject urea. The excess NH_3 then needs to be oxidized to N_2 to prevent slip. Ammonia slip catalysts may be needed to convert this ammonia to N_2 (rather than NO_x and N_2O).

The effect of increasing NO_2 as a fraction of total NO_x depends on SCR catalyst temperature and space velocity. Increasing the NO_2/NO_x molar ratio to about 0.5 increases the low-temperature NO_x conversion activity and efficiency. At higher ratios, the high-temperature activity and the achievable NO_x conversion efficiency are significantly reduced. The higher NH_3 stoichiometry needed to convert NO_2 (1.33) compared to converting NO (1.0), and the slower reaction of NO_2 with NH_3 results in ammonia slip

which must then be controlled. [87] A Pt upstream oxidation catalyst can increase the NO_2/NO_x ratio in the exhaust (which is typically 10 to 15% in diesel exhaust, as shown in Fig. 11.16) to over 80%, depending on the catalyst temperature.

Vanadia-based SCR catalysts are more tolerant to sulfur poisoning, and are used in world regions where fuel sulfur levels are higher. The sulfur accumulates in the top layer of the catalyst washcoat as ammonium sulfate. When heated to 500°C the SCR catalyst performance is recovered. Hydrocarbons can be absorbed on the catalyst surface at lower temperatures and then ignite at higher catalyst temperatures and, through the energy that releases, cause thermal deterioration of the catalyst surface. [88]

11.6.4 Particulate Filters or Traps

An exhaust treatment technology that substantially reduces engine particulate emissions is particulate filters or traps. A temperature-tolerant filter or trap removes the particulate material from the exhaust gas: the filter is then "cleaned off" by oxidizing the accumulated particulates—the regeneration process. Diesel particulate filters (DPF) are in common use. The technology can also be used on gasoline engines and may be needed on direct-injection gasoline engines since their particulate emissions are significantly higher than equivalent port-fuel-injected gasoline engine emissions, though lower than diesel engine-out levels.

The challenges to implementing particulate filter technology are many: (1) the filter, even when clean, increases the pressure in the engine exhaust manifold; (2) this pressure increase steadily rises as the filter collects particulate matter; (3) under normal diesel engine operating conditions, the collected particulate matter is not hot enough to spontaneously ignite and oxidize; and (4) once ignition of the filtered particulate matter occurs, the burn-up process must be controlled to prevent excessively high temperatures and trap damage or destruction. Particulate filters are now in production for light-duty automobile diesel engines as well as heavy-duty diesel engines.

Types of particulate filters include: wall-flow ceramic monoliths, sintered metal on stainless steel wire mesh, ceramic fiber woven (wound and knitted) silica. Each of these has different inherent pressure drop and filtering efficiency. [79] The majority of filters are ceramic wall-flow or fiber based,

and most of these are wall-flow filters of the type shown in Fig. 11.68. Such wall-flow filters are based on monolith catalyst substrates where half the cells are plugged at the inlet end and the other half at the outlet end. Thus the exhaust gas is forced to flow through the porous ceramic walls. The thickness of these walls and their porosity are carefully controlled to provide good particulate filtration with a low enough pressure drop. The particle separation is effected by depositing the particles in the exhaust gases onto the filter surface. This generally occurs initially within the porous ceramic wall as various force fields hold the particles on the surfaces of the pores (depth filtration). Then, as the amount of filtered particulate builds up, a layer of particulate may collect on the channel wall surface which then acts as a sieve, further filtering out the particulate ("cake" filtration).

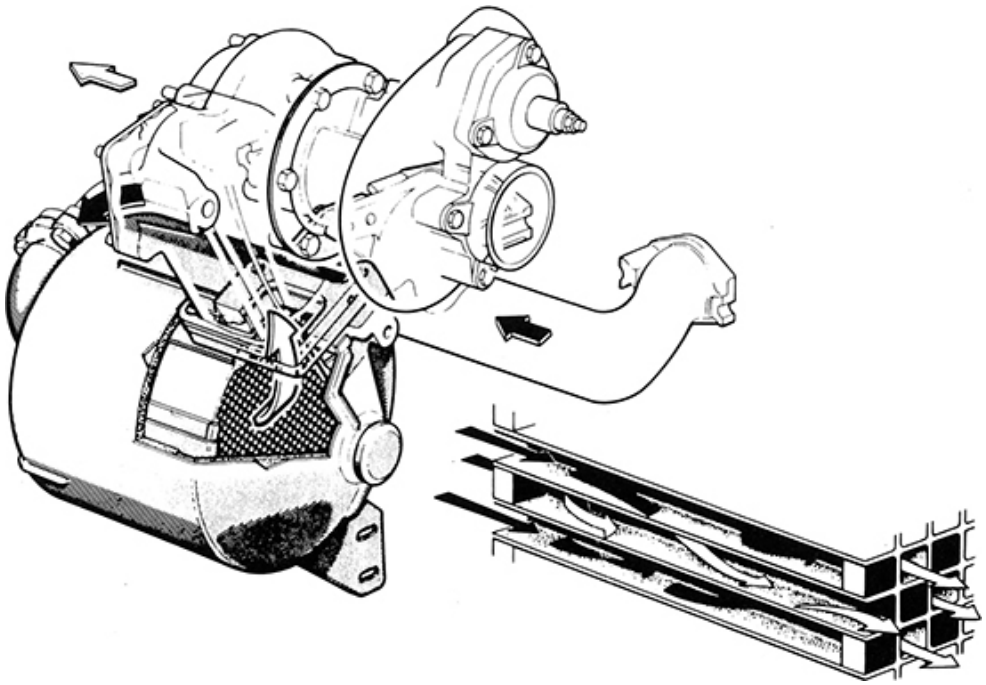


Figure 11.68 Ceramic monolith particulate filter mounted in the exhaust of a turbocharged automobile diesel engine.[89]

Due to the low packing density of diesel particulate (about 0.1 g/cm ^{3}) such filters quickly accumulate considerable volumes of soot. The pressure drop across the DPF steadily increases as particulate matter accumulates, and to avoid excessively high pressures in the engine's exhaust manifold the filter must be regenerated. There are several approaches to filter

regeneration. Thermal regeneration is the primary method: the solid particulates largely (soot) in the DPF are oxidized to CO_2 (and some H_2O). The rate of oxidation of soot is significantly lower than oxidation of gaseous emissions such as HC and CO: the reason that solid particulates are not oxidized in flow-through oxidation catalysts designed to remove gaseous pollutants is that they provide relatively short residence times (see Sec. 11.6.3). Diesel soot oxidation rates (under controlled conditions) are shown as a function of temperature in Fig. 11.69: temperatures above 500°C are required in the absence of catalysts. The temperature of diesel exhaust gas is normally well below this level. So means for raising the exhaust gas temperature, or use of a catalyst to reduce the activation energy and achieve adequate oxidation rates at lower temperature, must be employed. The former is categorized as active regeneration and the latter as passive.

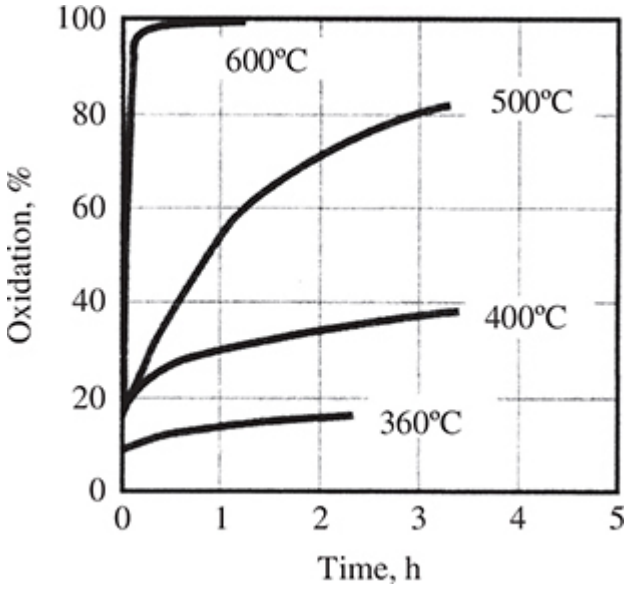


Figure 11.69 Diesel soot oxidation rates, as a function of temperature.[79]

Combinations of active and passive approaches have been explored and are being deployed. One such combination approach to regeneration is illustrated in Fig. 11.70 which shows a particulate filter regeneration process lasting about 200 s. The DPF system in this light-duty diesel vehicle consists of a filter (like that shown in Fig. 11.68) with a DOC in the exhaust system ahead of the DPF. The engine's common- rail fuel-injection system can provide three injection pulses: a pilot or pre-injection (to achieve a shorter

ignition delay), a main injection (primarily responsible for generating bmep/torque), and a late or after injection well into the expansion process primarily to raise the exhaust gas temperature to initiate the particulate filter regeneration process. Figure 11.70 shows the start of regeneration at 25 seconds where late fuel injection occurs and the energy release from that fuel burning starts to raise the exhaust temperature above about 200°C. The DOC through additional oxidation of this third injected fuel pulse further raises the exhaust gas temperature downstream of the DOC and upstream of the filter. At 175 seconds, the soot in the filter starts oxidizing and releasing its chemical energy causing the filter exit gas temperature to rise sharply. As the soot within the DPF burns (following the start of regeneration), the filter pressure drop falls (bottom curve).

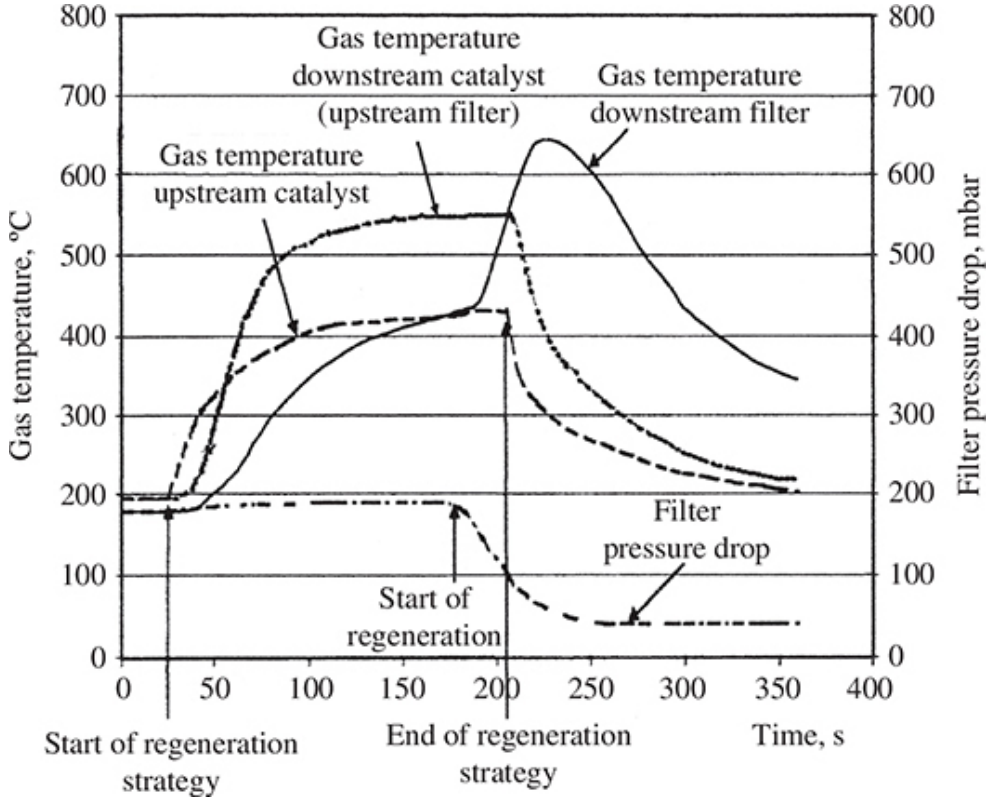


Figure 11.70 Example of exhaust gas temperature variation during particulate filter regeneration, passenger car diesel engine, under steady-state conditions.[90] Late injection increases exhaust gas temperature, lighting off the catalyst: soot burns and back pressure drops. Cerium, added to the fuel, is used to drop the regeneration temperature, catalytically.

In this DPF system, a cerium-based fuel additive is injected into the fuel tank in modest amounts: the ceria acts as a catalyst in the soot oxidation process within the filter allowing regeneration initiation to occur at about 400°C rather than at the higher temperatures (some 600°C; see Fig. 11.69) needed in the absence of a catalyst. Metal-based fuel additives, however, must be controlled for many reasons: stable catalytic activity, cost, environmental and health impacts, degradation of catalytic activity in the exhaust treatment system. A more permanent alternative to catalytic fuel additives is to build the catalyst into the filter surface, creating a more "passive" catalyzed particulate filter. These usually utilize platinum or base-metal catalyst materials. As in the DPF example described above, a thermal regeneration strategy is often employed as well to raise the exhaust gas temperature to the level at which the catalyzed filter initiates the soot oxidation process to achieve the required degree of control of trap or filter regeneration process. [79]

An alternative to using oxygen to oxidize the soot is to use NO_2 in a platinum-catalyzed DPF. The NO_2 is generated through use of an upstream oxidation catalyst from the nitric oxide (NO) which constitutes most (some 90%) of the NO_x in diesel exhaust gases. The NO_2 fraction, with an appropriate oxidation catalyst, can be increased to about three-quarters of the NO_x at temperatures of about 300°C. NO_2 is a sufficiently active soot oxidizing agent that the filter need not be catalyzed. However, the NO_2 is reduced to NO and not to N_2, so a deNO_x catalyst would have to follow the DPF in the total exhaust treatment system.

Control of the regeneration process is critical to the durability of filter systems. Whatever process is used for regeneration must be reliable and robust. The soot holding capacity of filter systems is typically several hours. While some DPF systems are designed to continuously regenerate (e.g., a catalyzed filter in an application where exhaust gas temperatures are relatively high), an active regeneration process once a day, or more often, may be the needed approach. The filtration efficiency ^{n} of DPFs (based on particulate mass) is 95 to close to 100% for soot (predominantly elemental carbon) and 50 to 90% for the absorbed SOF with the lower values for non-catalyzed filters. The overall efficiency for total PM is in the 70 to 95% range. Ash from the inorganic material in the lubricating oil and fuel is also trapped in the filter and is not burned-off in the regeneration process. Thus

ash accumulates in the filter incrementally over time, causing a gradual increase in the filter pressure drop. Periodic "cleaning" of the filter of ash, every year or so, is required through washing with water or reverse gas flow through the filter walls to "unblock" the pores. [79]

Controlling the particulate and filter temperature during regeneration is a critical challenge. To start the regeneration process, the temperature must be high enough to initiate the soot oxidation process. As explained above, for non-catalyzed filters this temperature is higher; for catalyzed filters it is lower. The PM oxidation process converts its chemical energy (soot, SOF, HCs) to thermal energy (these are exothermic reactions), which heats up the exhaust gases, the particulate matter, and the filter monolith structure. The change in within-filter temperatures depends on the rate of PM burning (which is strongly dependent on the exhaust gas and soot temperature), and the partitioning of the thermal energy thereby released between heating the exhaust gases, and the soot and the filter material. Thus, as shown in Fig. 11.70, the gas temperature downstream of the filter rises as the PM oxidation rate increases, reaches a maximum and then falls, as the soot is increasingly consumed and the rate of energy release falls. The filter pressure loss curve at the bottom of the figure indicates what the rate of PM burn-up actually is: it first increases and then decreases. Peak temperatures within the ceramic filter structure during regeneration must be limited to below about 1200°C due to ash sintering and reactions of the cordierite with the ash in the filter. At higher temperatures the cordierite itself may sinter and melt.

The high particulate trapping efficiencies of wall-flow DPFs have as a consequence a high-pressure drop. And, as the trapped PM builds up, that pressure drop increases substantially. The pressure drop has several components: the loss of pressure at the entrance and exit of the filter monolith due to the significant contraction and expansion that occurs; the pressure loss due to channel-wall flow friction; the pressure drop across the channel walls; and, of course, the additional channel wall pressure drop due to the steadily increasing particulate loading. The increase in pressure drop above that of a clean wall-flow filter initially rises rapidly as particulate is deposited in the ceramic wall pores, then rises steadily over time as particulate builds up as a layer on the wall surface. This building soot deposit layer constitutes the largest part of the pressure drop. The pressure drop across the channel walls with a clean filter is in the 1 to 2 kPa range; a moderately loaded filter has a pressure drop of some 10 kPa. [o] This pressure

drop can be modeled quite accurately, and used as a signal that defines the extent of the DPFs particulate loading to trigger trap regeneration.

Particulate filters are now a practical and effective control technology that significantly reduces particulate emissions. In diesel engines almost two orders of magnitude reduction in particulate number density emissions can be achieved using a trap. The most important issues with such exhaust treatment devices are reliable filter regeneration with minimum impact on engine operation (especially any fuel consumption penalty), and their cost. Particulate filters could also be used effectively with DI gasoline engines (especially if these are stratified DI SI engines), but again, the additional cost of using this particulate emissions control technology, relative to in-cylinder particulate reduction is a critical factor.

11.6.5 Exhaust Treatment Systems

These specific catalyst and particulate trap devices need to be combined into an exhaust aftertreatment system since there are several different pollutants in engine exhaust gases whose concentrations need to be substantially reduced. In spark-ignition (gasoline) engines, the three main strictly-regulated air pollutants (especially in vehicles) are CO, hydrocarbons, and NO_x. To these have been added fine particulates which, while low in port-fuel-injection SI engines, are much higher in direct-injection gasoline engines. In diesel engines, the primary engine-out emissions needing major reduction are particulates and NO_x. Note also that the exhaust gas composition (relative *A/F* or fuel-air equivalence ratio) and exhaust gas temperature ranges for these two types of engines are substantially different. So we discuss each type of engine in turn.

Gasoline SI Engines

The predominant aftertreatment system incorporates three-way catalyst technology with stoichiometric engine operation, and oxygen (λ) exhaust-gas sensors for closed-loop fuel feedback control (see Sec. 7.8). Figure 11.71 illustrates, with schematics, the historical development of exhaust aftertreatment systems. First, Fig. 11.71 *a* shows an early 1970s system with an oxidation catalyst to remove CO and HC. An option was to pump secondary air into the exhaust port to ensure sufficient oxygen was available when the engine operated fuel-rich to achieve complete CO and HC burn up.

Then came three-way catalysts with a lambda sensor in the exhaust manifold to control the engine's relative air/fuel ratio to close to stoichiometric. For effective removal of the three pollutants CO, HC, and NO_x together, ceria was built into the TWC washcoat to ensure sufficient oxygen storage so that the gas at the catalyst active surface sites was always at stoichiometric despite air-fuel ratio fluctuations within the engine itself.

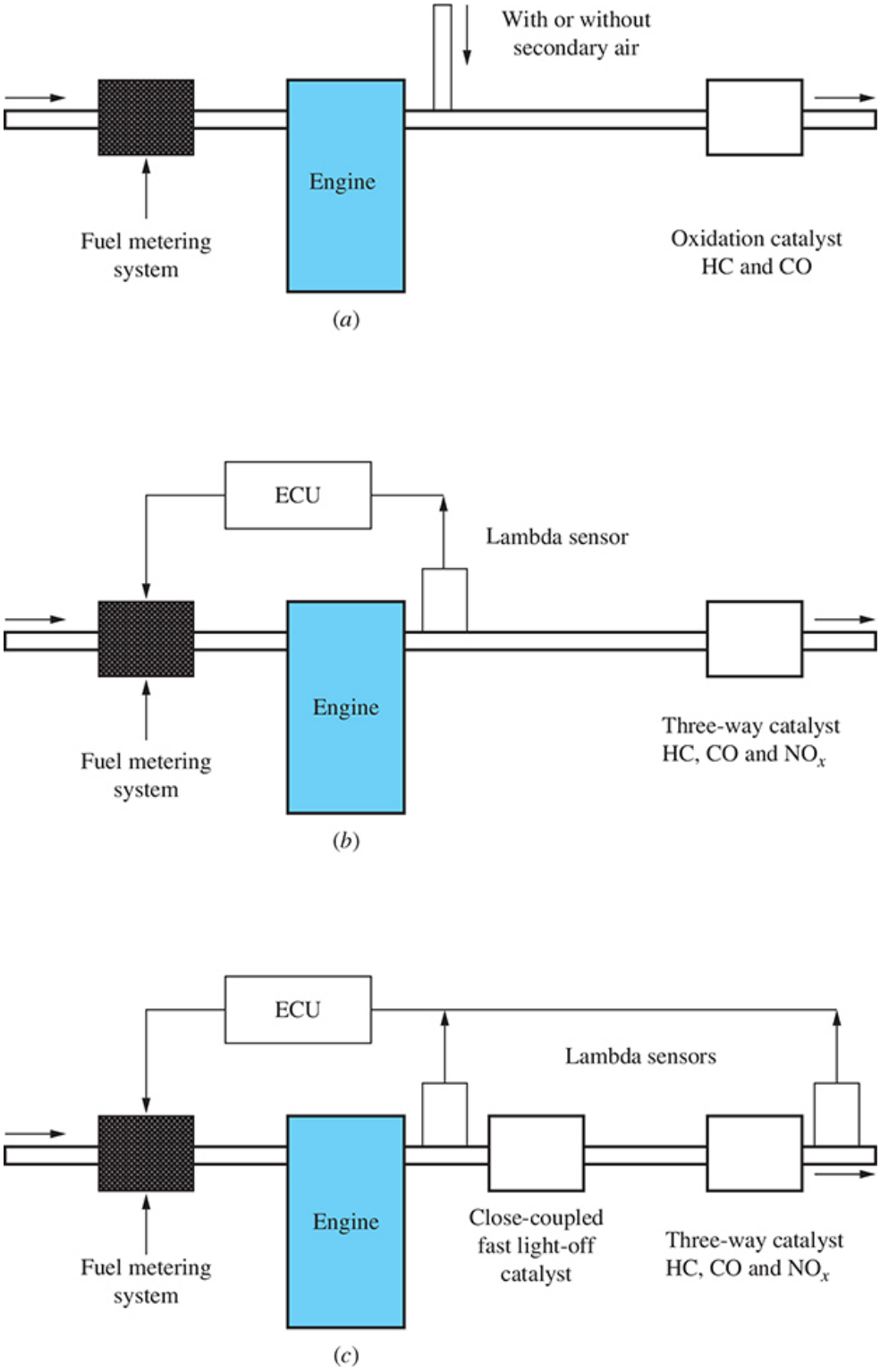


Figure 11.71 Development of gasoline SI engine exhaust aftertreatment systems: (*a*) oxidation catalyst for CO and HC removal, late 1970s; (*b*) three-way catalyst system with exhaust oxygen sensor and fuel metering

feedback control, 1980s; (*c*) close-coupled fast light-off catalyst (largely for cold engine HC control) and under-floor three-way catalyst for further HC, CO, and NO_x reduction, with close-mounted and downstream exhaust oxygen (λ) sensors for feedback engine control, and as a catalyst oxygen storage diagnostic.

Figure 11.71 *c* shows a more complex system with a close-coupled fast-light-off version of the three-way catalyst designed to heat up quickly following a cold start,. To limit HC emissions during starting and early warm up (see Fig. 11.33). Then a larger TWC (under the floor of a light-duty vehicle) completes the catalyst's HC, CO, NO_x clean up task. Some systems now use a larger, single, close-coupled fast-light-off three-way catalyst. As noted previously, very high catalyst efficiencies (about 98%) are now achieved with such systems. The two lambda sensors check whether the oxygen storage capability in the TWC is operating effectively. The upstream sensor follows the low-frequency (1-Hz) lambda oscillations (fuel-rich, then lean) that the engine fuel control system implements to maintain oxygen storage capacity so oxygen can be released or absorbed at the catalyst surface. The downstream sensor gives a steady non-oscillating signal when oxygen release or storage within the catalyst maintains the exhaust gases at stoichiometric.

Rapid light off of the close coupled catalyst during the early stages of engine warm up is usually achieved through low thermal inertia design, and use of substantial spark (i.e., combustion) retard for some 10 to 100 seconds following engine start up. Use of a fuel-rich mixture during engine starting, with secondary air added to the exhaust gases, with this spark retard, can be used to achieve a hotter exhaust through burning up the extra CO and HC emissions in the exhaust gases, in the exhaust port and manifold, prior to the catalyst location (see pp. 647 to 649).

With turbocharged SI engines, the turbine is normally mounted close to the exhaust manifold exit to minimize loss of exhaust gas enthalpy prior to turbine entry, and shorten the turbocharger response time to increases in engine load. The insertion of the turbine into the exhaust system upstream of the close-coupled catalytic converter can lengthen the catalyst light-off time and thus worsen emissions. However, the turbine can withstand higher exhaust gas temperatures than can the three-way catalyst, so the negative

impact can be partly offset by permitting higher exhaust gas enthalpy fluxes into the exhaust system.

Diesel Engines

Diesels always operate fuel lean: thus their exhaust gas temperatures are significantly lower than in gasoline SI engines. As a consequence of these factors, and that the primary pollutants of concern are NO_x (a gas which has to be reduced to N_2) and particulate matter (small particles with absorbed condensed hydrocarbons), the aftertreatment systems are more complex and require periodic regeneration.

The task of significantly reducing these two major diesel-engine air pollutants is spelled out in Fig. 11.72 where average test-cycle particulate emissions are plotted as a function of NO_x emissions, for a heavy-duty diesel engine. Implementation of engine emission control technologies (optimized combustion, low sulfur fuels, and EGR) reduces engine-out emissions to about 0.04 g/kW · h particulates and 4 g/kW · h NO_x. The emissions requirements in many parts of the world require additional reductions by a factor of 5 to 10 or so. Thus use of particulate traps and NO_x catalysts is essential.

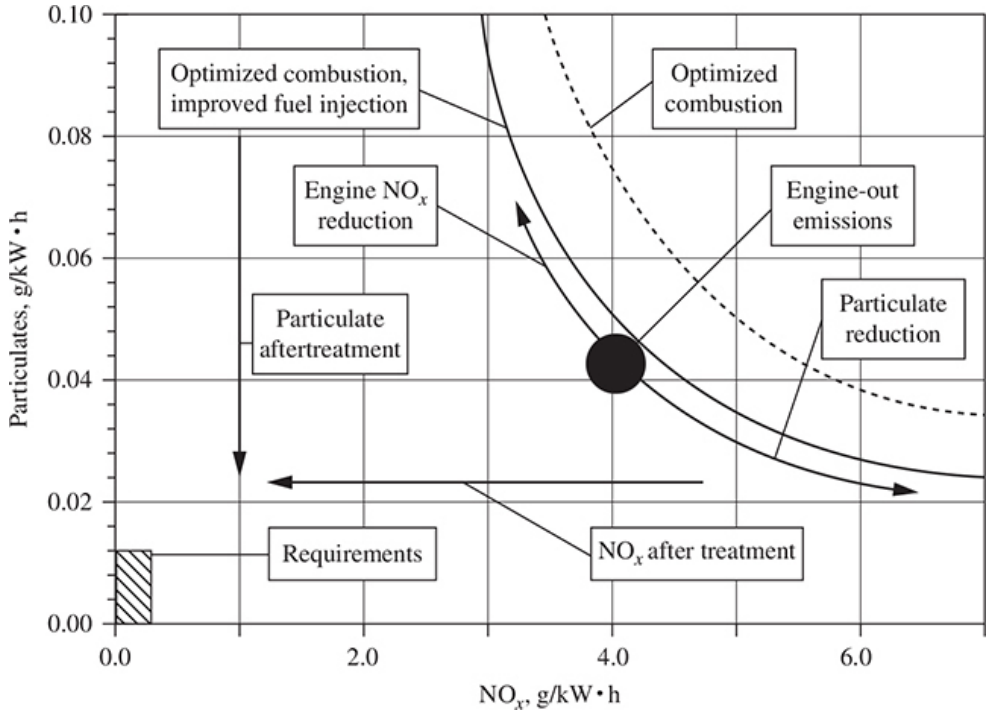


Figure 11.72 Trade-off curves between particulates and NO_x emissions for a typical heavy-duty diesel engine (average test cycle values in g/kWh). Dark circle indicates typical engine-out levels with engine emission controls. Cross-hatched low PM and NO_x region indicates levels required to meet emissions regulations in the United States and Europe. Source: Ref. 87.

Diesel engines now require particulate traps which must be regenerated—the filtered particulate must be burned up. This can be done by raising the engine's exhaust gas temperature to initiate trap regeneration by injecting additional fuel late in the expansion stroke (see Fig. 11.70, and accompanying text) to raise the exhaust gas and trapped particulate temperatures so the particulate matter is burned up. (In the example shown, a fuel additive, ceria, was used to catalyze the particulate oxidation reactions.) Or the particulate filter surface may have embedded catalyst sites.

In both heavy-duty and many light-duty diesel engines, the principal NO_x aftertreatment control strategy is selective catalytic reduction (SCR), which adds urea, the source of the reducing agent ammonia, to the exhaust gases. Figure 11.73 shows an example of an aftertreatment system that reduces HC, CO, particulates, and NO_x emissions for a heavy-duty engine. The continuously regenerating diesel particulate filter (CRDPF) comprises an oxidation catalyst followed by a wall-flow particulate filter. The oxidation catalyst oxidizes the engine-out HC and CO, and a portion of the NO to NO_2. NO_2 oxidizes the particulate in the filter at a much lower temperature (250°C). This lower temperature particulate oxidation by NO_2 allows passive and continuous particulate removal from the trap/filter surface. [91]

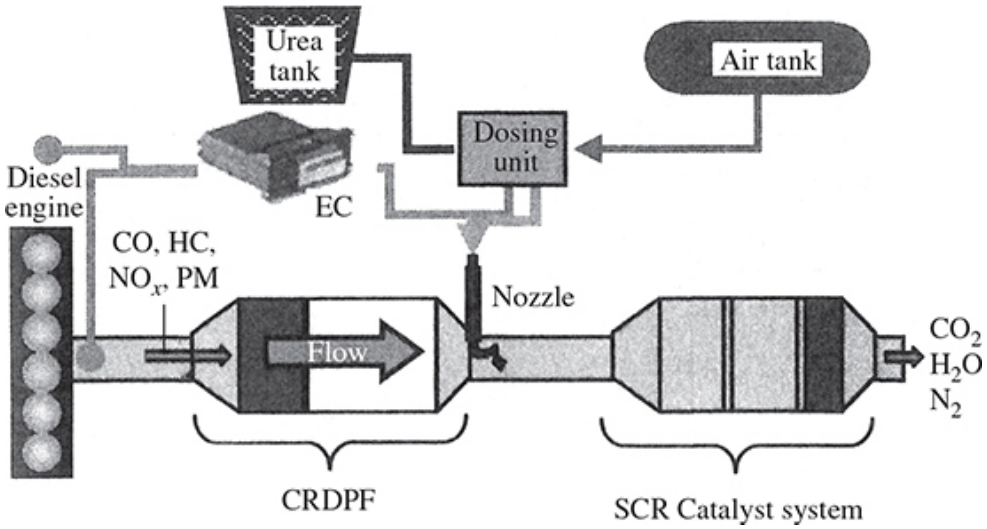


Figure 11.73 Layout of engine aftertreatment system embodying a continuously regenerating diesel particulate filter (CRDPF) and selective catalytic reduction (SCR) system, where the urea dispensing system shown provides the NO_x reducing agent, ammonia.[91]

The SCR system uses NH_3 derived from a urea solution via rapid hydrolysis in the "hydrolysis catalyst" section of the SCR device: see Fig. 11.66, and accompanying text. [87] Successful implementation requires matching the amount of urea injected to the amount of NO_x emitted from the engine. Two approaches are used: (1) a urea injection map based on engine operating information (essentially a map-based estimation of engine-out NO_x emissions) or (2) an NO_x sensor to give real time measurements of exhaust gas NO_x concentrations. The middle section of the SCR catalyst does the reduction of NO and NO_2 to N_2 with ammonia as the reducing agent. The final section (shown dark gray in Fig. 11.73) is an oxidation catalyst that oxidizes any excess NH_3 to N_2 (see Fig. 11.66). The layout of the urea tank, dosing control unit, air tank to provide the air stream that transports the urea through the injection nozzle into the engine's exhaust downstream of the CRDPF unit are also shown.

An alternative diesel exhaust aftertreatment approach is shown in Fig. 11.74 where a LNT is used for NO_x reduction in the diesel's fuel-lean exhaust gases. [92] The operation of the LNT is illustrated in Fig. 11.63 and discussed in the accompanying text. A catalyzed particulate filter, preceded

by a DOC, is the first aftertreatment system device, followed by the LNT. The oxidation catalyst converts some of the NO to NO_2: the NO_2 is more effective at trap regeneration since it oxidizes the particulate (soot) at lower temperatures than does oxygen. Details of the engine enrichment strategy to regenerate the LNT, desulfurization challenges because of the difficulty in attaining high enough temperatures due to the significant thermal capacity in the catalyzed particulate filter upstream of the LNT, and the need to control component temperatures throughout the aftertreatment system, can be found in Ref. 92. An alternative to the layout shown is to place the LNT first, followed by the catalyzed particulate filter.

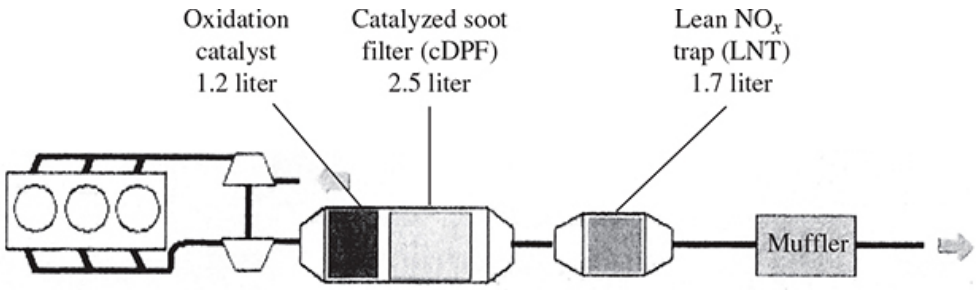


Figure 11.74 Layout of light-duty high-speed diesel aftertreatment system incorporating a catalyzed soot filter (cDPF) followed by a lean NO_x trap (LNT). The cDPF and the LNT both need to be periodically regenerated. [92]

Lean NO_x traps need to be regenerated (releasing the NO_2 and reducing the NO_x to N_2) about every 60 seconds: the duration of the rich regeneration phase is some 3 to 5 seconds. There is thus an engine fuel consumption penalty of several percent. The above approach is better suited to smaller high-speed DI diesels than to larger engines.

PROBLEMS

11.1 Figure 11.2 shows concentrations of NO, CO, and HC in a spark-ignition engine exhaust as a function of the relative air/fuel ratio. Assume the concentration scale is parts per million. Explain the trends shown as the mixture is first made richer and then leaner than stoichiometric.

11.2 Figure 11.2 is for a spark-ignition engine. Construct a similar qualitative graph of NO, CO, and HC concentrations versus equivalence ratio for a direct-injection four-stroke cycle diesel engine. Explain your graphs.

11.3 A spark-ignition engine driving a car uses, on average, 120 grams of gasoline per mile traveled. The average emissions from the engine (upstream of the catalyst) are 1.5, 2, and 20 grams per mile of NO_x (as NO_2), HC, and CO, respectively. The engine operates with a stoichiometric gasoline-air mixture. Find the average concentrations in parts per million of NO, HC (as ppm C_1), and CO in the engine exhaust.

11.4 Calculate the average combustion inefficiency corresponding to the spark-ignition engine engine-out emissions levels given in Prob. 11.3. Include any hydrogen you estimate would be present in the exhaust stream.

11.5 A three-way catalytic converter is used with the spark-ignition engine in Prob. 11.3. For 10% of the driving time, the catalyst is cold and ineffective, and does not reduce the engine's emissions. For 90% of the time, the catalyst is hot and has conversion efficiencies as given in Fig. 11.62. Estimate for a cold catalyst and a hot catalyst, the average *vehicle* emissions of NO_x, HC, and CO in grams per mile.

11.6 Figure 15.18 shows the variation in NO and HC emissions as concentrations (ppm) in the exhaust of a spark-ignition engine as a function of speed and load. Convert these data to graphs of indicated specific NO and HC emissions (g/kW · h) versus speed and imep. Assume η_v (based on atmospheric air density) = imep (kPa) $\times 10^{-3}$.

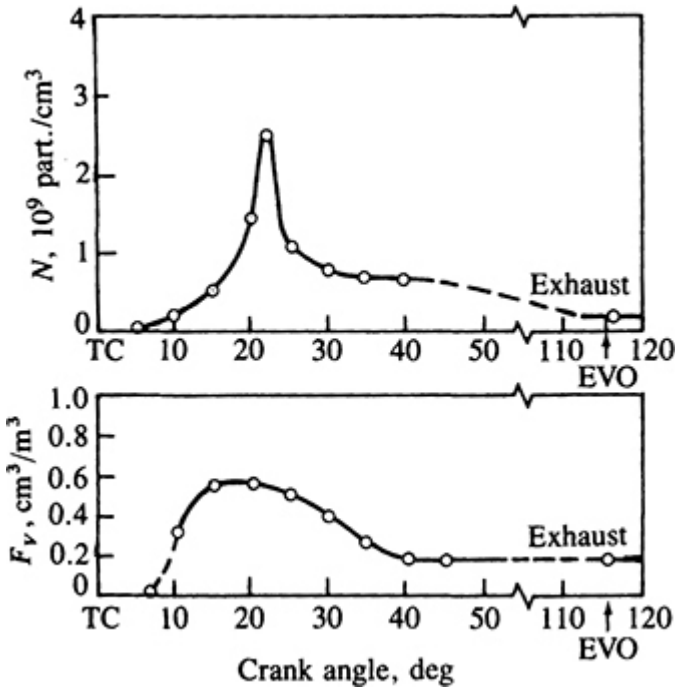
11.7 Use the data in the figure to estimate:

(a) The exhaust particulate emissions as a fraction of the maximum particulate loading during the cycle.

(b) The maximum measured soot loading and the exhaust soot loading as fractions of the fuel carbon.

(c) The equivalent sphere size of each soot particle at the number density peak (22° ATC) and in the exhaust.

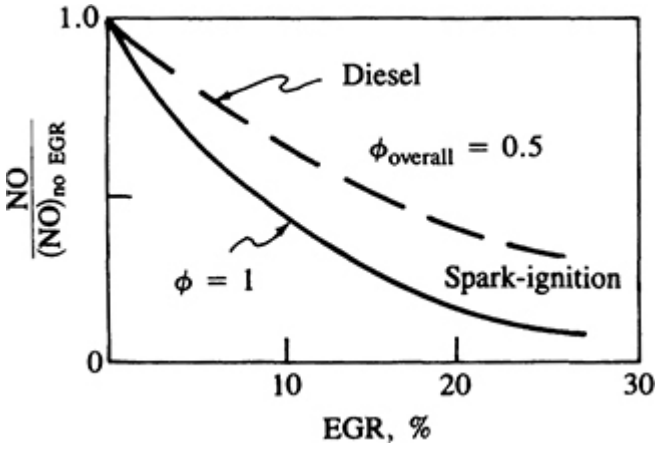
Assume a particulate density of 2 g/cm ^{3}. Note that the gas volumes in the figure are determined at standard temperature and pressure.



11.8 Explain the following emissions trends. Highest marks will be given for succinctsummaries of the *important* technical issues.

(a) Nitric oxide (NO) emissions from diesels and spark-ignition engines as the relative air/fuel ratio, or fuel/air equivalence ratio ϕ, is varied show significantly different behavior (see Figs. 11.8 and 11.16). Redraw these graphs on the same plot and explain the different trends for these two types of engines as ϕ decreases on the lean side of stoichiometric.

(b) Recirculation of a fraction of the exhaust gases to the intake is used to control engine nitric oxide emissions at part load. Exhaust gas recycle is usually more effective with spark-ignition engines than with diesels, as shown in the figure. Explain why these trends are different.



(c) Brake-specific particulate emissions from diesels are a major problem. Particulate emissions from port fuel injected spark-ignition engines are not that important. Briefly explain why the particulate emission levels from these two types of engines are so different in magnitude.

(d) Diesels have low carbon monoxide (CO) emissions. Spark-ignition engine CO emissions when *averaged* over a typical urban automobile trip (cold engine start, warm-up, cruise, idle, acceleration, etc.) are substantial and require an oxidation catalyst for effective control. Explain this difference in average CO emissions (upstream of any catalyst) from these two types of engines.

11.9 The following questions refer to an engine with these geometric and operating characteristics: $\phi = 1.0$, compression ratio = 8:1, bore = 85 mm, stroke = 85 mm, piston diameter above top ring = 84.4 mm, distance from piston crown top to top ring = 10 mm, volumetric efficiency = 0.8, temperature in cylinder at the start of compression = 333 K, pressure in cylinder at start of compression = 1 bar, mixture temperature before entering cylinder = 30°C, brake-specific fuel consumption = 300 g/kW · h.

A substantial fraction of spark-ignition engine hydrocarbon emissions comes from the crevice between the piston crown and cylinder wall. Gas is forced into this crevice as the cylinder pressure increases and flows out of this crevice as the cylinder pressure decreases. The gas in the crevice can be assumed to be at the wall temperature, 400 K. The gas pushed into the crevice ahead of the flame is unburned mixture; the gas pushed in behind the flame is burned mixture. About two-thirds of the crevice gas is unburned. The maximum cylinder pressure is 3 MPa (see sketch below).

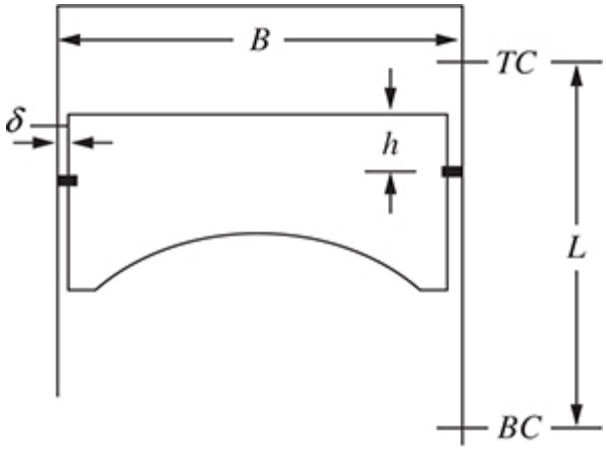
(a) Derive an expression for the fraction of the cylinder charge which is contained in this piston crevice region at crank angle of maximum cylinder pressure ($\theta_{p\ \max}$) in terms of (δ/B), (h/L), the cylinder volume $p_{\max}$, V_{cyl} ($\theta_{p\ \max}$), the clearance volume V_c, T_{cyl} ($\theta_{p\ \max}$) the mean cylinder-gas temperature at $\theta_{p\ \max}$, and wall temperature T_w. Assume that the pressure in the crevice equals the pressure in the cylinder and that the temperature of the gas in the crevice equals the wall temperature T_w.

(b) Calculate the mass fraction of the cylinder gas which is in the crevice between the piston and cylinder wall and above the first piston ring, at the time of peak pressure when the average in-cylinder gas temperature is 2400 K.

(c) Assuming that half of the unburned fuel in this region is oxidized within the cylinder and a further one-third is oxidized in the exhaust port, calculate the engine-out HC emissions from this source in parts per million (ppm C_1) by volume.

(d) Estimate the ratio of brake-specific hydrocarbon emissions to brake-specific fuel consumption.

(e) The vehicle exhaust HC emissions with this engine are 0.1 g/mile; the gasoline consumption is 25 miles per gallon. The catalyst efficiency for HC is 0.96. What percentage of the engine HC emissions is from the crevice?



11.10 Nitric oxide, NO, forms via reactions (11.1) to (11.3). Reaction (11.1) is "slow" and reactions (11.2) and (11.3) are "fast," so the initial rate of formation of NO is given by Eq. (11.8) :

$$\frac{d[\mathrm{NO}]}{dt} = 2k_1^+[\mathrm{N}_2]_e[\mathrm{O}]_e$$

where [ ] denote concentrations in gram-moles per cubic centimeter, k_1^+ is the rate constant for reaction (11.1), and the factor of 2 enters because the N atom formed in (11.1) immediately reacts via (11.2) or (11.3) to give an additional NO molecule:

$$k_1^+ = 7.6 \times 10^{13} \exp\left(\frac{-38{,}000}{T}\right) \mathrm{cm}^3/\mathrm{gmol}\cdot\mathrm{s}$$

where T is in kelvins.

Using the equilibrium composition data provided for mole fraction atomic oxygen (O), molecular nitrogen (N_2), and nitric oxide (NO):

(a) Plot the formation rate of NO as a function of the equivalence ratio at 3000 K and 5.5 MPa, and as a function of temperature for a stoichiometric mixture at 5.5 MPa pressure.

(b) Estimate approximately the time taken to reach equilibrium NO levels at $\phi = 1$, 2750 K and 3000 K, 5.5 MPa pressure.

(c) If the stoichiometric mixture inducted into the engine reaches 3000 K and 5.5 MPa after combustion, in the absence of any exhaust gas recirculation, estimate the percentage of the exhaust that must be recycled to the intake (at the initial intake temperature) to reduce the NO formation rate by a factor of 4 (assume the final pressure 5.5 MPa stays the same; of course, the final temperature decreases as the exhaust gas is recycled).

$p = 5.5$ MPa

		Mole fraction	
ϕ	T (K)	O	N_2
0.9	3000	2.1×10^{-3}	0.73
1.0	3000	1.5×10^{-3}	0.73
1.11	3000	1×10^{-3}	0.73

$\phi = 1.0, p = 5.5$ MPa

	Mole fraction		
T (K)	O	N_2	NO
2500	6×10^{-5}	0.73	–
2750	5×10^{-4}	0.73	4×10^{-3}
3000	1.5×10^{-3}	0.73	8×10^{-3}

11.11 The following data relate to a 2-liter gasoline engine operating at stoichiometric conditions at wide-open throttle and 2500 rev/min.

$$\dot{m}_{air} = 42 \text{ g/s}; \quad \dot{m}_{fuel} = 2.8 \text{ g/s}$$

and the exhaust gas composition was:

$$x_{CO_2} = 0.114; \; x_{CO} = 0.0114; \; x_{H_2O} = 0.120; \; x_{H_2} = 0.0057; \; x_{O_2} = 0.0142; \; x_{CH_2} = 0.0038; \; x_{N_2} = 0.731$$

This exhaust then flows steadily through an oxidizing catalyst system where the species in the engine exhaust that are not fully oxidized then oxidize completely. The exhaust gas temperature entering the catalyst is 700 K. The extensive surface area of the catalyst maintains the local catalyst and gas temperatures essentially the same as the exhaust flows through the catalyst, and its remaining chemical energy is released. Show that there is enough oxygen for complete oxidation of the exhaust gases, and calculate the temperature of the exhaust gas as it leaves the catalyst, neglecting any heat losses to the outside environment. The specific heat, c_p, of the exhaust gas is 1.28 kJ/kg·K.

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[a] Percentages or parts per million (ppm) are molar fractions times 100 or 10^6, respectively. Emission Indexes are g pollutant per kg fuel.

[b] This is often called the extended Zeldovich mechanism[1]. Zeldovich was the first to suggest the importance of reactions (11.1) and (11.2). Lavoie et al. [2] added reaction (11.3) to the mechanism; it contributes significantly.

[c] The steady-state approximation is used when the rate of change in concentration of a chemical species is sufficiently fast that the reactant and product species can maintain a quasi-equilibrium.

[d] It is well known that the mixture composition within the cylinder is not completely uniform and varies modestly from one cycle to the next. Both these factors contribute to cycle-by-cycle combustion variations. For the present discussion, the assumption of mixture uniformity is adequate.

[e] Spark timing also affects NO emissions, as discussed next. The above discussion relates to engines run with spark timing at MBT or with torque at a fixed percentage below (and close to) the maximum.

[f] This is because the standard detection instrument, a flame ionization detector (FID), is effectively a carbon atom counter: for example, one propane molecule generates three times the response generated by one methane molecule. Some data in the literature are presented as ppm propane (C_3), or ppm hexane (C_6); to convert to ppm C_1 multiply by 3 or by 6, respectively. Older measurements of hydrocarbon emissions were made with nondispersive infrared (NDIR) detectors which had different sensitivities for the different hydrocarbon compounds. For gasoline-fueled engines, HC emissions determined by FID analyzers are about twice the levels determined by NDIR analyzers, [28] though this scaling is not exact.

[g] About half of these HC are fuel compounds; about half are partially reacted and fragmented fuel compounds.

[h] Note that exhaust valve leakage is a different type of "source" than the others. It allows HC to bypass the in-cylinder oxidation process; this source is normally small.

[i] Note that there is a significant variation in the temperature of the exhaust gases during the exhaust process. The gas exhausted first is about 100 K hotter than the gas exhausted at the end of the process (see Sec. 6.5).

[j] Note that under normal engine operating conditions, the diesel engine combustion inefficiency is small, about 1%; see Sec. 4.9.4 and Fig. 3.9.

[k] With 3000 ppm S in the diesel fuel, the sulfate contribution to total particulate was about 5%. At 100 ppm S in the fuel, the sulfate contribution to the total particulate was an order of magnitude smaller, 0.4%. [64]

[l] While this choice of volume is not strictly correct, since it does not account for cell density or catalyst loading, it is straightforward to apply and is widely used.

[m] The engine efficiency decreases as the air/fuel ratio is reduced from very lean values to rich of stoichiometric.

[n] Filtration efficiency is the mass ratio of particulate collected in the filter to particulate entering the filter.

[o] 1 atmosphere is close to 100 kPa (1 bar).

CHAPTER 12

Engine Heat Transfer

12.1 IMPORTANCE OF HEAT TRANSFER

Heat transfer is a parasitic process that removes from the engine's cylinder, as thermal energy, a significant fraction of the fuel's chemical energy. It results in a loss in efficiency. It is an inherent loss because the peak burned gas temperature in the cylinder of an internal combustion engine is of order 2500 K. Maximum metal temperatures for the inside of the combustion chamber space are limited to much lower values by a number of considerations, and cooling for the cylinder head, cylinder, and piston must therefore be provided. These conditions lead to heat fluxes to the chamber walls that can reach as high as 10 MW/m^2 during the combustion period. However, during other parts of the engine's operating cycle, the heat flux is close to zero. The flux varies substantially with location and is periodic: regions of the chamber that are contacted by rapidly moving high-temperature burned gases generally experience the highest fluxes. In regions of high heat flux, thermal stresses must be kept below levels that would cause fatigue cracking (so temperatures must be less than about 400°C for cast iron and 300°C for aluminum alloys). The gas-side surface of the cylinder liner must be kept below about 180°C to prevent deterioration of the lubricating oil film. Spark plug and valves must be kept cool enough to avoid knock and preignition problems, which can result from overheated spark plug electrodes or exhaust valves. Solving these engine heat-transfer problems is obviously a major design task.

Heat transfer affects engine performance, efficiency, and emissions. For a given mass of fuel within the cylinder, higher heat transfer to the combustion chamber walls will lower the average combustion gas temperature and pressure, and reduce the work per cycle transferred to the piston. Thus specific power and efficiency are affected by the magnitude of engine heat transfer. Heat transfer between the unburned charge and the chamber walls in spark-ignition engines affects the onset of knock which, by limiting the compression ratio, also influences power and efficiency. Especially critical is heat transfer from the hot exhaust valve and piston crown to the unburned mixture. Changes in gas temperature due to heat-transfer can impact the emission formation processes both within the engine's cylinder and in the exhaust system where afterburning of CO and HC occurs. The exhaust temperature also governs the power that can be obtained from exhaust energy driven devices such as a turbocharger turbine. The exhaust gas temperature has a significant impact on the effectiveness of emissions after treatment devices such as catalysts and particulate traps. Engine friction is both affected by engine heat transfer and contributes to the coolant load. The cylinder liner temperature governs the piston and ring lubricating oil film temperature, and hence its viscosity. Piston and liner distortion due to temperature nonuniformities have a significant impact on the piston-ring component of engine friction. Much of the mechanical energy dissipated due to friction must be rejected to the atmosphere via the cooling system. The fan and water pump power requirements are determined by the magnitude of the heat rejected. The importance of engine heat transfer is clear.

To examine heat transfer more fully, it is helpful to consider the engine's subsystems. The intake system consists of intake manifold and inlet ports and valves. Heat transfer to the inflowing charge reduces volumetric efficiency (see Sec. 6.2.1). However, higher intake port and valve surface temperatures aid in vaporizing the fuel. Within the engine cylinder, the temperature of the charge relative to the wall temperature, and the flow field, vary substantially throughout the cycle. Both of these have a major influence on heat transfer. During the intake process, the incoming charge is usually cooler than the walls and flow velocities are high. During compression the charge temperature rises above the wall temperatures, and gas velocities decrease (see Sec. 8.2.2). Heat transfer thus changes direction, and is now from the cylinder gases to the chamber walls. During combustion, gas temperatures increase to very high values, and the local gas expansion that occurs as the

fuel-air mixture burns increases gas motion. This is the period when heat-transfer rates to the walls are highest. Also, as the cylinder pressure rises, a small fraction of the cylinder charge is forced into crevice regions, resulting in additional heat losses (see Sec. 8.6). During expansion, gas temperatures decrease so heat-transfer rates decrease. When the exhaust valve opens, however, the blowdown process (Sec. 6.5) produces high velocities within the cylinder, and past the exhaust valve and in the exhaust port. Substantial heat transfer from the exhausting gases to the exhaust valve, port, and (to a lesser extent) manifold, occurs during the exhaust process. An ability to predict the magnitude of the heat transfer between the working fluid, the walls of the intake system, combustion chamber, and exhaust system, and to the coolant is of obvious importance to the engine designer. Sitkei provides a useful overview of engine heat transfer. [1]

12.2 MODES OF HEAT TRANSFER

The following modes of heat transfer are important.

12.2.1 Conduction

Heat is transferred by molecular motion, through solids and through fluids at rest, due to a temperature gradient. The heat transfer by conduction, per unit area per unit time, $\dot{Q}$, in a steady situation is given by Fourier's law:

$$\dot{\mathbf{q}} = k\nabla T \tag{12.1}$$

where k is the thermal conductivity. For a steady one-dimensional temperature variation

$$\dot{q} = \frac{\dot{Q}}{A} = -k\frac{dT}{dx}$$

Heat is transferred by conduction through the cylinder head, cylinder liner, and piston; through the piston rings to the cylinder wall; through the engine block and manifolds.

12.2.2 Convection

Heat is transferred through fluids in motion, and between a fluid and a solid surface. When the fluid motion is produced by forces other than gravity, the term *forced convection* is used. In engines, the fluid motions are turbulent (see Chap. 8).

Heat (more accurately, thermal energy) is transferred by forced convection between the in-cylinder gases and the cylinder head, valves, cylinder walls, and piston during the induction, compression, expansion, and exhaust processes. Heat is transferred by forced convection from the cylinder walls and head to the coolant (which may be liquid or gas), and from the piston when lubricant is used as a piston coolant. Substantial convective heat transfer occurs to the exhaust valve, exhaust port, and exhaust manifold during the exhaust process. Heat transfer by convection from the intake port and valve surfaces raises the temperature of the incoming charge. Heat is also transferred by convection from the engine block to the surrounding environment by convection.

In steady-flow forced-convection heat-transfer problems, the heat flux $\dot{Q}$ transferred to a solid surface at temperature T_w from a flowing fluid stream at temperature T is determined from the relation

$$\dot{q} = h_c(T - T_w) \tag{12.2}$$

where h_c is the heat-transfer coefficient. [a] For many flow geometries (such as flow through pipes or over a plate) h_c is given by relations of the form

$$\left(\frac{h_c L}{k}\right) = \text{constant} \times \left(\frac{\rho v L}{\mu}\right)^m \left(\frac{c_p \mu}{k}\right)^n \tag{12.3}$$

where L and v are a characteristic length and velocity of the fluid flow. The terms in brackets from left to right are the Nusselt, Reynolds, and Prandtl dimensionless numbers, respectively. For gases, the Prandtl number ($c_p \mu/k$) varies little and is about 0.7 (see Sec. 4.8). Values for m and n are typically 0.8 and 0.3.

When nucleate boiling occurs at the surface (i.e., vapor bubbles are formed in the liquid at a hot wall), as may be the case in high heat flux areas

on the coolant side in a water-cooled cylinder head, then h_c is increased and different relationships for h_c must be used.

12.2.3 Radiation

Heat exchange by radiation occurs through the emission and absorption of electromagnetic waves. The wavelengths at which significant thermal energy is transferred are in the visible range (0.4 to 0.7 μm) and the infrared (0.7 to 100 μm) . Heat transfer by radiation occurs from the high-temperature combustion gases and the flame region to the combustion chamber walls (although the magnitude of this radiation heat transfer relative to convective heat transfer is only significant in diesel engines). Heat transfer by radiation to the environment occurs from all the hot external surfaces of the engine.

The theory of radiant heat transfer starts from the concept of a "black body," that is, a body that has a surface that emits or absorbs equally well radiation of all wavelengths and that reflects none of the radiation falling on it. The heat flux from one plane black body at temperature T_1 to another at temperature T_2 parallel to it across a space containing no absorbing material is given by

$$\dot{q} = \sigma(T_1^4 - T_2^4) \tag{12.4}$$

where σ is the Stefan-Boltzmann constant 5.67×10^{-8} W/m$^2 \cdot$ K^4. Real surfaces are not "black" but reflect radiation to an extent, which depends on wavelength. Gases are far from this black-body idealization. They absorb and emit radiation almost exclusively within certain wavelength bands characteristic of each molecular species. These departures from black-body behavior are usually dealt with by applying a scaling factor (an *emissivity*, ε) to Eq. (12.4) . Similarly, a shape factor is applied to account for the fact that the angle of incidence of the radiation usually varies over any actual surface. These factors can be calculated for simple cases.

12.2.4 Overall Heat-Transfer Process

Figure 12.1 shows, schematically, the overall heat-transfer process driven by hot gases within the cylinder through the combustion chamber wall to the

coolant flow. The heat flux into the wall has in general both a convective and a radiation component. The heat flux is conducted through the wall, and then convected from the wall to the coolant. A schematic temperature profile, and mean gas and coolant temperatures, $\bar{T}_g$ and $\bar{T}_c$, are shown.

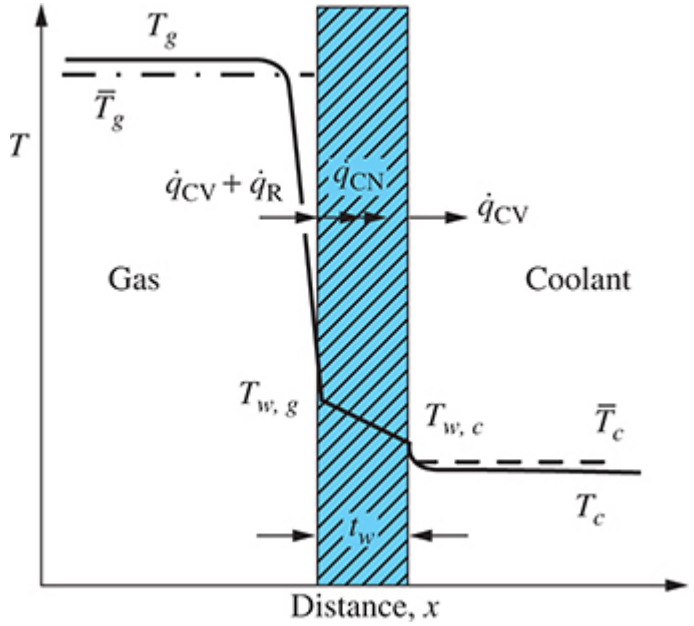


Figure 12.1 Schematic of temperature distribution and heat flow across the combustion chamber wall.

In internal combustion engines, throughout each engine operating cycle, the heat transfer takes place under conditions of varying gas pressure and temperature, and with local velocities which vary more or less rapidly depending on intake port and combustion chamber configuration (see Chap. 8). In addition, the surface area of the combustion chamber varies through the cycle. The heat flux into the containing walls changes continuously from a small negative value during the intake process to a positive value of order several megawatts per square meter during combustion, early in the expansion process. While the flux variation lags slightly behind the change in gas temperature, investigators have concluded that the assumption that the heat-transfer process is *quasi steady* is sufficiently accurate for most purposes. However, gas temperature and gas velocities vary significantly across the combustion chamber, so the heat flux distribution over the combustion chamber walls is nonuniform.

For a steady one-dimensional heat flow through a wall as indicated in Fig. 12.1, the following equations relate the heat flux $\dot{q} = \dot{Q}/A$ and the temperatures indicated:

$$\text{Gas side:} \qquad \dot{q} = \dot{q}_{CV} + \dot{q}_R = h_{c,g}(\bar{T}_g - T_{w,g}) + \sigma\varepsilon(\bar{T}_g^4 - \bar{T}_{w,g}^4) \tag{12.5}$$

where ε is the emissivity. The radiation term is usually negligible for SI engines.

$$\text{Wall:} \qquad \dot{q} = \dot{q}_{CN} = \frac{k(T_{w,g} - T_{w,c})}{t_w} \tag{12.6}$$

$$\text{Coolant side:} \qquad \dot{q} = \dot{q}_{CV} = h_{c,c}(T_{w,c} - \bar{T}_c) \tag{12.7}$$

If $h_{c,g}$ and $h_{c,c}$ are known, the temperatures $\bar{T}_g$, $T_{w,g}$, $T_{w,c}$, and $\bar{T}_c$ can be related to each other.

Equations (12.5) , (12.6), and (12.7) show that the heat flux per unit area for these different heat transfer modes scale with the temperature difference [e.g., between the hot fluid and wall, Eq. (12.5) , for gas-side convection, and hotter and cooler sides of the cylinder head for conduction]. The analogy with electrical circuit behavior is clear: The heat flux term ($\dot{q}$ or $\dot{Q} = \dot{q}A$) is effectively the electrical current, and the driving temperature difference is the voltage difference, V. Since $I = V/R$, these equations define the three thermal resistances involved in this sequence of heat transfers (illustrated in Fig. 12.1) as

$$R_{CV,g} = \frac{1}{h_{c,g}A}, \; R_{CN} = \frac{t_w}{kA}, \; R_{CV,c} = \frac{1}{h_{c,c}A} \qquad (12.8a, b, c)$$

Note that if the "current" is $\dot{Q}$ rather than $\dot{Q}$, then the surface area A drops out. Also, the radiation term in Eq. 12.5 has been omitted: it is negligible for spark-ignition engines. The radiation term, important in the diesel gas-to-wall heat transfer process, can be linearized in $(\bar{T}_g - \bar{T}_w)$.[b]

Since the thermal inertia of the metal combustion chamber walls and the coolant are large, this sequence of heat transfer processes is effectively quasi-steady. Thus the thermal "circuit" acts as three resistances in series,

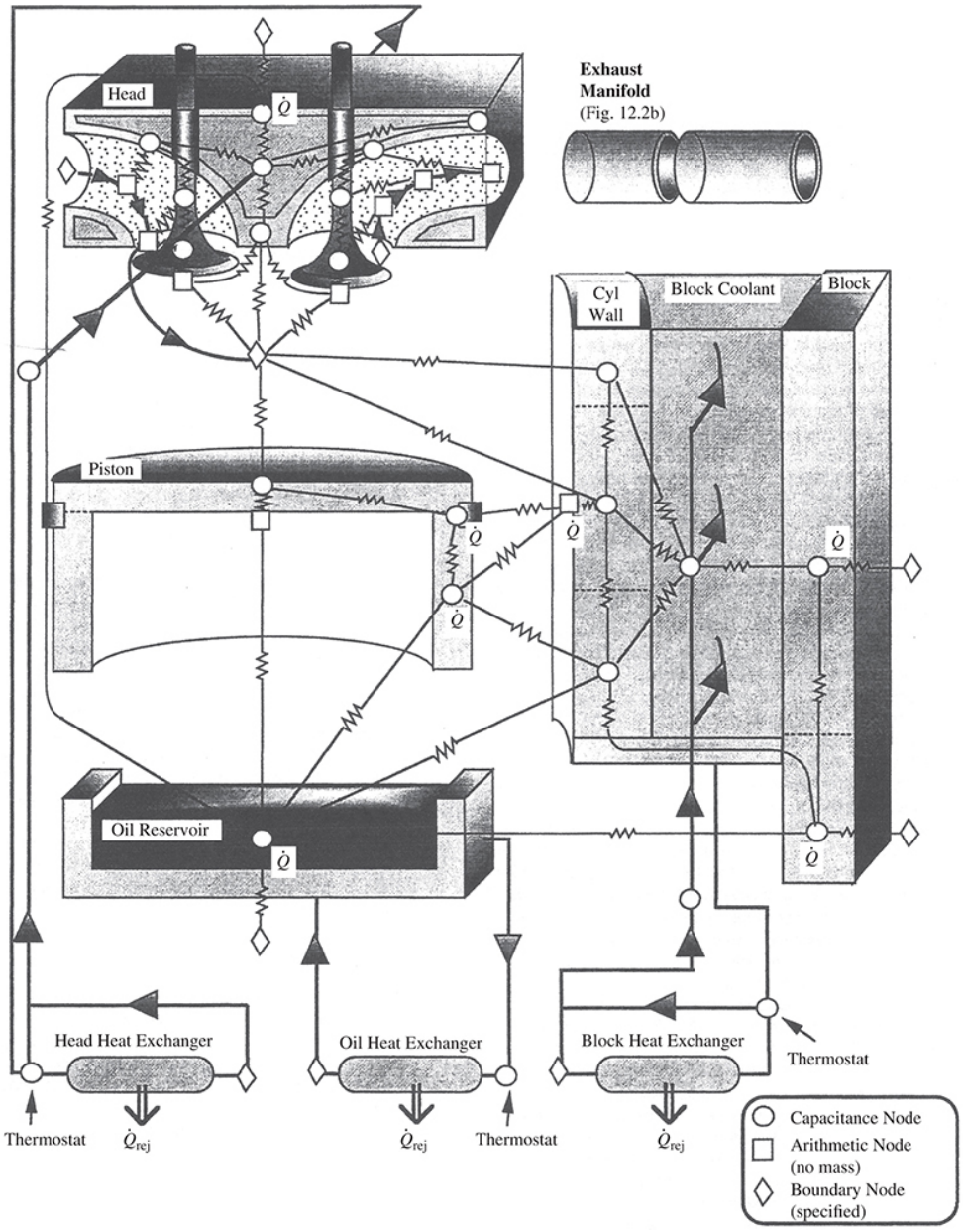
and the overall resistance (defined by an overall heat transfer coefficient h) is thus

$$\frac{1}{hA} = \frac{1}{h_g A} + \frac{t_w}{kA} + \frac{1}{h_c A} \tag{12.9}$$

Again, A can be removed from each term in Eq. (12.9) if the heat flux per unit area $\dot{q}$ is used instead of $\dot{Q}$. The temperature profile sketched in Fig. 12.1 (which is roughly to scale) indicates that the gas-to-metal thermal resistance [first term on the right-hand side in Eq. (12.9)] is by far the largest during the portion of the cycle when the in-cylinder gas temperature is much higher than the wall temperature. Note that k for an aluminum wall is three times k for cast iron (180 and 60 W/m · K), respectively.

This *thermal resistance* approach, analogous to an electrical resistance network, is commonly used to analyze more complex heat transfer problems. It can, of course, be coupled with thermal capacitances, analogous to electrical capacitance, to analyze unsteady engine heat transfer situations.

An example of a resistor-capacitor network model of one cylinder of an engine (piston, block, cylinder head, exhaust manifold) is shown in Fig. 12.2. [2] Such a model can simulate engine heat fluxes and component temperatures for steady-state and transient engine operating conditions (such as engine warm-up). Lumped capacitances are used to model the thermal capacity transients: heat-transfer submodels (e.g., conduction, convection) provide the resistive linkages. This lumped parameter method applies energy conservation to each node of the system. Here, three types of nodes are included. Capacitance nodes possess thermal capacity and thus provide the dynamic elements of the simulation. Arithmetic nodes have no mass or thermal capacity so respond instantaneously to changes in temperature of the surrounding nodes. These are appropriate when the time constant of a node's response is much shorter than that of the rest of the model. The third type of node is a boundary node, which has a specified temperature and is a boundary condition. Each node connects only with its adjacent nodes through conductive, convective, and radiative heat flows modeled, and shown, as resistors. The engine's fluids (intake flow—air and fuel, in–cylinder gases—which burn, exhaust flow, coolant flow, oil flow) drive or respond to this network. Reference 2 gives further details and results.


Head
$\dot{Q}$
Exhaust
Manifold
(Fig. 12.2b)
Cyl
Wall
Block Coolant
Block
Piston
$\dot{Q}$
$\dot{Q}$
$\dot{Q}$
$\dot{Q}$
Oil Reservoir
$\dot{Q}$
$\dot{Q}$
Head Heat Exchanger
Oil Heat Exchanger
Block Heat Exchanger
Thermostat
Thermostat
$\dot{Q}_{rej}$
$\dot{Q}_{rej}$
Thermostat
$\dot{Q}_{rej}$
Capacitance Node
Arithmetic Node
(no mass)
Boundary Node
(specified)

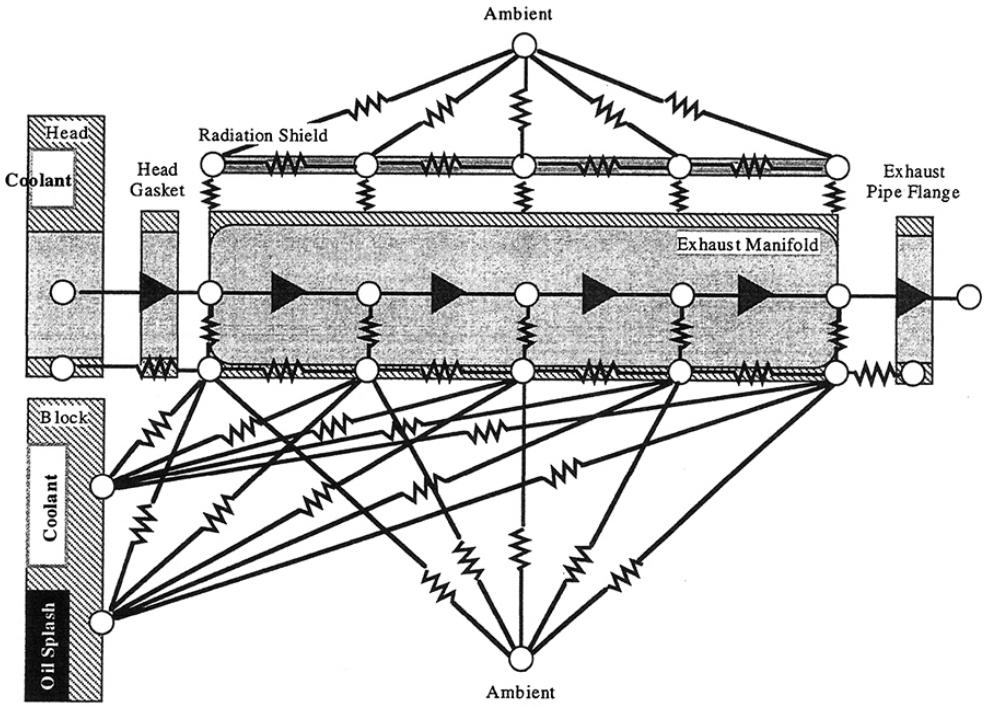


Figure 12.2 (*a*) Resistor-capacitor network model for analyzing heat flows and the resulting component temperatures in an internal combustion engine: piston, cylinder liner, cylinder head and valves, with coolant and oil systems. (*b*) Resistor-capacitor network model of the exhaust manifold that connects with the engine block network in (*a*). [2]

12.3 HEAT TRANSFER AND ENGINE ENERGY BALANCE

An overall first law energy balance for an engine provides useful information on the disposition of the initial fuel energy. For a control volume which surrounds the engine (see Fig. 3.8), the steady-flow energy-conservation equation is

$$\dot{m}_f h_f + \dot{m}_a h_a = P_b + \dot{Q}_{\text{cool}} + \dot{Q}_{\text{misc}} + (\dot{m}_f + \dot{m}_a) h_e \qquad (12.10)$$

where P_b is the brake power, $\dot{Q}_{\text{cool}}$ is the heat-transfer rate to the cooling medium, $\dot{Q}_{\text{misc}}$ is the heat rejection rate to the oil (if separately cooled) plus convection and radiation from the engine's external surface. It proves

convenient to divide the exhaust enthalpy h_e into a sensible part $h_{e,s} = h_e(T) - h_e(298\ \text{K})$, plus the exhaust reference state enthalpy (see Sec. 4.5). Then Eq. (12.10) can be written:

$$P_b + \dot{Q}_{\text{cool}} + \dot{Q}_{\text{misc}} + \dot{H}_{e,\text{ic}} + \dot{m}h_{e,s} = \dot{m}_f Q_{\text{LHV}} \tag{12.11}$$

where $\dot{H}_{e,\text{ic}}$ represents the exhaust enthalpy loss due to incomplete combustion of the fuel. Typical values of each of these terms relative to the fuel flow rate × heating value are given in Table 12.1.

TABLE 12.1 Energy balance for automotive engines at maximum power

	P_b	$\dot{Q}_{\text{cool}}$	$\dot{Q}_{\text{misc}}$	$\dot{H}_{e,\text{ic}}$	$\dot{m}h_{e,s}$
	(percentage of fuel heating value)				
SI engine	25–30	17–25	5–10	6–15	25–35
Diesel	35–40	20–35	2–6	1–2	22–35

Sources: From Sitkei,[1] Khovakh,[3] Burke et al.,[4] and Gruden et al.[5]

The energy balance within an engine is more complicated and is illustrated in the energy flow diagram shown in Fig. 12.3. The indicated power is the sum of the brake power and the friction power. A substantial part of the friction power is dissipated between the piston skirt and rings, and the cylinder wall, and is transferred as thermal energy to the cooling medium. The remainder of the friction power is dissipated in the bearings, valve mechanism, or drives auxiliary devices, and is transferred as thermal energy to the coolant, oil, or surrounding environment (in $\dot{Q}_{\text{misc}}$). The enthalpy in the exhaust gases exiting the cylinder can be subdivided into the following components: a sensible enthalpy (70%), an exhaust kinetic energy (7%), an incomplete combustion term (10%), and a heat transfer to the exhaust system (12%) (part of which is radiated to the environment, with the remainder ending up in the cooling medium). [c]

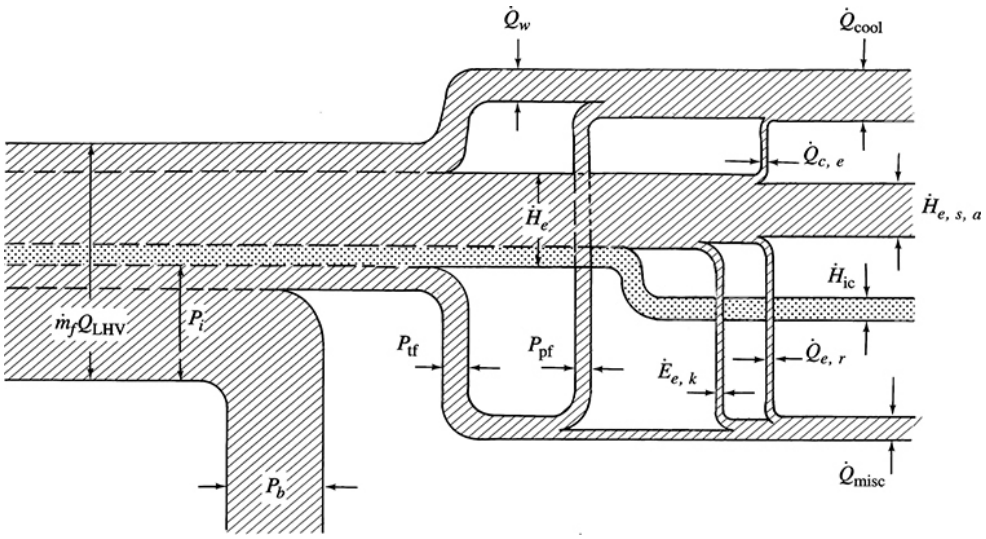


Figure 12.3 Energy flow diagram for a gasoline SI engine. $(\dot{m}_f \dot{Q}_{LHV})$ = fuel flow rate × lower heating value, $\dot{Q}_w$ = heat-transfer rate to combustion chamber wall, $\dot{H}_e$ = exhaust gas enthalpy flux, P_b = brake power, P_{tf} = total friction power, P_i = indicated power, P_{pf} = piston friction power, $\dot{Q}_{cool}$ = heat-rejection rate to coolant, $\dot{Q}_{c,e}$ = heat-transfer rate to coolant in exhaust ports, $\dot{Q}_{oil}$ heat rejection to oil, $\dot{H}_{e,s,a}$ = exhaust sensible enthalpy flux entering atmosphere, $\dot{H}_{e,ic}$ = exhaust chemical enthalpy flux due to incomplete combustion, $\dot{Q}_{e,r}$ = heat flux radiated from exhaust system, $\dot{E}_{e,k}$ = exhaust kinetic energy flux, $\dot{Q}_{misc}$ = sum of remaining energy fluxes and transfers such as radiation and convection from the outside surface of the engine.

Thus the heat carried away by the coolant medium consists of heat transferred to the combustion chamber walls from the gases in the cylinder, heat transferred to the exhaust valve and port in the exhaust process, and a substantial fraction of the friction work. The heat (thermal energy) transferred to the combustion chamber walls $(\dot{Q}_w)$ and the thermal energy that ends up in the coolant are not equal (though they are connected as illustrated in Fig. 12.3). This distinction is significant. First, $\dot{Q}_w$, is important primarily because it impacts the processes that take place inside the combustion chamber: second, $\dot{Q}_{cool}$, is important because it determines the design of the engine cooling system. The heat loss to the walls of the combustion chamber is only about two-thirds of the coolant thermal load; thermal energy generated by engine friction is the remaining one third.

Figure 12.4 shows an energy balance for a water-cooled spark-ignition engine, as a function of load (bmep), where the total initial fuel chemical energy is the normalizing factor. The brake power P_b expressed as a percent of fuel chemical energy is the engine's brake fuel conversion efficiency and rises from about 15% at low load to 34% at higher load (until enriching the mixture above stoichiometric to increase torque and avoid knock causes efficiency to level off and decline). At part-load, a greater fraction of the fuel chemical energy is absorbed into the coolant and oil, and leaves the cylinder as thermal energy in the exhaust. At low speeds and loads, the total coolant heat-transfer rate is about two times the brake power. Note also that at about three-quarters load, the exhaust kinetic energy $(\dot{E}_{e,k})$ has become significant as the engine's mass flow rate increases. The exhaust unreleased chemical energy (CO, H_2, and HC) also increases above about three-quarters load in this SI engine example because the mixture fuel-air equivalence ratio is enriched (steadily increased above stoichiometric) above this load to increase maximum power and prevent knock.

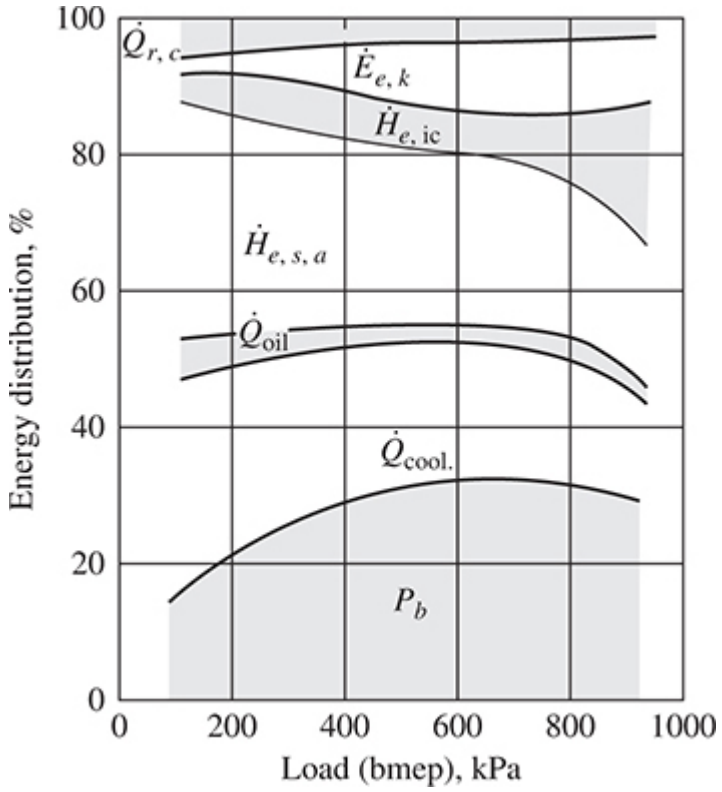


Figure 12.4 Energy balance for a gasoline spark-ignition engine as a function of load: brake power, coolant and oil thermal load, sensible exhaust

enthalpy, exhaust chemical and kinetic energy, and radiation and convection heat transfer from the block and head to the surrounding air; energy transfers as percent of fuel flow rate × heating value. Operating conditions: 2000 rev/min, bmep from 100 to 920 kPa. [5]

Although the heat losses are such a substantial part of the fuel energy input, elimination of heat losses would only allow a fraction of the heat transferred to the combustion chamber and exhaust port walls to be converted to useful work. The remainder would leave the engine as sensible exhaust enthalpy. Consider an example of an automotive naturally-aspirated high-speed CI engine with a compression ratio of 15. The indicated efficiency is 45%, and 25% of the fuel energy is carried away by the cooling water. Of this 25%, about 2 percentage points is due to friction. Of the remaining 23%, about 8% is heat loss during combustion, 6% heat loss during expansion, and 9% heat loss during exhaust. Of the 8 percentage points lost during combustion about, half (or 4% of the fuel energy) could be converted into useful work on the piston (see Fig. 5.9). Of the 6% heat loss during expansion, about one-third (or 2% of the fuel energy) could have been utilized. Thus, of the 25% lost to the cooling system, only about 6% (one-quarter) could have been converted to useful work on the piston. This upper bound (zero heat loss) would only increase the indicated efficiency of the engine from 45 to 51%.

For a spark-ignition engine, the conversion to useful work will be lower, because the compression ratio is lower. However, as shown in Fig. 12.4, the heat losses at part-load (an important operating regime for automobile use) are a substantially larger fraction of the fuel heating value. Studies with computer simulations of the SI engine operating cycle indicate that at typical part-load operating conditions a proportional reduction in combustion-chamber-wall heat losses of 10% results in a relative increase (an improvement) in brake fuel conversion efficiency of about 3%. [6]

12.4 CONVECTIVE HEAT TRANSFER

12.4.1 Dimensional Analysis

While the overall time-averaged heat transfer to the coolant medium is adequate for some design purposes, the instantaneous heat flux during the engine cycle, between the in-cylinder gases and combustion chamber walls, is a necessary input for realistic cycle calculations (see Sec. 14.4) and provides the fundamental input for obtaining the heat flux distribution to the various parts of an operating engine. Equations (12.5) , (12.6), and (12.7) provide the framework for calculating the heat flux $\dot{Q}$ based on the assumption that at each point in the cycle the heat-transfer process is quasi steady. For example, neglecting radiation, if $\overline{T}_g$, $\overline{T}_{w,g}$, and $h_{c,g}$ can be calculated at each point in the cycle, $\dot{q}(\theta)$ is obtained. Alternatively, if $\overline{T}_c$, $h_{c,c}$, $\overline{T}_g$, and $h_{c,g}$ are known, $\dot{Q}$, $T_{w,g}$, and $T_{w,c}$ can be computed.

Dimensional analysis can be used to develop the functional form of relationships which govern the gas-side heat-transfer coefficient. [7] The engine convective heat-transfer process can be characterized geometrically by a length dimension—say the bore B—and a number of length ratios y_1, y_2, y_3, etc. (of which one will be the axial cylinder length z divided by the bore z/B), which define the cylinder and combustion chamber geometry. The flow pattern, similarly, may be characterized by one chosen velocity v and a set of velocity ratios u_1, u_2, u_3, etc. The gas properties of importance are the thermal conductivity k, the dynamic viscosity μ, the specific heat c_p, and the density ρ. If there is combustion, then the chemical energy release rate per unit volume $\dot{q}_{\text{ch}}$ may be important. The engine speed N and relative position in the cycle denoted by crank angle θ introduce the cyclical nature of the process. Thus

$$f(h_c, B, z, y_1, y_2, \ldots y_m, v, u_1, u_2, \ldots, u_n, k, \mu, c_p, \rho, \dot{q}_{\text{ch}}, N, \theta) = 0$$

Applying dimensional analysis, with mass, length, time, and temperature as the independent dimensions, reduces this set of actual variables to four-fewer nondimensional groups:

$$F\left(\frac{h_c B}{k}, \frac{\rho v B}{\mu}, \frac{c_p \mu}{k}, \frac{c_p T}{v^2}, \frac{NB}{v}\frac{\dot{q}_{\text{ch}}}{\rho c_p N T}, \frac{z}{B}, y_1, \ldots, y_m, u_1, \ldots, u_n, \theta\right) = 0$$

The first three groups are the familiar Nusselt, Reynolds, and Prandtl numbers. The next has the form of a Mach number since c_pT is proportional to the square of the sound speed. For Mach numbers much less than 1, the Mach number dependence is known to be small and can be omitted. It is usual to take for v the mean piston speed $\bar{S}_p = 2LN$. Then, by introducing the bore/stroke ratio B/L, the term NB/v is eliminated. z/B is a function of the compression ratio r_c, the ratio of connecting rod to crank radius $R = l/a$, and θ. Thus

$$F\left(\frac{h_c B}{k}, \frac{\rho \bar{S}_p B}{\mu}, \frac{c_p \mu}{k}, \frac{B}{L}, \frac{\dot{q}_{ch}}{\rho c_p N T}, r_c, R, y_1, \ldots, y_m, \theta, u_1, \ldots, u_n\right) = 0 \tag{12.12}$$

The dimensionless groups may be varied (but not reduced in number) by combination. While Eq. (12.12) reveals nothing about the functional form of the relationship between the groups, it provides a basis for evaluating the correlations which have been proposed.

Many formulas for calculating instantaneous engine heat-transfer coefficients have been developed (see Ref. 8 for a review). Only those with a functional form that fits Eq. (12.12) will be summarized here. The basis of these correlations is the assumption that the Nusselt, Reynolds, and Prandtl number relationship will have the same form as that used for turbulent flow in pipes and over flat plates:

$$\mathrm{Nu} = a\,\mathrm{Re}^m\,\mathrm{Pr}^n \tag{12.13}$$

Distinctions should be made between correlations intended to predict the *time- averaged* heat flux to the combustion chamber walls, the *instantaneous spatially averaged* heat flux to the chamber walls (which is required for engine performance analysis), and the *instantaneous local* heat fluxes (which are not uniform over the combustion chamber and may be required for thermal stress calculations). In using these heat-transfer correlations, the critical choices to be made are (1) the velocity to be used in the Reynolds number; (2) the gas temperature at which the gas properties in Eq. (12.13) are evaluated; and (3) the gas temperature used in the convective heat-transfer equation, Eq. (12.2) .

The most widely used correlations and the basis of their derivation will now be summarized. Because the experimental data for evaluating these correlations in CI engines includes both convective and radiative heat fluxes, comparison of these correlations with data is deferred to Sec. 12.6.4 .

12.4.2 Correlations for Time-Averaged Heat Flux

Taylor and Toong [9] have correlated overall heat-transfer data from 19 different engines. It was assumed that coolant and wall temperatures varied little between designs and that the effects of geometrical differences were small. Thus, at a given fuel/air ratio, the convective part of the heat flux should correlate with Reynolds number. To allow for variations in fuel/air ratio, Taylor and Toong defined an average effective gas temperature $T_{g,a}$ such that

$$\int Ah_c(T - T_{g,a})d\theta = 0$$

over the engine cycle. $T_{g,a}$ is the temperature at which the wall would stabilize if no heat was removed from the outside of the wall. $T_{g,a}$ was obtained by extrapolating average heat-transfer data plotted versus gas-side combustion chamber surface temperature back to the zero heat-transfer axis. The Nusselt number, defined as

$$\mathrm{Nu} = \frac{\bar{\dot{Q}}B}{(\pi B^2/4)(T_{g/a} - T_c)k_g} = \frac{4\bar{\dot{Q}}}{\pi B k_g (T_{g,a} - T_c)} \tag{12.14}$$

plotted against Reynolds number, defined as

$$R_e = \frac{\dot{m}B}{\mu_g(\pi B^2/4)} = \frac{4\dot{m}}{\pi \mu_g B} \tag{12.15}$$

where $\dot{m}$ is the charge mass flow rate, is shown in Fig. 12.5. Taylor and Toong proposed a Reynold's number power law with m in Eq. (12.13) equal to 0.75. [d] Annand[7] suggests three separate lines for the three different types of engines covered, with slope 0.7. The diesel line is about 25% higher than the

spark-ignition engine line (which corresponds in part to the radiative heat flux component present in diesels). The air-cooled engine line is lower than the liquid-cooled line presumably because surface temperatures are higher. The average gas temperature values developed by Taylor and Toong are shown in the insert in Fig. 12.5.

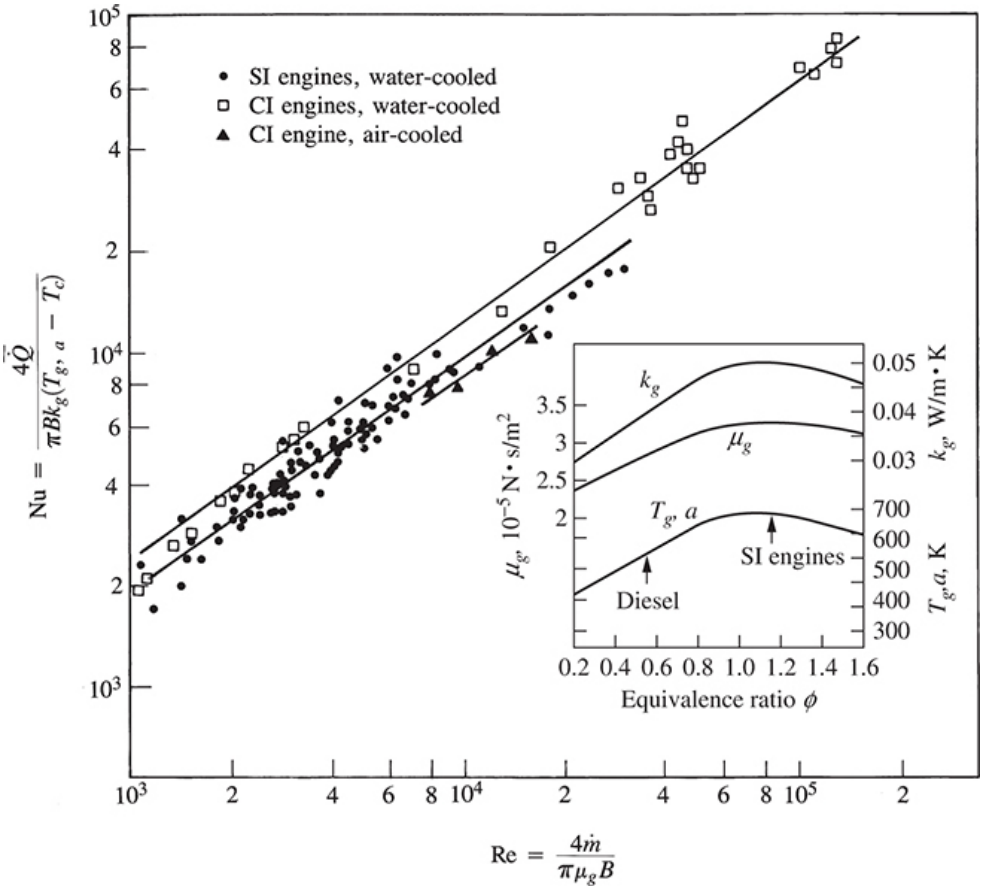


Figure 12.5 Overall engine heat-transfer correlation: gas-side Nusselt number versus Reynolds number for different types of IC engines. See text for definition of symbols. Insert gives effective gas temperature (wall temperature for adiabatic operation), gas viscosity μ_g and thermal conductivity k_g. Lines have slope 0.7. [7, 9]

12.4.3 Correlations for Instantaneous Spatial Average Coefficients

Annand [7] developed the following convective heat-transfer correlation to match previously published experimental data on instantaneous heat fluxes to selected cylinder head locations:

$$\left(\frac{h_c B}{k}\right) = a\left(\frac{\rho \bar{S}_p B}{\mu}\right)^b \tag{12.16}$$

The value of a varied with intensity of charge motion and engine design. With normal combustion, $0.35 \leq a \leq 0.8$ with $b = 0.7$; a increases with increasing intensity of charge motion. Gas properties are evaluated at the cylinder-average charge temperature $\bar{T}_g$:

$$\bar{T}_g = \frac{pVM}{m\tilde{R}} \tag{12.17}$$

The same temperature is used in Eq. (12.2) to obtain the convective heat flux.

Note that in developing this correlation, the effects of differences in geometry and flow pattern between engines [the ratios $y_1, \ldots, y_m$ and $u_1, \ldots, u_n$ in Eq. (12.12)] have been incorporated in the proportionality constant a, and the effect of chemical energy release is omitted. While only data from cylinder head thermocouple locations were used as a basis for this correlation, it has often been used to estimate instantaneous spatial average heat fluxes for the entire combustion chamber.

Woschni [10] assumed a correlation of the form

$$\mathrm{Nu} = 0.035\ \mathrm{Re}^m \tag{12.18}$$

With the cylinder bore B taken as the characteristic length, with w as a local average gas velocity in the cylinder, and assuming $k \propto T^{0.75}$, $\mu \propto T^{0.62}$, and $p = \rho RT$, the above correlation can be written

$$h_c = CB^{m-1} p^m w^m T^{0.75-1.62m} \tag{12.19}$$

During intake, compression, and exhaust, Woschni argued that the average gas velocity should be proportional to the mean piston speed. During combustion and expansion, he attempted to account directly for the gas

velocities induced by the change in density that results from combustion, which are comparable to mean piston speeds (~10 m/s). Thus a term proportional to the pressure rise due to combustion (p - pm) was added (p_m is the motored cylinder pressure). The coefficients relating the local average gas velocity w to the mean piston speed and (p - pm) were determined by fitting the correlation, integrated over the engine cycle, to time-averaged measurements of heat transfer to the coolant for a wide range of engine operating conditions for a direct-injection four-valve diesel without swirl. T in Eq. (12.19) is the mean cylinder gas temperature defined by Eq. (12.17) ; the same temperature is used to obtain the heat flux from the heat-transfer coefficient h_c. Thus this correlation represents *spatially averaged* combustion chamber heat fluxes.

The average cylinder gas velocity w (meters per second) determined for a four-stroke, water-cooled, four-valve direct-injection CI engine without swirl was expressed as follows:

$$w = \left[C_1 \bar{S}_p + C_2 \frac{V_d T_r}{p_r V_r} (p - p_m) \right] \tag{12.20}$$

where V_d is the displaced volume, p is the instantaneous cylinder pressure, p_r, V_r, T_r are the working-fluid pressure, volume, and temperature at some reference state (say inlet valve closing or start of combustion), and p_m is the motored cylinder pressure at the same crank angle as p, with:

For the gas exchange period:	$C_1 = 6.18,$	$C_2 = 0$
For the compression period:	$C_1 = 2.28,$	$C_2 = 0$
For the combustion and expansion period:	$C_1 = 2.28,$	$C_2 = 3.24 \times 10^{-3}$

Subsequent studies in higher-speed engines with swirl indicated higher heat transfer than these velocities predicted. For engines with swirl, cylinder averaged gas velocities were given by Eq. (12.20) with:

For the gas exchange period:	$C_1 = 6.18 + 0.417 \frac{v_s}{\bar{S}_p}$
For the rest of cycle:	$C_1 = 2.28 + 0.308 \frac{v_s}{\bar{S}_p}$

where v_S, *the swirl velocity, is* $B\omega_p/2$ and ω_p is the rotation speed of the paddle wheel used to measure the swirl velocity (see Sec. 8.3.1).[11] Spark-ignition engine tests showed that the above velocities gave acceptable predictions for this type of engine also.[12]

Woschni's correlation, with the exponent m in Eq. (12.19) equal to 0.8, has been widely used, and can be summarized as:

$$h_c(\mathrm{W/m^2 \cdot K}) = 3.26B(\mathrm{m})^{-0.2}\, p(\mathrm{kPa})^{0.8}\, T(\mathrm{K})^{-0.55}\, w(\mathrm{m/s})^{0.8} \tag{12.21}$$

with w defined above, Eq. (12.20) .

Hohenberg [13] examined Woschni's formula and made changes to give better predictions of time-averaged heat fluxes measured with probes in a direct-injection diesel engine with swirl. The modifications include use of a length based on instantaneous cylinder volume instead of bore, changes in the effective gas velocity, and in the exponent of the temperature term. Han et al . [14] have proposed as empirically based correlation of similar form to Eq. (12.21) with the same variables (B, p, T, and w) but modified exponents, and with a different method of calculating the representative in-cylinder gas velocity through compression, combustion, and expansion. They showed acceptable agreement with the heat transfer coefficients they determined from time-resolved surface temperature measurements in a gasoline-fueled SI engine.

12.4.4 Correlations for Instantaneous Local Coefficients

LeFeuvre et al . [15] and Dent and Sulaiman [16] have proposed the use of the flat-plate forced convection heat-transfer correlation formula

$$\left(\frac{h_c l}{k}\right) = 0.036\left(\frac{\rho v l}{\mu}\right)^{0.8}\left(\frac{\mu c_p}{k}\right)^{0.333} \tag{12.22}$$

where l is the length of the plate and v the flow velocity over the plate. This formula has been applied to DI diesel engines with swirl, with l and v evaluated at a radius r as

$$l = 2\pi r \; v = r\omega$$

ω being the solid-body angular velocity of the charge. The heat flux at any radius r (with Pr = 0.73) is then given by

$$\dot{q}(r) = 0.023 \frac{k}{r} \left(\frac{\omega r^2}{\nu} \right)^{0.8} [\bar{T}_g(r) - T_w(r)] \tag{12.23}$$

This equation can be used if the swirl variation with crank angle is known (see Sec. 8.3) and an appropriate local gas temperature can be determined. It would not be consistent to use the cylinder average gas temperature given by Eq. (12.17) because during combustion substantial temperature non-uniformities exist between burned gases and air or mixture which has yet to burn or mix with already burned gas.

An alternative approach is zonal modeling, where the combustion chamber is divided into a relatively small number of zones each with its own temperature, heat-transfer coefficient, and heat-transfer surface area history. This approach has been applied to spark-ignition engines (e.g., Ref. 17), where the division of the in-cylinder gases during combustion into a higher-temperature burned gas region behind the flame and lower-temperature unburned gas region ahead of the flame is clear (see Fig. 9.6). The heat transfer to the combustion chamber surfaces in contact with the unburned and burned gas zones [analogous to Eq. (12.2)] is given by

$$\dot{Q}_u = A_{u,w} h_{c,u} (T_u - T_w) \quad \dot{Q}_b = A_{b,w} h_{c,b} (T_b - T_w) \tag{12.24a, b}$$

respectively. Since h_c depends on local gas properties and velocities, $h_{c,u}$ and $h_{c,b}$ are not necessarily the same. Examples of how the burned-gas wetted areas on the piston, cylinder head, and liner vary during the combustion process are given in Fig. 14.8. Since the burned gas temperature T_b is much larger than the unburned gas temperature, the heat flux from the burned gas zone dominates.

One useful development of this two-zone approach is the division of the burned gas zone into an adiabatic core and a thermal boundary layer. The advantages are: (1) this corresponds more closely to the actual temperature distribution (see Sec. 12.6.5); and (2) a model for the boundary-layer flow

provides a more fundamental basis for evaluating the heat-transfer coefficient. The local heat flux is then given by [18]

$$\dot{q} = \frac{k_e(T_{ac} - T_w)}{\delta} \tag{12.25}$$

where k_e is the effective thermal conductivity in the boundary layer, T_{ac} is the adiabatic core temperature, and δ the boundary-layer thickness. In the laminar regime δ would grow as $t^{1/2}$; in the turbulent regime δ would grow as $t^{0.8}$. Both growth regimes are observed.

Zonal models have also been used to describe DI diesel engine heat transfer. [19] A bowl-in-piston chamber was divided into three flow regions and two gas-temperature zones during combustion. An effective velocity

$$w = (U_x^2 + U_y^2 + 2k)^{1/2}$$

was used to obtain the heat-transfer coefficient, where U_x and U_y are the two velocity components parallel to the surface outside the boundary layer and k is the turbulent kinetic energy. Zonal models would be expected to be more accurate than global models. However, only limited validation has been carried out.

12.4.5 Exhaust and Intake System Heat Transfer

Convective heat transfer in the intake and exhaust systems is driven by higher flow velocities than in-cylinder heat transfer. Intake system heat transfer is usually described by steady, turbulent pipe-low correlations. [7] With liquid fuel present in the intake, the heat-transfer phenomena become more complicated (see Sec. 7.6). Exhaust flow heat-transfer rates are the largest in the entire cycle due to the very high gas velocities developed during the exhaust blowdown process and the high gas temperature (see Sec. 6.5). Exhaust system heat transfer is especially important since it affects emissions burn-up in the exhaust port, catalyst, or particulate trap: it also influences turbocharger performance, and contributes significantly to engine cooling requirements.

The highest heat-transfer rates occur during the exhaust blowdown, to the exhaust valve and port. Detailed exhaust port convective heat-transfer correlations have been developed and tested. These are based on Nusselt-Reynolds number correlations. For the valve open period, relations of the form

$$\mathrm{Nu} = K\,\mathrm{Re}_j^n \tag{12.26}$$

have been proposed and evaluated. [20] For $L_v/D_v \lesssim 0.2$, the flow exits the valve as a jet, and $\mathrm{Re}\,j = v_j D_v/\nu$, where D_v is the valve diameter, v_j the velocity of the exhaust gases through the valve opening, and ν the kinematic viscosity. For $L_v/D_v \lesssim 0.2$, the port is the limiting flow area and a pipe flow model with $\mathrm{Re} = v_p D_p/\nu$ is more appropriate. v_p is the gas velocity in the port and D_p is the port diameter.

For the valve closed period, the correlation

$$\mathrm{Nu} = 0.022\overline{\mathrm{Re}}_D^{0.8} \tag{12.27}$$

was developed. Here $\overline{\mathrm{Re}}_D = \bar{v}_p D_p/\nu$, where is the time-averaged exhaust port gas velocity and D_p is the port diameter.

For straight sections of exhaust pipe downstream of the port, an empirical correlation based on measurements of average heat-transfer rates to the pipe has been derived: [21]

$$\mathrm{Nu} = 0.0483\,\mathrm{Re}^{0.783} \tag{12.28}$$

The Reynolds number is based on pipe diameter and gas velocity. Heat-transfer correlations for steady developing turbulent flow in pipes predict values about half that given by Eq. (12.28) , in part because the exhaust gas flow is pulsating. Detailed analysis of exhaust system heat transfer in a gasoline engine can be found in Ref. 21. Included are measurements of wall temperature and heat flux measurements, and water condensation effects, during the engine warm-up process.

Heat transfer between the intake system walls and the intake airflow (plus fuel flow into the intake port with port fuel injection) is much less in

magnitude than exhaust flow heat transfer, but does impact the engine's volumetric efficiency, propensity to knock at higher loads, and NO_x emissions, due to heating of the air on its way into the engine cylinder. Intake port wall and valve surface temperatures rise during warm-up from ambient levels in a cold engine to the quasi-steady levels of a warmed-up engine (intake port walls are close to the coolant temperature: intake valve head surface temperatures are 150 to 250°C, depending on operating conditions). While most of the intake flow is toward the cylinder, during the valve overlap period initiated by the intake valve opening, especially with throttled engine operation, there is a strong backflow of hot burned gases from the cylinder into the intake port. Under these throttled operating conditions, the intake pressure is significantly lower than the exhaust pressure so this backflow occurs with high gas velocities, exiting the valve seat region as a conical jet that impacts the valve head and port walls close to the valve, significantly. For a short time period (20 to 40 crank-angle degrees, some 2 to 4 ms), this hot gas backflow results in some heating of the intake valve and upstream port surfaces. With port fuel injection systems, the intake port and valve surface temperatures are high enough to affect the fuel vaporization process that results from these several intake thermal phenomena. The fraction of the injected gasoline that vaporizes is small (~10%) with a cold (~20°C) engine, rises increasingly rapidly with rising surface temperatures to close to 100% for a fully warmed-up engine. [22]

Additional details of intake system heat transfer rates (per cycle, and cycle resolved), on intake valve surface temperatures, and effects on fuel vaporization in port fuel injected engines, can be found in Refs. 23 to 26. Typically, as a result of heat transfer from the intake valve and port walls, the temperature of the intake mixture entering the cylinder of a gasoline or diesel engine is 20 to 30°C higher than the air temperature in the intake manifold.

12.5 RADIATIVE HEAT TRANSFER

There are two sources of radiative heat transfer within the cylinder: the high-temperature burned gases and the soot particles which burn up in the diesel engine flame. In a spark-ignition engine, the flame propagates across the combustion chamber from the point of ignition through previously mixed fuel

and air. Although the flame front is slightly luminous (see Fig. 9.1), all the chemical intermediaries in the reaction process are gaseous. Combustion is essentially complete within the first third of the expansion stroke. In the compression-ignition engine the fuel partially burns in a rich reaction region within each spray: combustion then continues in a turbulent diffusion flame as this partially reacted fuel and air mix together and burn up. That diffusion flame separates the fuel sprays from the surrounding air (see Figs. 10.4 and 10.5). The flame is highly luminous because soot particles (which are mostly carbon) form within the fuel sprays, enter the diffusion flame, burn, become hot and radiate.

The radiation from soot particles in the diesel engine flame is about five times the radiation from the high-temperature gaseous combustion products. Radiative heat transfer in conventional spark-ignition engines is small in comparison with convective heat transfer from the hot burned gases. However, radiative heat transfer in diesel engines is not negligible; it contributes 20 to 35% of the total heat transfer and a higher fraction of the maximum heat-transfer rate.

12.5.1 Radiation from Gases

Gases absorb and emit radiation in narrow wavelength bands rather than in a continuous spectrum as do solid surfaces. The simpler gas molecules such as H_2, O_2, and N_2 are essentially transparent to radiation. Of the gases important in combustion, CO, CO_2, and H_2O emit sufficient energy to warrant consideration. In gases, emission and absorption of radiation will occur throughout the gas volume. These processes will be governed by the temperature, and the number of molecules, along the radiation path. For each species, the latter will be proportional to the product of the species partial pressure p_i and the path length l. Also, the radiative capacity depends on gas temperature T_g. Thus the emissivity of the gas ε_g can be expressed as

$$\varepsilon_g = f(T_g, p_1 l, \ldots, p_n l) \tag{12.29}$$

The mean path length for a volume V with surface area A is given with sufficient accuracy by

$$l = 0.9 \times \frac{4V}{A} \tag{12.30}$$

$4V/A$ is the mean path length for a hemispherical enclosure.

Standard methods have been developed for estimating ε_g for mixtures of CO_2 and H_2O by Hottel and others (see Ref. 27). Charts based on experimental data give ε_{CO_2} and ε_{H_2O} as functions of $p_i l$ and T_g. Correction factors are applied for total pressures above one atmosphere and for the overlapping of spectral bands of CO_2 and H_2O. Estimates for engine combustion gases at peak conditions give $\varepsilon_g \approx 0.1$ and peak heat fluxes due to gas radiation of order 0.2 MW/m^2. This amounts to some 5% of the peak convective heat transfer. Since gas radiation is proportional to T_g^4, this radiative flux falls off more rapidly from peak values than convective heat flux as gas temperatures fall and, when integrated over the cycle, can be neglected.

12.5.2 Flame Radiation

Flame radiation in diesel engines is a much more complex process because the detailed geometry and chemical composition of the radiating region are not well defined. Since the radiation from the optically transparent nonluminous flames of spark-ignition engines is small, we will deal only with luminous nontransparent flames where the radiation comes from incandescent soot particles and has a continuous spectrum. Because the particle size distribution, number density, temperature, and flame geometry in a diesel are not easily quantified, flame emissivities cannot be calculated from first principles. Direct measurements of flame emissivities are required. A number of measurements of the magnitude and spectral distribution of radiation from a diesel engine combustion chamber have been made (see Ref. 8 for a summary). Flynn et al. [28] in a direct-injection diesel engine used a monochromator to measure intensity of radiation at seven wavelengths. The viewing path cut through the piston crown into the central region of the bowl-in-piston combustion chamber. Fuel was injected through a five-hole nozzle and some air swirl was provided. At any given crank angle, the distribution of energy over the seven wavelengths was used to

reconstruct the complete energy spectrum and to calculate the apparent radiation temperature and optical thickness. This approach has been used by others. [29]

The radiant energy distribution is skewed from that of a gray-body model (for which emissivity is independent of wavelength). The monochromatic emissivity is well fitted by an equation of the form

$$\varepsilon_\lambda = 1 - \exp\left(\frac{-kl}{\lambda^\alpha}\right) \tag{12.31}$$

used by Hottel and Broughton [30] and others to describe the emissivity of clouds of soot particles in flames. There is much discussion as to the appropriate value of a. A value of 0.95 is often used. Figure 12.6 shows sample results for the monochromatic emissive power and monochromatic emissivity. Equation (12.31) , combined with Planck's equation for black-body monochromatic emissive power

$$e_{b,\lambda} = \frac{2\pi K_1}{\lambda^5 (e^{K_2/\lambda T_R} - 1)} \tag{12.32}$$

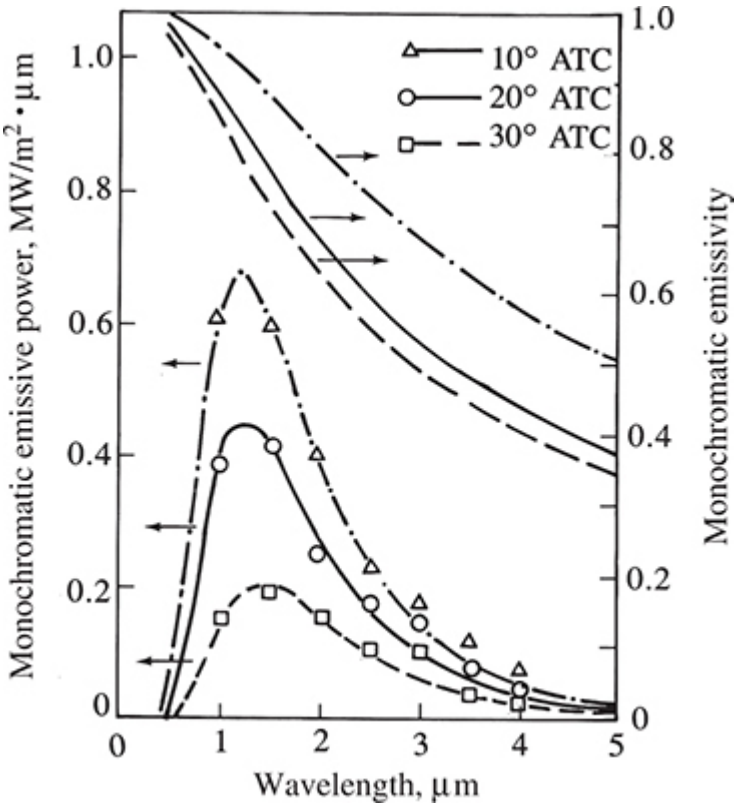


Figure 12.6 Variation in monochromatic emissive power and emissivity, with wavelength, at three different crank angles. DI diesel engine with 114-mm bore, 1995 rev/min, overall equivalence ratio 0.46. Radiation from piston bowl measured through cutout in piston crown. [28]

where $K_1 = 0.59548 \times 10^{-16}$ W · m^2 and $K_2 = 1.43879$ cm · K, defines an apparent radiation temperature T_R and optical thickness kl for the radiating medium. An apparent gray-body emissivity ε_a can then be calculated from the standard definition of ε_a,

$$\varepsilon_a = \frac{\int \varepsilon_\lambda e_{b,\lambda} d\lambda}{\int e_{b,\lambda} d\lambda}$$

Additional details of the methodology can be found in Ref. 29.

Figure 12.7 shows the radiant heat flux rate (MW/m^2) measured in the cylinder head of a single-cylinder version of the Cummins N14 turbocharged

six-cylinder in-line DI diesel engine, which has a low-swirl Mexican-hat-shaped combustion chamber. Five different individual cycles and an average over 200 cycles are shown. There is significant cycle-to-cycle variation in the profile, presumably due to differences between the development of each individual diffusion flame.

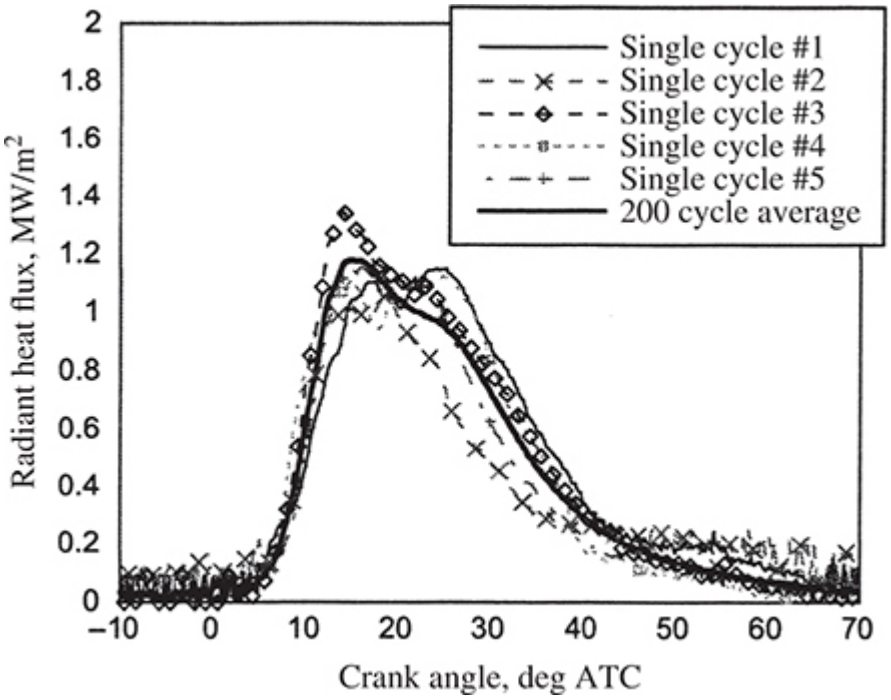


Figure 12.7 Radiant heat flux, for five individual cycles and 200-cycle average, as function of crank angle for DI diesel engine at high loads. Engine and measurement details in reference: fuel/air equivalence ratio 0.635, 1200 rev/min. [29]

The shapes of these radiant heat flux profiles follow that of the second, mixing-controlled diffusion-flame, apparent heat release rate (AHRR) profiles (see Fig. 12.8, and Sec. 10.3.3). This would be expected since the AHRR scales with fuel burning rate, and thus the surface area of the diffusion flame surrounding each fuel spray. This approximate scaling is evident in the two radiant flux and AHRR profiles a high (on left) and low load (on right) shown in Fig. 12.8.

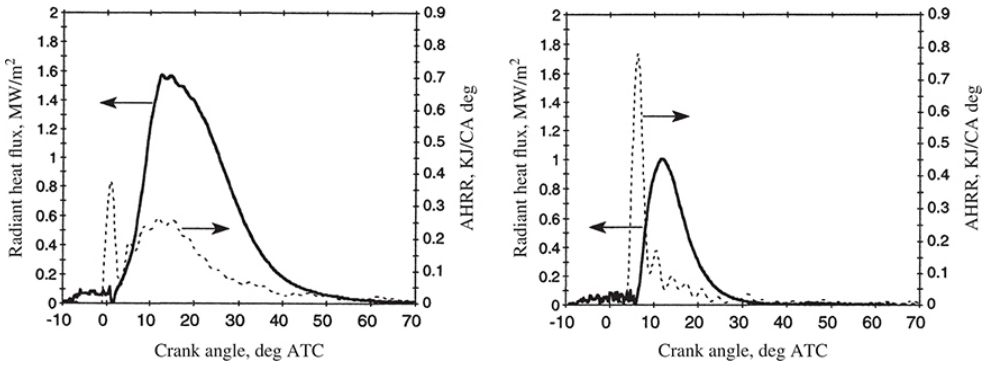


Figure 12.8 Radiant heat flux and apparent heat release rate (AHRR) profiles at 100% load (on left) and 50% load (on right): start of injection 14.5° CA before TC, 1500 rev/min; single-cylinder version of Cummins N14 turbocharged DI diesel engine.[29]

During the period of maximum radiation, the apparent emissivity is 0.8 to 0.9; it then decreases as the expansion process proceeds. [28] A comparison of radiant and total heat fluxes (both peak and average) shows that the radiation heat flux can be a substantial fraction of the peak heat flux. The average radiant heat flux is about 20% of the average total flux: the percentage varies with load.

The radiation and the apparent flame temperatures measured in diesels by several investigators show consistent results (see Fig. 12.9). Also included in the figure during the combustion and expansion process are typical values of: (1) the temperature of any air not yet mixed with fuel or burned gases; (2) the average temperature of the cylinder contents; and (3) the maximum possible flame temperature [corresponding to combustion of a slightly rich mixture ($\phi = 1.1$) with air at the temperature shown]. The measured radiation temperatures fall between the maximum flame temperatures and the bulk temperature. Models for obtaining the relevant temperatures have been proposed (e.g., the diffusion flame reaction zone is stoichiometric [16, 19]) to fit the available data, and have been used to define an appropriate flame temperature. [31]

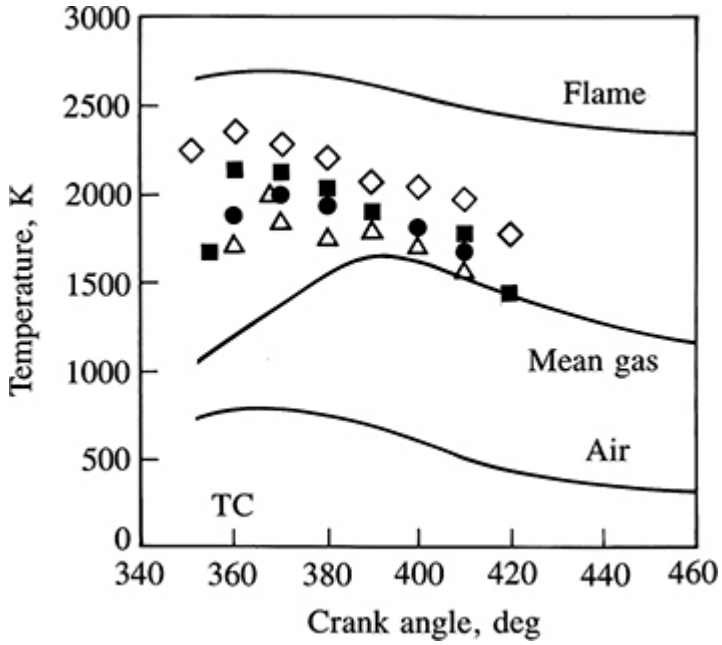


Figure 12.9 Apparent radiation temperatures measured in diesel engines, compared to calculated maximum adiabatic flame, cylinder-mean, and air temperatures, during the combustion period. Adiabatic flame temperature is temperature attained by burning air at the air temperature with fuel for an equivalence ratio of 1.1. (*Data from Dent and Sulaiman,* [16] *Flynn et al.,* [28] *Lyn,* [32] *Kamimoto et al.*; [33] *calculated curves from Assanis and Heywood.* [31])

The emissivity of an incandescent soot-burning flame can be calculated from a knowledge of the monochromatic absorption coefficient, which is given by

$$k_\lambda = 36\pi \frac{c_s}{\rho_s} \frac{n^2\kappa}{\lambda[(n^2+n^2\kappa^2)^2+4(n^2-n^2\kappa^2+1)]} = 36\pi \frac{c_s}{\rho_s} f(\lambda,T) \qquad (12.33)$$

where n is the refractive index, κ is the absorption term in the complex refractive index, c_s is the soot concentration in kilograms per cubic meter, and ρ_s is the density of the soot particles ($\approx$2 g/m^3). [34] The absorption coefficient depends on wavelength and temperature, is independent of particle size, and depends only on the soot mass loading. Figure 12.10 shows several estimates of k_λ/c_s as a function of λ. The strong dependence on λ

shows that clouds of soot particles are markedly not gray. There is a considerable spread in the different estimates shown. This could be the result of different soot compositions (the C/H ratio affects the optical properties) and temperatures. To find a mean value for the absorption coefficient of a wide spectrum of radiation, the expression

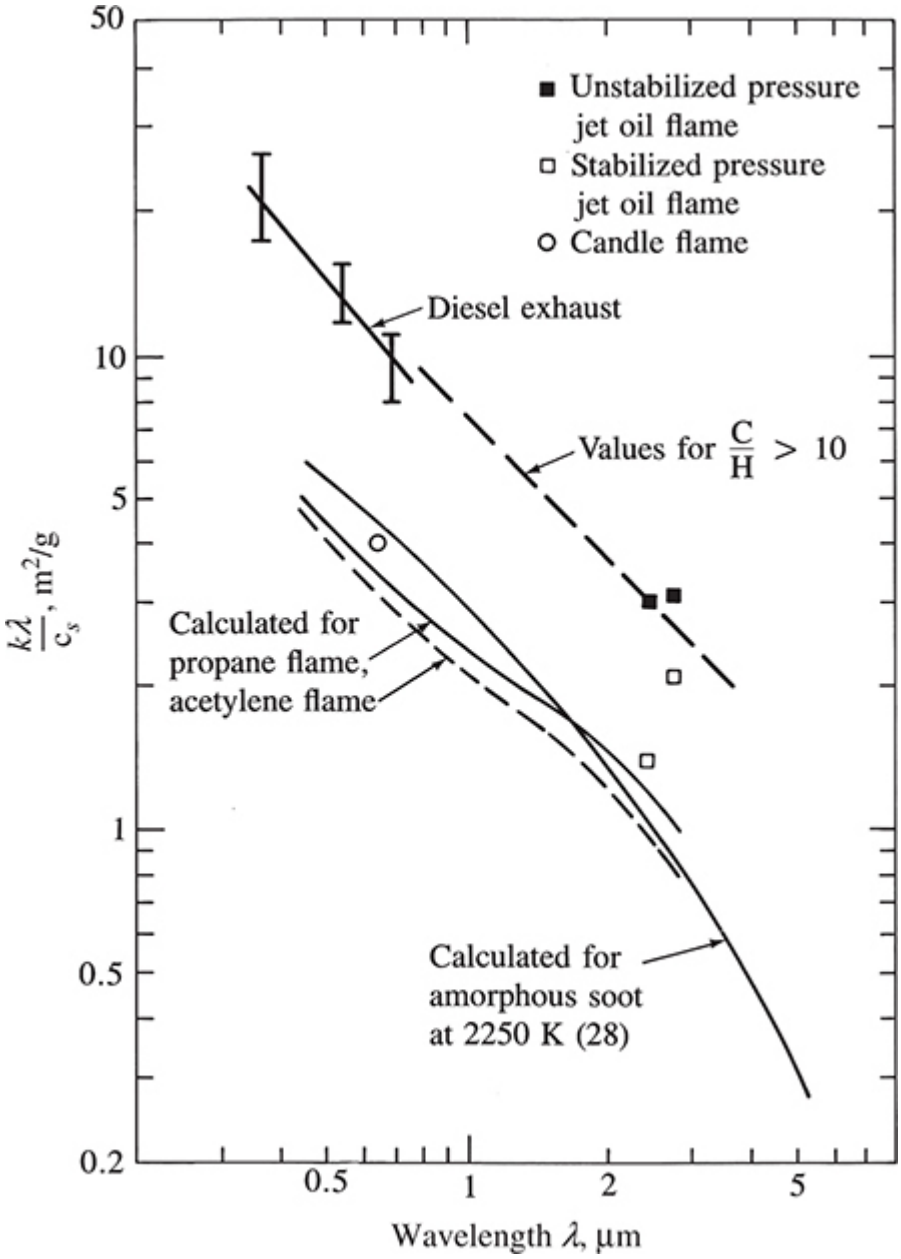


Figure 12.10 Measured and calculated values of k_λ/c_s as a function of wavelength λ. k_λ is monochromatic absorption coefficient, c_s is the soot concentration. (*From Field et al.* [34] *and Greeves and Meehan.* [35])

$$k_a = \frac{\int k_\lambda e_{b,\lambda} d\lambda}{\int e_{b,\lambda} d\lambda}$$

must be evaluated. For example, for radiation from a black body at temperature 1800 K, the mean absorption coefficient is about 1300 c_s per meter, where c_s is in kilograms per cubic meter. At higher black-body temperatures, the value of k_a would be higher.

Annand [36] has applied this approach to a diesel engine. The apparent flame emissivity was related to the apparent mean absorptivity by

$$\varepsilon_a = 1 - \exp(-k_a l) \tag{12.34}$$

For Flynn's data, the peak emissivity is 0.8 which gives $k_a\, l = 1.6$. Since $l \approx 0.07$ m, this gives $k_a \approx 22\ \text{m}^{-1}$ and $c_s \approx 16\ \text{g/m}^3$ ($\approx 1\ \text{g/m}^3$ at NTP), which is a soot loading comparable with values measured in diesel engines during combustion (see Sec. 11.5.2).

Well-accepted prediction formulas for radiant heat flux in diesel engines are not available. Annand has proposed a radiation term of the form

$$\dot{q}_R = \beta\sigma(\overline{T}_g^4 - T_w^4) \tag{12.35}$$

where σ a is the Stefan-Boltzmann constant, $\overline{T}_g$ is the mean gas temperature, and T_w is the wall temperature. This term was coupled with a convective heat flux term to give a correlation for predicting total heat flux. In a first evaluation, when coupled with Eq. (12.14) , $\beta = 0.6$ was proposed.[8] In a later study with a modified convective heat-transfer correlation,[37] $\beta \approx 1.6$ was proposed.

12.6 MEASUREMENTS OF INSTANTANEOUS HEAT-TRANSFER RATES

12.6.1 Measurement Methods

Values of instantaneous heat flux into the combustion chamber walls can be obtained from measurements of the instantaneous surface temperature. The temperature variation in the wall is a result of the time-varying boundary condition at the gas/wall interface. It is damped out within a small distance (~1 mm) from the wall surface, so measurements must be made at the surface. Various types of thermocouple or thermistor have been used. [8] One-dimensional unsteady heat conduction into the wall is then assumed:

$$\frac{\partial T}{\partial t} = \frac{1}{\rho c}\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) \tag{12.36}$$

A sinusoidal variation with time of heat flux into a semi-infinite solid can be shown to produce a sinusoidal variation of surface temperature of the same frequency displaced in phase by 90°. The surface temperature T_w can be expressed as a Fourier series:

$$T_w = T_m + \sum_{n=1}^{N}[A_n \cos(n\omega) + B_n \sin(n\omega)] \tag{12.37}$$

where T_m is the time-averaged value of T_w, A_n and B_n are Fourier coefficients, n is a harmonic number, and ω is the angular frequency (radians per second). The boundary conditions are $T = T_w(t)$ at $x = 0$ and $T = T_l$ (constant) at $x = l$. The solution of Eq. (12.37) is[38]

$$T(x,t) = T_m - (T_m - T_l)\frac{x}{l} + \sum_{n=1}^{N}\exp(-\phi_n x)F_n(x,t) \tag{12.38}$$

where

$$F_n = A_n \cos(n\omega - \phi_n x) + B_n \sin(n\omega - \phi_n x)$$

and $\phi n = (n\omega/2\alpha)^{1/2}$, where α is the thermal diffusivity of the wall material $k/(\rho c)$.

The heat flux components at each frequency that caused that variation with time can be calculated via Fourier's law [Eq. (12.1)], and summed to give the total fluctuation of heat flux with time:

$$\dot{q} = \frac{k}{l}(T_m - T_i) + k\sum_{n=1}^{N}\phi_n[(A_n + B_n)\cos(n\omega) - (A_n - B_n)\sin(n\omega)] \qquad (12.39)$$

Alternative approaches for solving Eq. (12.36) are through use of an electrical analog to heat flow and by numerical methods. The latter become necessary if wall material properties depend significantly on temperature, as do combustion chamber deposits and some insulating ceramic materials. Several measurements of this type in spark-ignition and diesel engines have been made. A summary of these measurements can be found in Ref. 8. Radiant heat fluxes can be determined directly by several techniques: for example, photo-detector and infrared monochromator; thermocouple shielded by a sapphire window; pyroelectric thermal detector.

12.6.2 Spark-Ignition Engine Measurements

Figure 12.11 shows the surface temperature variation with crank angle, and the heat flux variation calculated from it, on the cylinder head of a spark-ignition engine at a part-load low-speed operating condition. The swing in surface temperature at this point (about halfway from the on-cylinder-axis spark plug to the cylinder wall) is 7 K. The heat flux rises rapidly when the flame arrives at the measurement location, has its maximum at about the time of peak cylinder pressure when burned gas temperatures peak (see Sec. 9.2.1), and then decays to relatively low levels by 60° ATC as expansion cools the burned gases. Peak heat fluxes on the cylinder head of 1 to 3 MW/m ^{2} were measured over the normal engine speed and load range. [39, 41]

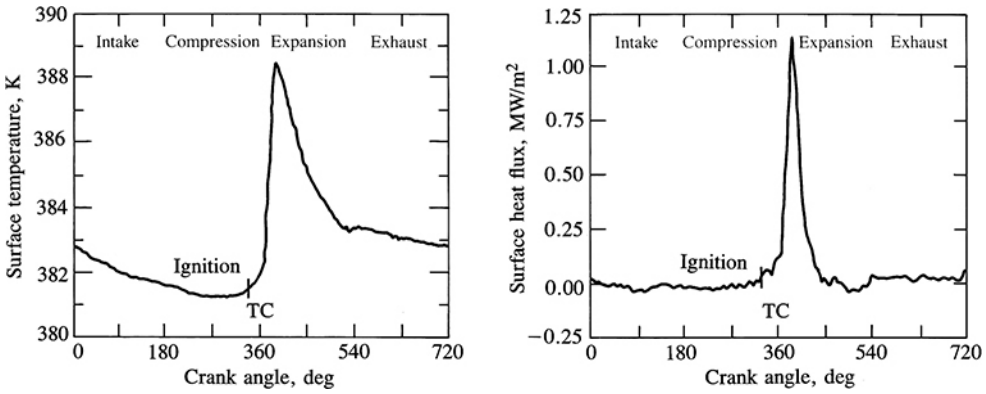


Figure 12.11 Surface temperature measured with thermocouple in cylinder head, and surface heat flux calculated from that surface temperature, as a function of crank angle. Spark-ignition engine operated at part-load. [39]

The heat flux profiles vary with location in a given engine, and also from one cycle to the next. (Figure 12.7 showed typical cycle-to-cycle variations for a DI diesel.) The rapid rise in heat flux depends on the flame arrival time at the measurement location. As a consequence, the heat fluxes determined from surface temperature data averaged over many cycles usually show a less rapid rise rate. Figure 12.12 also shows the effect of spark retard at a given location. The later arrival of the flame at the measurement location delays the start of the rapid rise in heat flux. Also, the magnitude of the peak heat flux rate is reduced as combustion is retarded since peak pressure is later and lower, and the peak burned gas temperature is consequently lower, too.

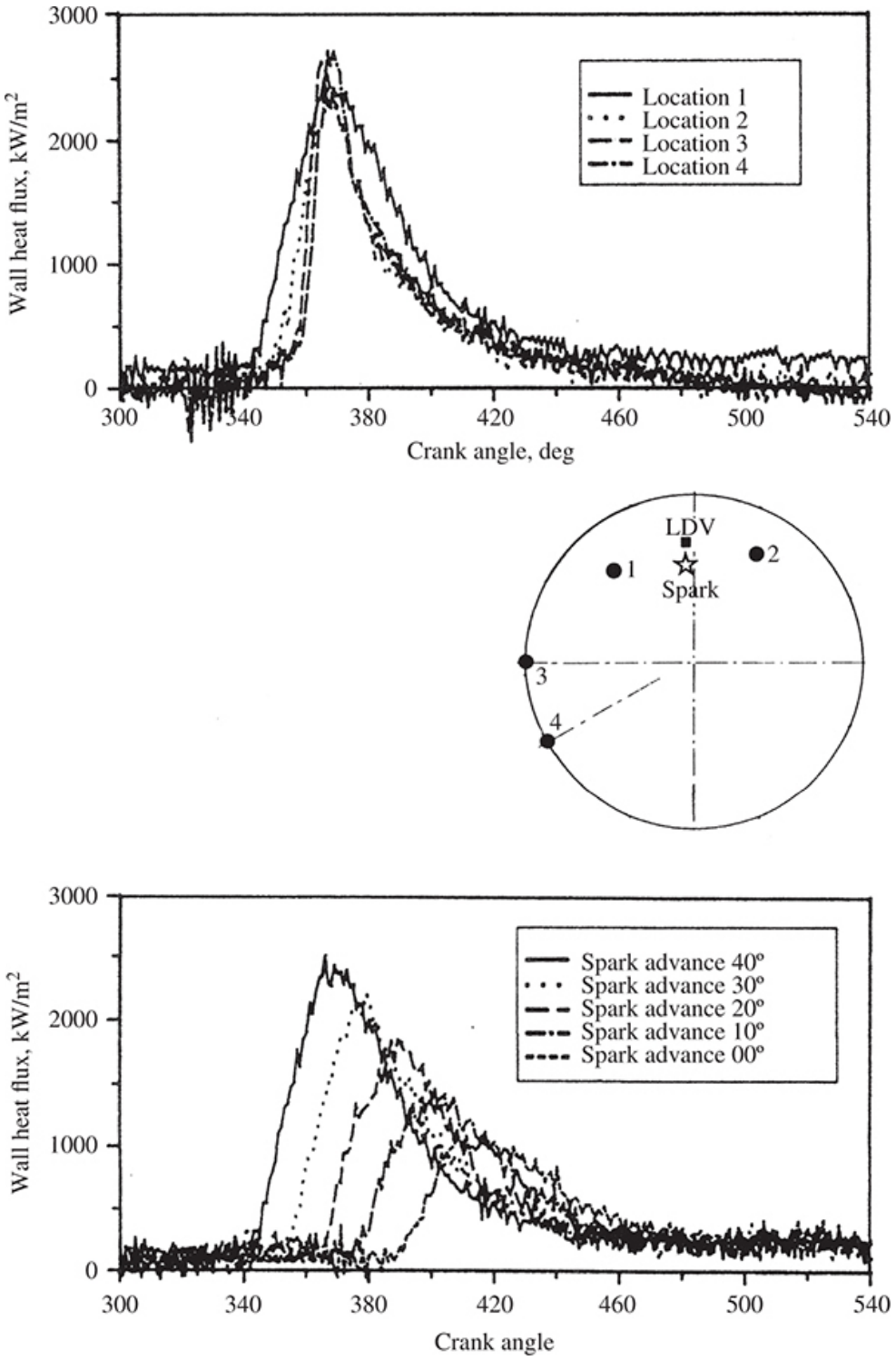


Figure 12.12 (*a*) Variation of surface heat flux with crank angle at four temperature measurement locations in the cylinder head of a naturally-aspirated spark-ignition engine: speed 1000 rev/min, stoichiometric operation, volumetric efficiency 0.8, spark advance 40°BTC. (*b*) Effect of

retarding spark timing from 40° BTC to TC in 10° CA intervals on heat flux profile, location 1, same spark-ignition engine operating conditions as in (*a*). [40]

Figure 12.13 shows the effects of engine load (defined by the engine's volumetric efficiency) and speed on heat flux profile. Such measurements show that increasing engine load and increasing engine speed increase the surface heat flux. Maximum heat fluxes occur with close-to-stoichiometric mixtures. [39–41] All these trends would be expected from the variations in burned gas temperature that result from these changes in engine operation.

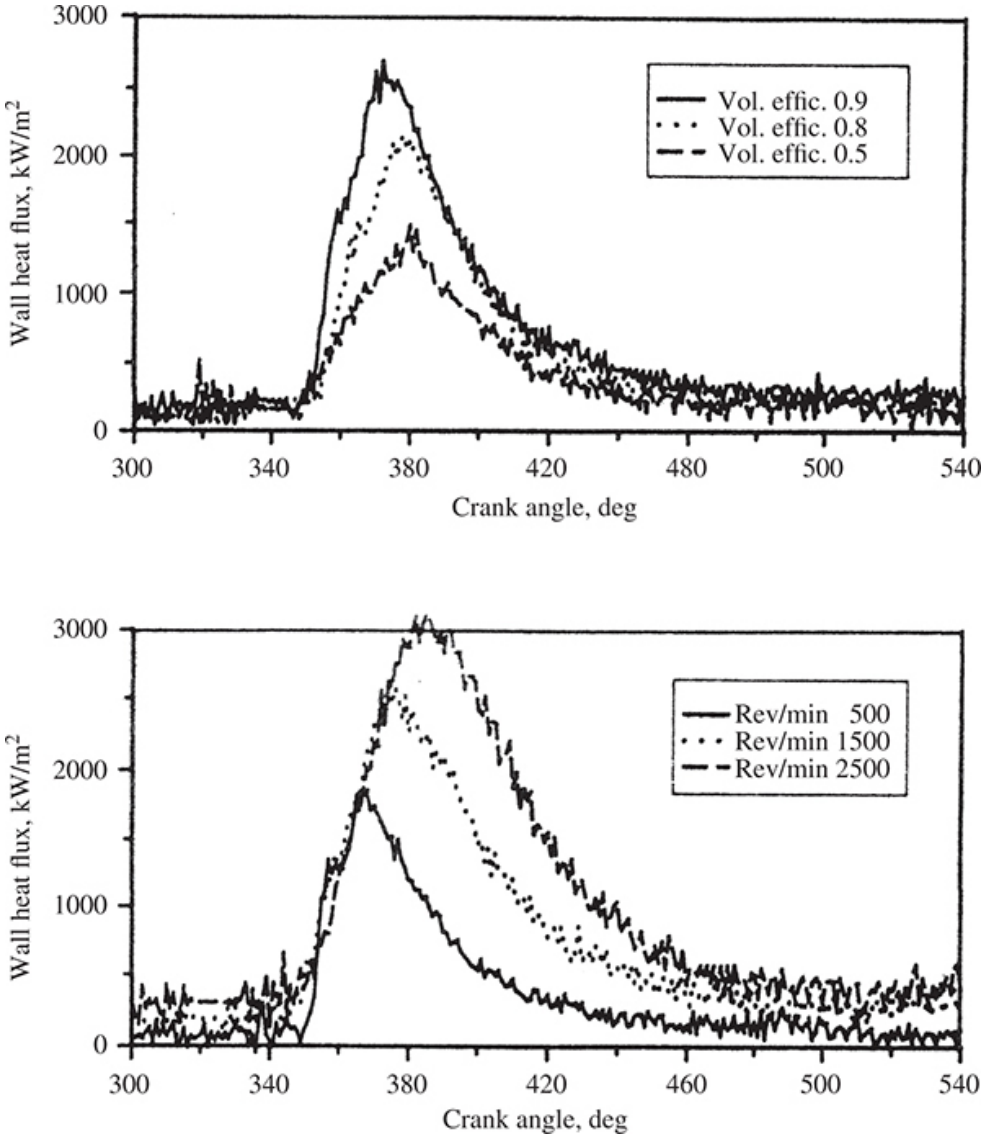


Figure 12.13 Variation of surface heat flux with engine load (volumetric efficiency) and speed: n_v = 0.5 (mid-load), 0.8, and 0.9 (high-load); speed (rev/min), 500, 1500, 2500, location 1. Same spark-ignition engine as in Fig. 12.12, and same base operating conditions. [40]

12.6.3 Diesel Engine Measurements

Measurements of instantaneous heat fluxes in diesel engines show similar features. The heat flux distribution over the combustion chamber walls is usually highly nonuniform. There are also variations between the heat flux profiles from individual cycles. Figure 12.14 shows surface heat fluxes at two locations in a medium-swirl DI diesel, one over the piston bowl (higher heat flux) and the other over the piston squish area, in relation to the heat-release profile. The heat flux increases rapidly once combustion starts, reaches a maximum at close to the time of maximum cylinder pressure, and decreases to a low value by 40 to 60° ATC. Peak heat fluxes to the primary combustion chamber surfaces (the piston bowl and head directly above the bowl) are of order 10 MW/m^2.

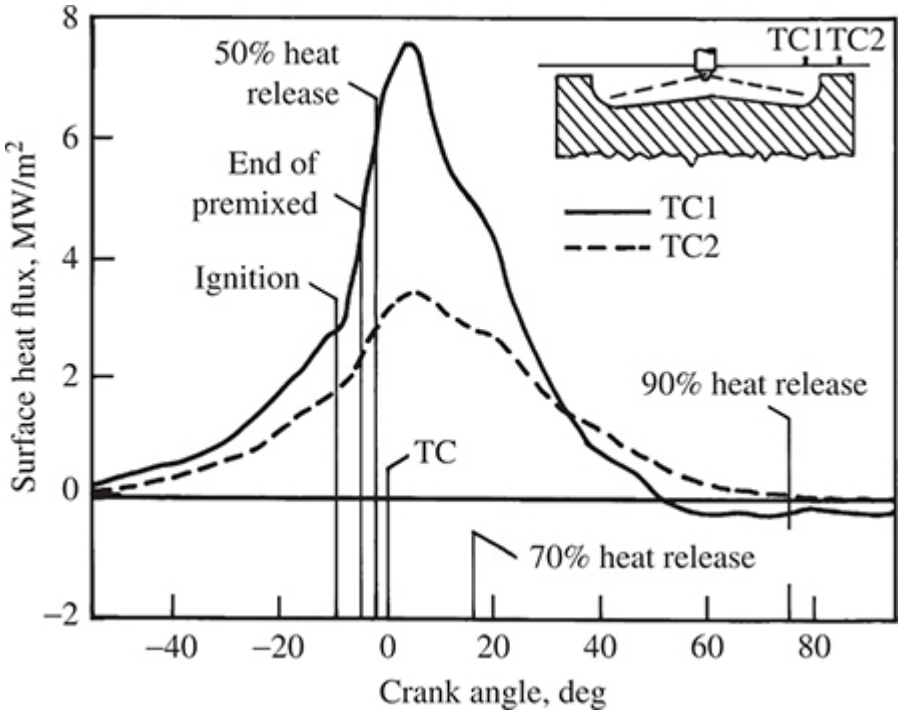


Figure 12.14 Measured surface heat flux at two locations in the cylinder head of a medium-swirl DI diesel engine. TC1 above the piston bowl, TC2 above the piston squish area, as shown. Percentages of heat release are indicated. Bore = stroke = 114 mm, r_c = 16, 2000 rev/min, overall equivalence ratio = 0.5, intake pressure = 1.5 arm. [42]

In smaller diesel engines with swirl, the mean heat flux to the piston within the piston bowl is usually higher than the mean heat fluxes to the cylinder head and the annular squish portion of the piston crown (by about a factor of two). [16, 37] This would be expected since the piston bowl is the zone where most of the combustion takes place and gas velocities are highest. There are, in addition, substantial variations in heat flux at different locations within the piston bowl, on the head, and on the annular region of the piston crown.

In contrast to the above results obtained on smaller high-speed (7- to 9-cm bore) diesels with swirl with deep bowl-in-piston combustion chambers of diameter about half the piston diameter, results from tests on a medium-speed 30-cm bore quiescent shallow-bowl piston direct-injection supercharged diesel showed a much more uniform heat flux distribution over the combustion chamber walls. [43]

Heat fluxes to the cylinder liner are much lower still (an order of magnitude less than the peak flux to the combustion chamber surface) and are also nonuniform. Figure 12.15 compares heat fluxes to the cylinder head with three locations along the liner. Even at the top of the liner, the peak heat flux is only 15 to 20% of the flux to the primary combustion chamber surfaces. Again this would be expected, since the combustion gases do not contact the lower parts of the cylinder wall until later in the expansion stroke when their temperature is much below their peak value.

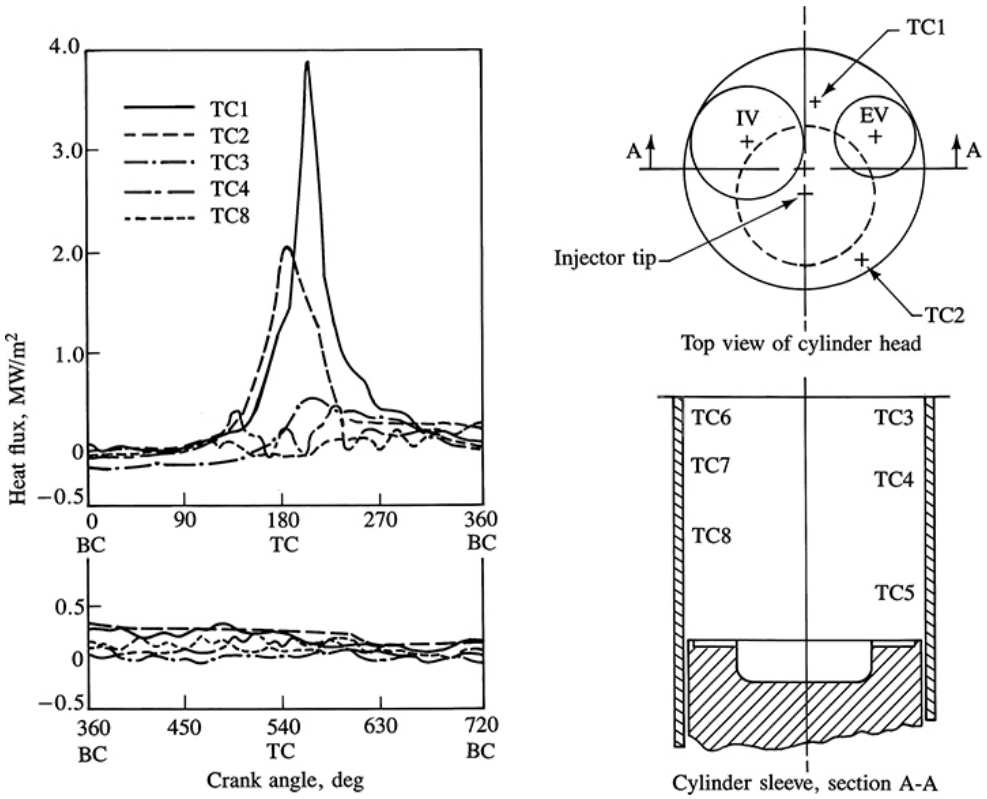


Figure 12.15 Measured surface heat fluxes at different locations in cylinder head and liner of naturally-aspirated four-stroke cycle DI diesel engine. Bore = stroke = 114 mm, 2000 rev/min, overall equivalence ratio = 0.45. [15]

Figure 12.16 shows examples of radiant heat flux measured above the piston bowl of a medium-swirl DI diesel engine as a function of engine speed and load.[16] The limited radiant heat flux data available exhibit the following trends. The rapid increase in radiant heat flux following combustion is delayed relative to the start of pressure rise due to combustion by about 5°; this delay increases with increasing speed. This delay occurs because the radiation results from soot burn up in the second diffusion-flame phase of the diesel combustion process. The peak radiant flux remains approximately constant with increasing equivalence ratios up to $\phi \approx 0.5$. Further increases in the equivalence ratio produce a drop in level of radiant flux. The time-averaged radiant heat flux increased approximately linearly with increasing manifold pressure; however, peak radiative flux levels remained essentially unchanged. Peak and time-averaged values of the

radiant heat flux decreased as injection timing was retarded due to lower flame temperatures.

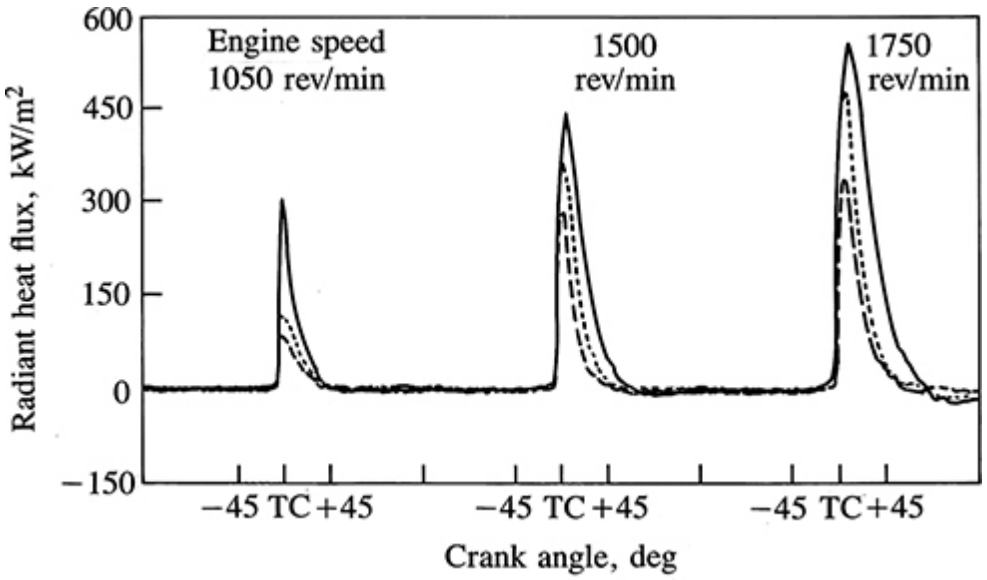


Figure 12.16 Measured radiant heat flux to cylinder head above the piston bowl in a high-swirl DI diesel engine when load and speed are varied. Solid curve: 80% load. Dotted curve: 40% load. Dashed curve: no load. [16]

In diesel engines, the relative importance of radiant heat transfer (as a percentage of the total heat flux) depends on the location on the combustion chamber surface, crank angle, engine load, engine size, and engine design details. The time-averaged radiant heat transfer increases as a proportion of the total heat transfer, with increasing load, as indicated in Fig. 12.17. [8] At high load, the radiant heat flux is between 25 and 40% of the total time-averaged heat flux.

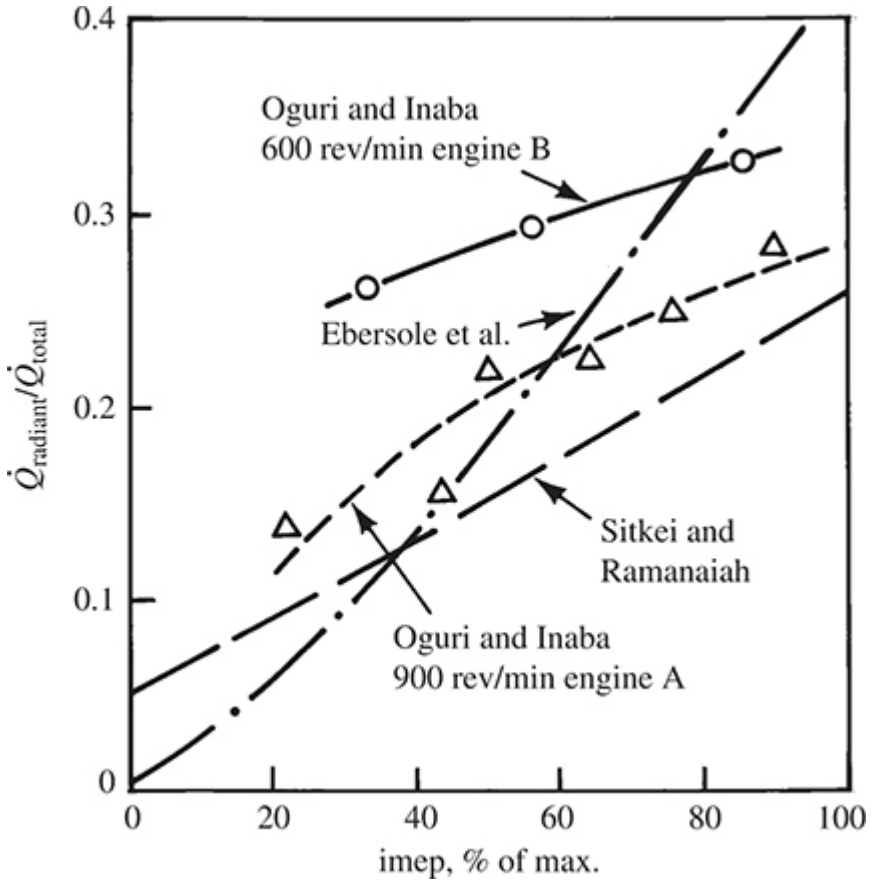


Figure 12.17 Radiant heat flux as fraction of total heat flux over the load range of several different diesel engines. See Ref. 8 for sources of engine data.

12.6.4 Evaluation of Heat-Transfer Correlations

The available convective (or combined convective plus radiative) heat flux correlations have been compared with instantaneous engine heat-transfer measurements. One difficulty in such evaluations is the determination of spatially-averaged combustion-chamber heat fluxes from the experimental data for comparison with correlations intended to predict the mean chamber heat flux as a function of crank angle. Area-averaged instantaneous heat flux prediction using Woschni's equation (12.19) for the spark-ignition engine conditions have generally been found to be similar in magnitude to measured heat fluxes to the cylinder head (e.g., Ref. 44).

Comparisons have been made of predictions with data for diesel engines. Annand's correlations, Eqs. (12.14) and (12.35) with $a = 0.06$, $b = 0.85$, and

β (for the combustion phase only) = 0.57, gave reasonable agreement with instantaneous cylinder head heat fluxes, and overall time-averaged heat fluxes for a medium-speed quiescent DI engine design.[43] In a small high-speed diesel with swirl, values of $a = 0.13$, $b = 0.7$, and $\beta = 1.6$ gave a reasonable fit to estimates of the instantaneous area-mean heat flux[37] and time-averaged heat flux to the piston.[45] Comparisons of the Annand and Woschni correlations generally show that the Annand correlation predicts somewhat higher heat fluxes at the same crank angle.[10] The most careful comparison of all three correlations summarized in Secs. 12.4.3 and 12.4.4 with experimental data has been made by Dent and Sulaiman in a small high-speed diesel engine with swirl.[16] The mean experimental heat flux was estimated from a number of thermocouple measurements located at different points around the combustion chamber surface. Figure 12.18 shows the comparison. The Woschni correlation is lower than the others at light load because the combustion-induced velocity term is smaller. Expansion stroke heat fluxes are underpredicted by all three correlations. Given the uncertainty in converting the measurements to an average heat flux value, the agreement is reasonably good.

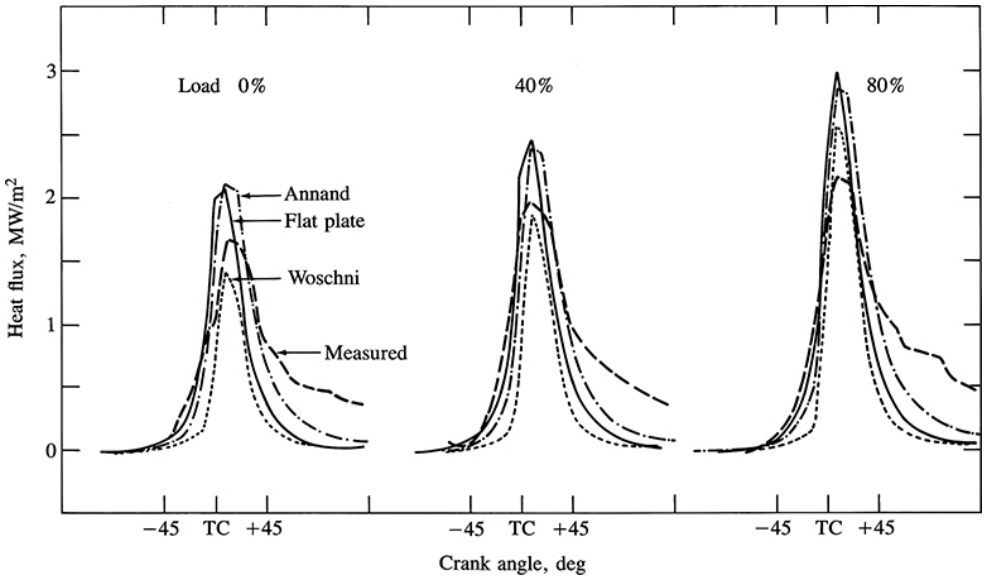


Figure 12.18 Comparison of measured mean heat fluxes on the cylinder head at 1050 rev/min in a fired high-swirl DI diesel engine with various prediction equations: Annand, Eq. (12.14) ; Woschni, Eq. (12.19) ; flat plate, Eq. (12.20) using measured gas motion.[16]

Dent examined additional modifications to the flat plate formula [Eq. (12.20)], which was based on the cylinder-mean gas temperature. During combustion a two-zone model is more appropriate. Assuming an equivalence ratio for the burned gas (Dent assumed stoichiometric), a combustion zone temperature can be determined from the relation

$$\bar{T}_b = \frac{m\bar{T}_g - m_a T_a}{m_b}$$

where m_a is the mass of air, T_a is the air temperature, and m_b is the burned gas mass, $m_b = m - m_a$. In addition, the observed swirl enhancement which occurs due to the combustion was included by multiplying the swirl velocity used in the heat-transfer correlation by the square root of the ratio (density of air in the motored case)/(density of burned gas in fired case). The combination of both effects (see Fig. 12.19) improves the shape of the predictions by broadening and lowering the peak.

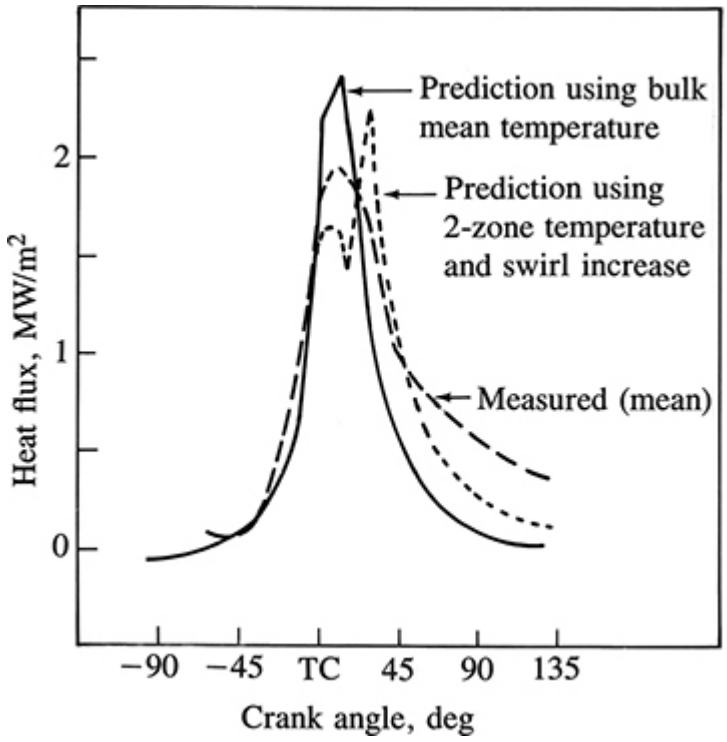


Figure 12.19 Measured heat fluxes on cylinder head in fired high-swirl DI diesel engine at 1050 rev/min and 40% load compared with predictions

based on bulk mean gas temperature and using temperatures based on two-zone (air and burned gas) model.[16]

Each of the convective-heat-transfer correlations described has experimental support. However, under engine design and operating conditions different from those under which they were derived, the predictions should be viewed with caution. Woschni's correlation is the correlation used most extensively for predicting spatially averaged instantaneous convective heat fluxes. It has been shown that average heat transfer rates calculated using Woschni's correlation compare well in an SI engine with heat loss values estimated from an engine energy balance. [46] However, the empirical constants which relate the mean piston speed and combustion-induced gas motion to the velocity used in the Reynolds number, determined by Woschni, will not necessarily apply to all the different types of engine. If local velocities are known, the flat-plate-based correlations provide the best available approach. Annand's correlation has the advantage of being the simplest correlation to use. Since the radiation component in diesel engine heat transfer is normally 20 to 40% of the total, an approximate estimate of its value may suffice. A more extensive review of these heat transfer correlations and comparisons with experimental data on heat transfer to the engine's combustion chamber walls [44] reaches similar conclusions. While the functional forms of these correlations (based on the relevant dimensionless numbers) are well grounded, they are inherently empirical and conceptually over simplified. The combustion, thermal, and fluid flow phenomena occurring inside the engine's cylinders are extremely complex, creating significant modeling and measurement challenges.

12.6.5 Boundary-Layer Behavior

Measurements of thermal boundary-layer thickness in an operating spark-ignition engine have been made using schlieren photography in a flow-visualization engine. Figure 12.20 shows one set of measurements on the cylinder wall in the clearance volume opposite the spark plug. The boundary-layer thickness decreases during intake and increases steadily during compression and expansion to about 2 mm. It stops growing and becomes unstable during the exhaust process, separating from the cylinder

wall and becoming entrained into the bulk gas leaving the cylinder. The thickness of the thermal boundary layer varies substantially at different locations throughout the chamber. While the trends with crank angle were similar, the layers on the cylinder head and piston crown were significantly (up to two to three times) thicker during compression and expansion in the simple disc-shaped chamber studied. [47] This different behavior probably results because there is no bulk flow adjacent to the head and piston crown.

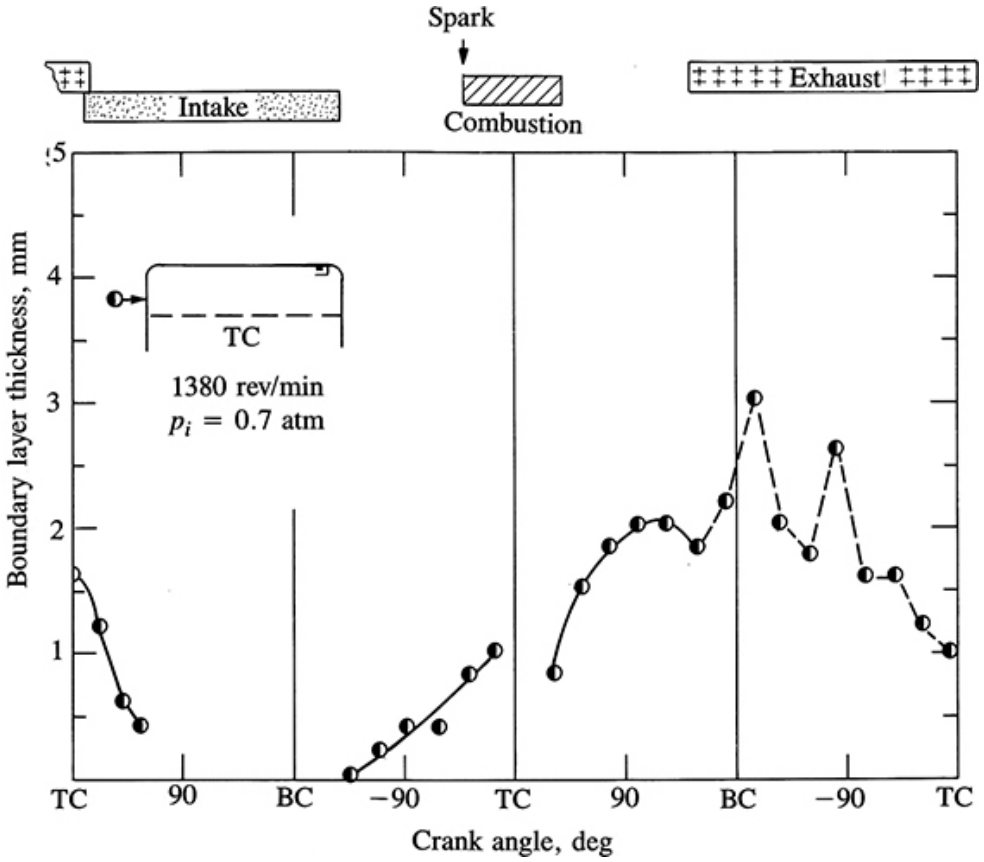


Figure 12.20 Thermal boundary-layer thickness, at the top of the cylinder wall in the clearance volume, determined from schlieren photographic measurements in a special visualization square-piston spark-ignition engine.[47]

Estimates of thermal boundary-layer thickness in spark-ignition engines, based on convective-heat-transfer correlations and thermal energy conservation for the growing layer give thicknesses comparable to these measurements. Note that a substantial fraction of the cylinder mass is

contained within the thermal boundary layer. For example, for an average thickness of 3 mm at 90° ATC during expansion, the volume of the boundary layer is 20% of the combustion chamber volume for typical engine dimensions. Since the average density in the boundary layer is about twice that in the bulk gases, some 30 to 40% of the cylinder mass would be contained within this boundary layer.

12.7 THERMAL LOADING AND COMPONENT TEMPERATURES

The heat flux to the combustion chamber walls varies with engine design and operating conditions. Also, the heat flux to the various surfaces that bound the combustion chamber is not uniform. This nonuniform heat flux, combined with the different thermal impedances between locations on the combustion chamber surface and the cooling fluid, results in different surface temperatures and temperature distributions within the engine's components. This section reviews the variation in heat flow into the components that comprise the combustion chamber, and the temperature distribution in these components.

12.7.1 Effect of Engine Variables

The following variables affect the magnitude of the heat flux to the different surfaces of the engine combustion chamber and thus the temperature distribution in the components that bound the chamber: engine speed; engine load; overall equivalence ratio; compression ratio; spark or injection timing; swirl and squish motion; mixture inlet temperature; coolant temperature and composition; wall material; wall deposits. Of these variables, speed and load have the greatest effect. Equation (12.19) , derived from the Nusselt-Reynolds number relation,

$$h_c = \text{constant} \times B^{-0.2} p^{0.8} T^{-0.55} w^{0.8}$$

and the relation for heat-transfer rate per unit area, Eq. (12.2) ,

$$\dot{q} = h_c(T - T_w)$$

are useful for predicting these thermal trends as engine operating and design variables change. The temperature T in the above two equations is the average temperature of the in-cylinder gases. This choice is made for pragmatic reasons: the average gas temperature is readily obtained from the cylinder pressure (which can be measured or calculated), the ideal gas law ($pV = mR\bar{T}$), and the mass in the cylinder. However, heat transfer to the combustion chamber walls is most significant when the hot burned gases contact or "wet" these walls. Figure 12.21 illustrates the situation in a spark-ignition engine where, during combustion, the burned gases behind the propagating flame contact only part of the cylinder head and piston crown. The heat transfer from the burned gas zone is given by:

$$\dot{Q}_b = \sum_i A_{ci,b} h_b (T_b - T_{w,i}) \tag{12.40}$$

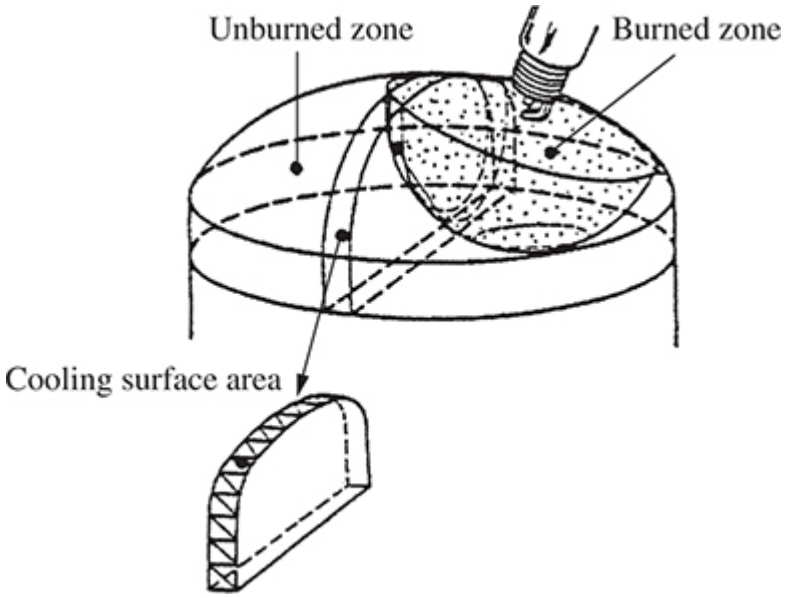


Figure 12.21 Illustration of heat transfer process during combustion in a spark-ignition engine. The burned gases are hot (~2500 K); the unburned gases are significantly cooler (400 to 800 K). The area wetted by the burned gases and unburned gases are shown, and can be calculated as indicated to obtain the burned zone and unburned zone heat fluxes [see Eqs. (12.40) and (12.41)].

where $A_{ci,b}$ is the area of the combustion chamber surface i wetted by the hot burned gases at temperature T_b. The heat transfer from the unburned gas zone is

$$\dot{Q}_u = \sum_i A_{ci,u} h_u (T_u - T_{w,i}) \tag{12.41}$$

where $\dot{Q}_u$ is the heat transfer from the unburned zone, $A_{ci,u}$ is the area of the combustion chamber wetted by the unburned gas, denoted by the subscript u. Other symbols have similar definitions to those in Eq. (12.40) but for the unburned gas zone. Because T_b is some four times higher than T_u (in K), the burned zone heat flux is much larger than the unburned zone heat flux.

This fundamentally-based description of the engine heat transfer process is the appropriate model to use to assess heat transfer effects of changes in engine design and operating variables. Figure 12.22 shows a set of results obtained with this two-zone combustion and heat transfer model in a cycle simulation of a spark-ignition engine at mid-speed and high load. As combustion proceeds, the mass fraction burned x_b, rises from zero at time of spark to approach unity at about 30° CA ATC, the cylinder pressure rises, as do the gas temperatures in the unburned and burned gas zones. The heat transfer rate rises from a low value before the spark discharge (unburned mixtures are at relatively low temperatures, less than about 600 K) to a peak value (more than an order of magnitude higher) at time of peak pressure and burned gas temperature (some 2500 K) at about 15° CA ATC, and then decays to about one-quarter of the peak rate by about 60° CA ATC. During the early stages of combustion, the total unburned gas wetted area is much larger than the burned gas wetted area; see Fig. 12.21. During combustion, the burned gas density is one-fourth that of the unburned (much lower temperature) gas: this wetted area ratio (unburned/burned) transitions through unity to values much less than one. Thus the changing relative size of $A_{c,u}$ and $A_{c,b}$ offsets the substantial difference between (T_b - T_w) and (T_u - T_w) in Eqs. (12.40 and 12.41). This is a major reason why heat transfer estimates based on the average temperature of the in-cylinder gases, and the total combustion chamber surface area, do not differ that much from the two-zone model estimates such as the one shown in Fig. 12.22. (Note that the boundary

layer thickness grows to some 3% of the cylinder bore of 80 to 90 mm, i.e. a few mm, which is consistent with the measurements shown in Fig. 12.20.) [e]

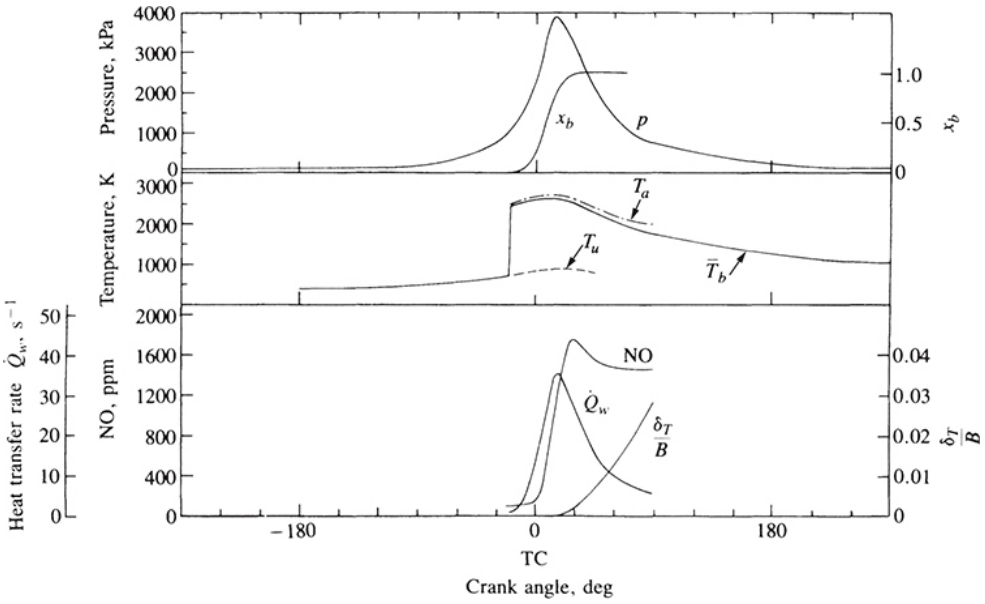


Figure 12.22 Cylinder pressure p, mass fraction burned x_b, unburned and burned gas temperatures (T_u = unburned, T_a = adiabatic burned core, $\bar{T}_b$ = mean burned gas temperatures), heat-transfer rate $\dot{Q}_w$ (normalized by fuel flow rate × heating value), thermal boundary-layer thickness δ_T and mean nitric oxide concentration in the burned gases, through a four-stroke engine operating cycle, predicted by thermodynamic-based cycle simulation. 5.7-dm^3 displacement eight-cylinder engine operating at wide-open throttle, 2500 rev/min, with equivalence ratio = 1.1. Gross indicated mean effective pressure is 918 kPa and specific fuel consumption is 254 g/kW · h. [48]

With this fundamental description of the of the loss of thermal energy from the hot burned gases to the combustion chamber walls, the effect of changes in the above listed variables on engine and component heat flux will now be summarized. The comments which follow apply primarily to spark-ignition engines. In compression-ignition engines, the basic geometry of the hot burned gas zones and the air are different, and distribution of heat flux and temperature varies greatly with the size of cylinder and thus its combustion chamber, and with the in-cylinder flow.

Speed, Load, and Equivalence Ratio

The convective wall heat flux varies significantly with engine speed and load. Measured time-resolved values (in kW/m ^{2}) taken in a gasoline spark-ignition engine are shown in Figs. 12.12 and 12.13. [40] The heat flux profiles at the four measuring locations show comparable peak values though this is not always the case. The load scaling (based on volumetric efficiency: 90% corresponds to full-load; 50% would be about half load) and the speed scaling, in terms of heat transfer rate, are significant, as Eq. (12.19) indicates. The gas pressure (load) and speed scaling have exponents of 0.8.

Cycle simulation predictions of spark-ignition engine combustion chamber heat transfer (per cycle) as a function of speed and load are shown in Fig. 12.23. The cycle heat transfer is expressed as a percent of the fuel's chemical energy (mass of fuel × Q_{LHV}). The heat transfer to the total combustion chamber surface (excluding the exhaust port) was calculated using a thermodynamic-based cycle simulation (see Sec. 14.4). The relative importance of heat losses *per cycle* decreases as speed and load increase (Fig. 12.23): the *average* heat transfer *per unit time*, however, increases as speed and load increase (Fig. 12.13).

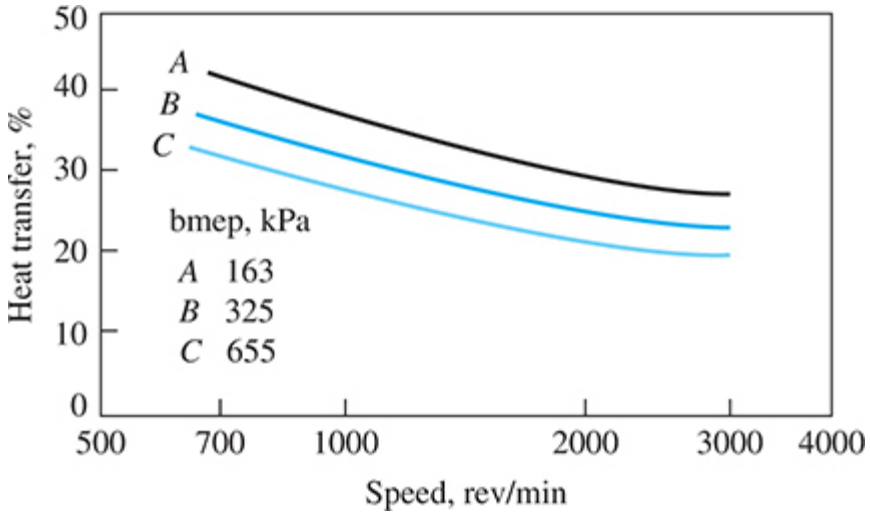


Figure 12.23 Predicted average heat-transfer rate (as percent of fuel flow rate × Q_{LHV}) to combustion chamber walls of an eight-cylinder 5.7-dm^3 spark-ignition engine as a function of speed and load. Stoichiometric operation; MBT timing. [48]

Since speed and load affect p, T, and w in Eq. (12.19) , simpler correlations have been developed to predict component heat fluxes directly from experimental data. Time-averaged heat fluxes at several combustion chamber locations, determined from measurements of the temperature gradient in the chamber walls, have been fitted with the empirical expression

$$\dot{q} = \text{constant} \times \left(\frac{\dot{m}_f}{A_p} \right) \tag{12.42}$$

with n between 0.5 and 0.75 (the value of n depending on engine type and location within the combustion chamber and A_p the piston crown area). Results for several spark-ignition and diesel engine designs, with appropriate values of n, can be found in Refs. 49 to 51. While this correlation is not dimensionless and is different from Eq. (12.19) , it provides a convenient method for scaling experimental heat flux data. The heat flux to the cylinder head and liner for a spark-ignition engine were well correlated by Eq. (12.42) with $n = 0.6$. The heat flux distributions over the cylinder head at a fuel flow rate per unit piston area of 0.195 kg/s · m ^{2} for several different DI diesel engines were comparable in magnitude. The effect of speed at wide-open throttle on component temperatures for a spark-ignition engine can also be found in Ref. 51. Exhaust valve, piston crown and top ring groove, and nozzle throat temperatures for a Comet prechamber diesel as a function of fuel flow rate can be found in Ref. 49.

The peak heat flux in an SI engine occurs at the mixture equivalence ratio for maximum power, $\phi \approx 1.1$, and decreases as ϕ is leaned out or enriched from this value. The major effect is through the gas temperature in Eqs. (12.2) and (12.19). However, as a fraction of the fuel's chemical energy, the heat transfer per cycle is a maximum at $\phi = 1.0$ and decreases for richer and leaner mixtures as shown by the thermodynamic-based cycle-simulation predictions in Fig. 12.24.

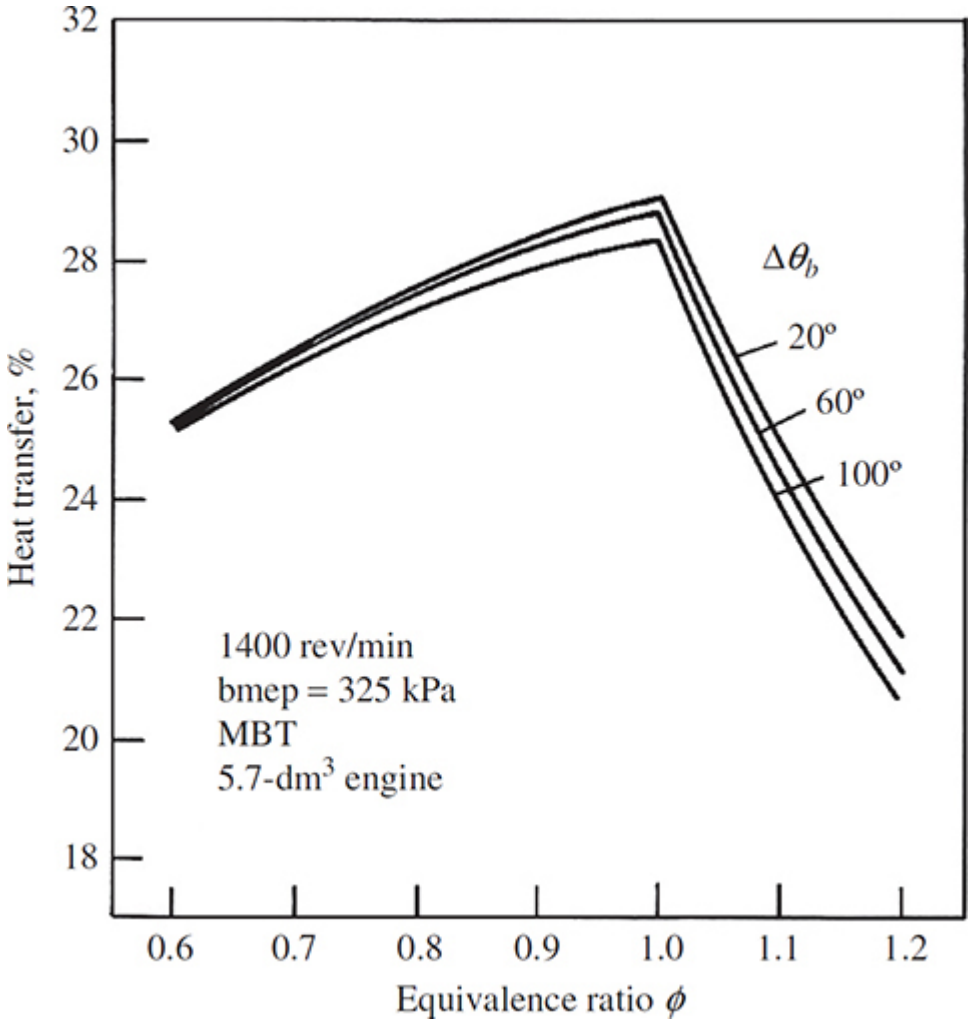


Figure 12.24 Predicted average heat-transfer rate (as percent of fuel flow rate × Q_{LHV}) to combustion chamber walls of an eight-cylinder 5.7-dm^3 spark-ignition engine as a function of equivalence ratio and burn rate ($\Delta\theta_b$ = combustion duration).[48]

Measurements and predictions of the effects of variations in engine speed and load in a DI diesel (with a 2.3 ℓ cylinder size) show similar trends to SI engines. Significant variations with combustion chamber surface location were observed, with the piston bowl and cylinder head above the bowl showing higher peak heat fluxes do to the proximity of these surfaces to the hot combustion gases with these surfaces. One expects higher convective and radiative heat transfer rates in these locations. In CI engines, the overall

air/fuel ratio variation is incorporated directly in the load variation effects. [52]

Compression Ratio

Increasing the compression ratio in an SI engine decreases the total heat flux to the coolant until $r_c \approx 10$; thereafter heat flux increases slightly as r_c increases. [53] The magnitude of the change is modest; for example, a 10% decrease in the maximum heat flux (at the valve bridge) occurs for an increase in r_c from 7.1 to 9.4. [49] Several gas properties change with increasing compression ratio (at fixed throttle setting): cylinder gas pressures and peak burned gas temperatures increase; gas motion increases; combustion is faster; the surface/volume ratio close to TC increases; the gas temperature late in the expansion stroke and during the exhaust stroke is reduced. Measured mean exhaust gas temperatures confirm the last point, which probably dominates the trend at lower compression ratios. As the compression ratio increases further, the other factors (which all increase heat transfer) become important.

The effect of changes in compression ratio on component temperatures depends on location. Generally, head and exhaust valve temperatures decrease with increasing compression ratio, due to lower late-expansion and exhaust stroke temperatures. The piston and spark plug electrode temperatures increase, at constant throttle setting, due to the higher peak combustion temperatures at higher compression ratios. If knock occurs (see Sec. 9.6), increases in heat flux and component temperatures result; see below.

Spark Timing

In a naturally-aspirated SI engine, retarding the spark timing decreases the heat flux as shown in Figs. 12.12*b* and 12.25. A similar trend in CI engines with retarding injection timing would be expected. The burned gas temperatures are decreased as timing is retarded because combustion occurs later when the cylinder volume is larger. Surface temperature trends vary with component. Piston and spark plug electrode temperatures change most with timing variations; exhaust valve temperature increases as timing is retarded due to higher exhausting gas temperatures. [49] In turbocharged SI engines, the impact of retarding combustion is more complex because the

engine's exhaust gas temperature increases, thus driving the turbocharger faster. Depending on how the engine is being controlled, this could increase the compressor boost level and airflow. Depending on the intercooler performance, the engine intake air temperature may then increase, resulting in an increase in burned gas temperature. However, the convective heat transfer to the combustion chamber walls is driven by the difference between the in-cylinder gas temperature and the wall temperatures. Most important, therefore, are the higher burned gas temperatures attained. These temperatures are not that strongly influenced by engine operating condition charges.

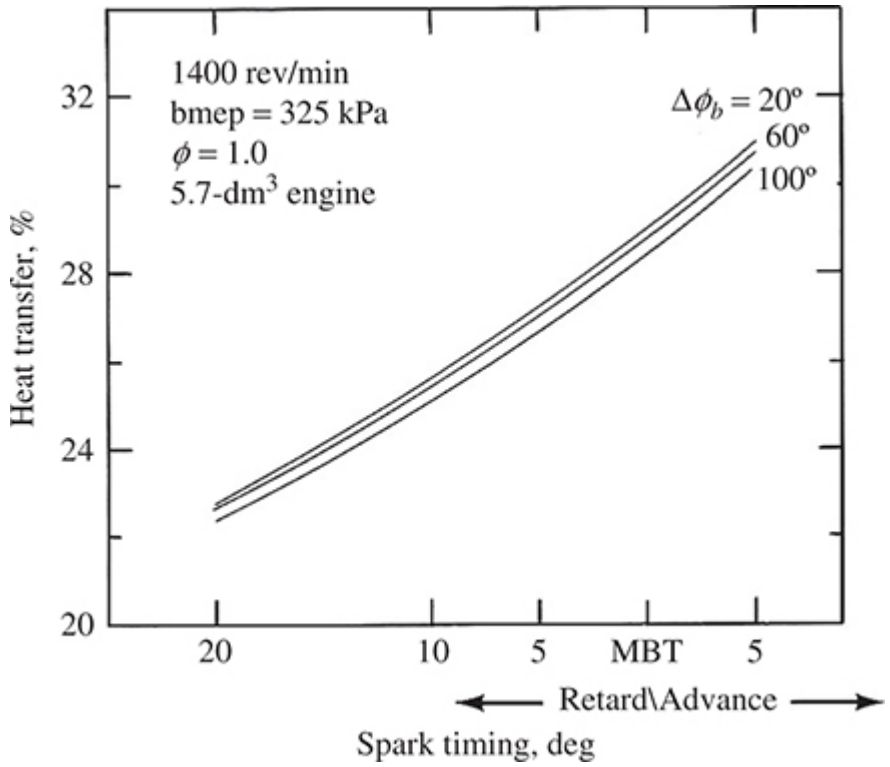


Figure 12.25 Predicted average heat-transfer rate (as percentage of fuel flow rate × Q_{LHV}) to combustion chamber walls of an eight-cylinder 5.7-dm^3 spark-ignition engine as a function of spark timing and burn rate ($\Delta\theta_b$ = combustion duration).[48]

Swirl and Squish

Increased gas velocities, due to swirl or squish motion, will result in higher heat fluxes. Equation (12.19) indicates that the effect on local heat flux,

relative to quiescent engine designs, should be proportional to local gas velocity raised to the 0.8 power. Experimental evidence supports this trend. [41]

Inlet Temperature

The heat flux increases with increasing inlet temperature since the gas temperatures throughout the cycle are increased. An increase of 100 K gives a 13% increase in heat flux. [40]

Coolant Temperature and Composition

Increasing coolant temperature increases the temperature of components directly cooled by the liquid coolant. Figure 12.26 shows the result of a 50-K rise in coolant temperature in a spark-ignition engine. The exhaust valve and spark plug temperatures are unchanged. The smaller response of the metal temperatures to coolant temperature change occurs at higher heat flux locations (such as the valve bridge), and indicates that heat transfer to the coolant has entered the nucleate-boiling regime in that region. The response is greater where heat fluxes are lower (e.g., the cylinder liner), indicating that there heat transfer to the coolant is largely by forced convection. When nucleate boiling occurs (i.e., when steam bubbles are formed in the liquid at the metal surface, although the bulk temperature of the coolant is below the saturation temperature), the metal temperature is almost independent of coolant temperature and velocity.

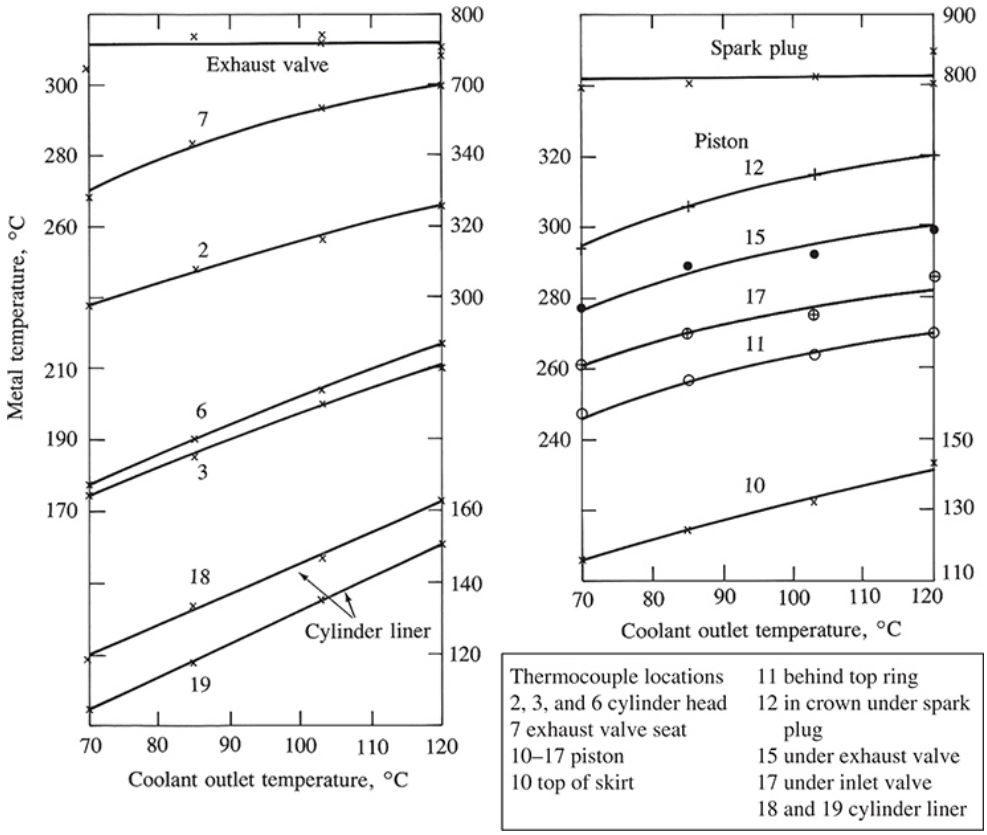


Figure 12.26 Effect of coolant temperature on cylinder head, liner, exhaust valve, valve seat, piston, and spark plug metal temperatures. Spark-ignition engine at 5520 rev/min and wide-open throttle. $r_c = 8.5$.[49]

The impact of adding nucleate boiling heat transfer to forced-convection heat transfer where coolant passage surface temperatures are high (e.g., between adjacent exhaust valve seats in the cylinder head) are illustrated in Fig. 12.27. [54] The heat flux measurements shows the influence of cooling water flowing at a given velocity (0.1 to 1 m/s) over a wall with various heat fluxes flowing through the wall surface. The resulting surface temperature was measured. The coolant water temperature was 95°C. As the heat flux rises, the wall temperature rises linearly at each flow velocity until about 125°C: the rate of increase in wall temperature with increasing heat flux then roughly halves. Higher heat fluxes can be accommodated with significantly lower increases in wall surface temperatures. The occurrence of nucleate boiling this moderates the impact of higher heat fluxes. However at much higher heat fluxes, nucleate boiling can produce so much steam that it forms a

vapor film which then acts as an insulating layer adjacent to the wall surface and forces the surface temperature to much higher and potentially damaging values.

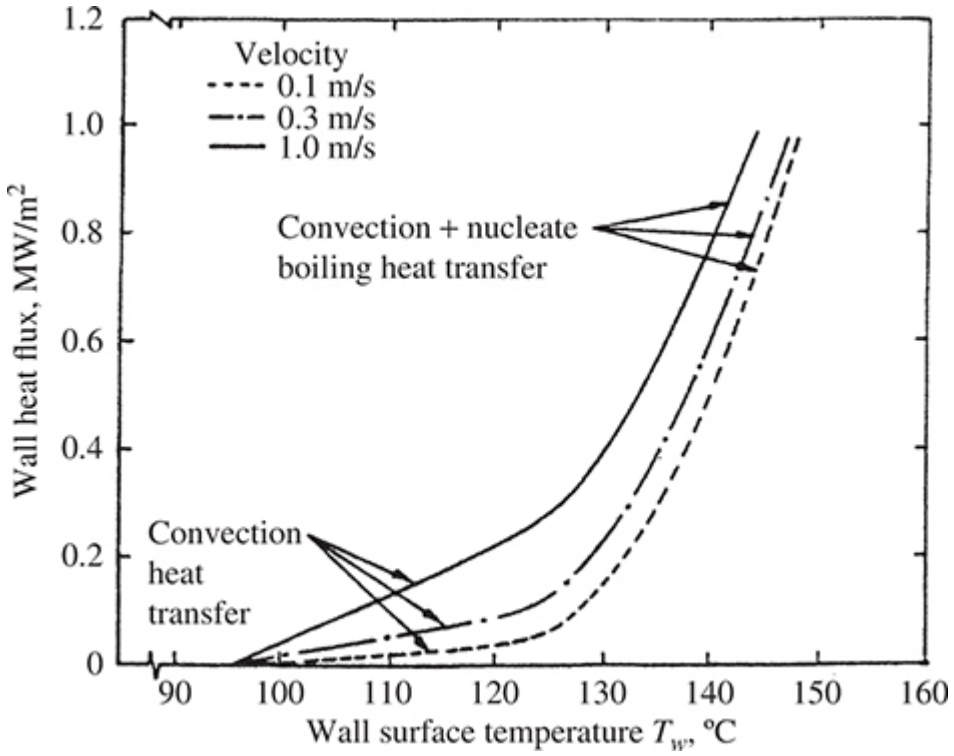


Figure 12.27 Impact of nucleate boiling heat transfer on forced-convection engine cooling: influence of cooling water velocity on relationship between heat flux and wall surface temperature (cooling water temperature 95°C, and pressure 2 bar).[54]

Automotive liquid coolants are mixtures of ethylene glycol (or propylene glycol) with water usually in about equal mass proportions. Water with its high specific heat and density is a very good cooling fluid, but its freezing temperature (of 0°C) makes it unsuitable for many applications in colder regions. Glycol (usually ethanol glycol) is added as an "anti-freeze": with glycol, the water no longer freezes but forms ice crystals that, though they are precipitated—the so-called ice flaking point, can still be pumped through the engine's cooling circuit. The coolant's boiling point is also raised. Table 12.2 shows the effect that adding various levels of glycol to water has on the ice flaking point and the boiling point. Coolants also contain additives to

protect metallic cooling-system components against corrosion, to prevent foaming, and protect glycol from oxidation.

TABLE 12.2 Ice flaking and boiling points for water-glycol mixtures [55]

Glycol, % by vol.	Ice flaking point, °C	Boiling point, °C
10	−4	101
20	−9	102
30	−17	104
40	−26	106
50	−39	108

Mixtures of glycol and water are not as effective coolants as water due to their different physical properties. Values of the physical properties of automotive coolants can be found in standard references (e.g., Ref. 56). Equations for the variation of the relevant physical properties of automotive coolants, with temperature, for 50% glycol aqueous mixtures have been developed by Alshamani. [57]

Wall Material

While the common metallic component materials of cast iron and aluminum have substantially different thermal properties, they both operate with combustion chamber surface temperatures (200 to 400°C) that are low relative to burned gas temperatures. Interest continues in exploring materials that could operate at much higher temperatures so that the heat losses from the working fluid would be reduced. Ceramic materials, such as silicon nitride and zirconia, have much lower thermal conductivity than cast iron, would operate at higher temperatures, and thereby insulate the engine. The thermal properties of some of these materials are listed in Table 12.3. With these thermally insulating materials, it is possible to reduce the heat transfer through the wall by a substantial amount.

TABLE 12.3 Thermal properties of wall materials

Material	Thermal conductivity k, W/m·K	Density ρ, kg/m^3	Specific heat c, J/kg·K	Thermal diffusivity α, m^2/s	$k\rho c$	Skin depth δ, mm	Peak temperature swing, K
Cast iron	54	7.2×10^3	480	1.57×10^{-5}	1.8×10^8	2.8	18
Aluminum	155	2.8×10^3	915	6.2×10^{-5}	3.9×10^8	5.4	12
Silicon nitride	5–10	2.5×10^3	710	2.8×10^{-6}	1.3×10^7	1.2	70
Sprayed zirconia	1.2	5.2×10^3	732	3.2×10^{-7}	4.6×10^6	0.39	95

This approach has been considered for diesel engines where there is the possibility of eliminating the conventional engine coolant system. Since the coolant-side heat transfer is essentially steady during each cycle, a high-enough thermal resistance in the wall material can bring the net heat transfer for the cycle to close to zero. However, there is still substantial heat transfer between the working fluid in the cylinder and the combustion chamber walls. Thermodynamic simulations of a turbocompounded diesel engine operating cycle, with the ceramic material at about 800 K surface temperature, show that the *net* heat transfer is much reduced compared with the normal metal wall case. However, there is substantial heat transfer to the gas from the ceramic walls during intake (which reduces volumetric efficiency) and compression (which increases compression stroke work), and still significant heat transfer from the gas during combustion and expansion. These negative effects of high combustion chamber surface temperatures reduce the efficiency benefits to only modest gains. [31] For spark-ignition engines, the heat transfer from the hot walls to the incoming charge makes thermally insulating materials more unattractive still. Such charge-heating heat transfer would increase the unburned mixture temperature leading to earlier onset of knock, the autoignition of the end gas fuel-air mixture (see Sec. 9.6).

The inherently unsteady nature of the heat-transfer interaction with the wall leads to time-varying combustion chamber surface temperatures. During combustion and expansion, the thermal energy transferred from the gas to the wall is stored in a thin layer of wall material adjacent to the surface. While some of this thermal energy diffuses through the wall, during intake and compression much of it is transferred back to the now low-temperature cylinder contents. The depth of penetration of the thermal wave into the material, the skin depth δ, is proportional to $\sqrt{\alpha/\omega}$ where $\alpha = k/(\rho c)$ is the thermal diffusivity and ω the frequency of the wave (proportional to engine speed). Values of α and δ are given in Table 12.3 for an engine speed of

1900 rev/min: $\sqrt{\alpha}/\delta \approx 1.4$ (units: $s^{-1/2}$), a constant. The magnitude of the temperature fluctuation (important because it is a source of fluctuating thermal stress) is proportional to $(\delta\rho c)^{-1}$: this varies as $(k\rho c)^{-1/2}$. Estimated peak surface temperature swings for the materials in Table 12.3 are tabulated.

Knock

Knock in an SI engine is the spontaneous ignition of the unburned "end-gas" ahead of the flame as the flame propagates across the combustion chamber. It results in an increase in gas pressure and temperature above the normal combustion levels (see Sec. 9.6). Knock results in increased local heat fluxes to regions of the piston, the cylinder head, and liner in contact with the autoignited end-gas. Increases to between twice and three times the normal heat flux in the end-gas region have been measured. [59, 60] It is thought that the primary knock damage to the piston crown in this region is due to metal fatigue resulting from the combination of high local pressure oscillations, and higher material temperatures.

12.7.2 Component Temperature Distributions

Here, we show illustrative examples of measured temperature distributions in various engine components. Normally, the heat flux to the combustion chamber surfaces is highest in the central region of each cylinder head, in the exhaust valve seat region, and to the center of the piston crown. It is lowest to the cylinder liner. Cast-iron pistons run about 40 to 80°C hotter than aluminum pistons. With flat-topped pistons (typical of spark-ignition engines) the center of the crown is hottest and the outer edge cooler by 20 to 50°C. Diesel engine piston crown surface temperatures are about 50°C higher than SI engine equivalent temperatures. As shown in Fig. 12.28, the maximum piston temperatures with DI diesel engine pistons are at the lip of the bowl. In prechamber diesel engines, maximum piston temperatures occur where the hot jet exiting the pre-chamber impinges on the piston crown.

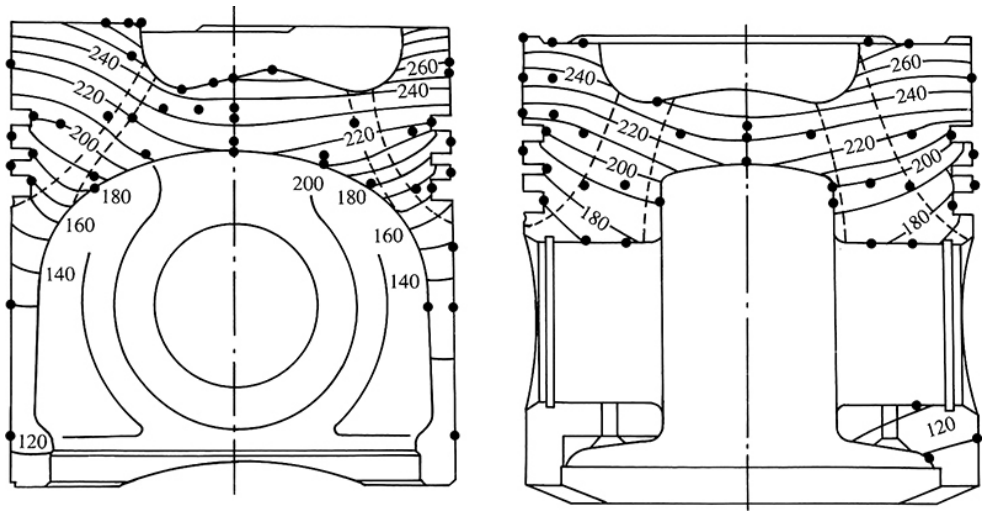


Figure 12.28 Isothermal contours (solid lines) and heat flow paths (dashed lines) determined from measured temperature distribution in piston of high-speed DI diesel engine. Bore 125 mm, stroke 110 mm, r_c = 17, 3000 rev/min, and full load. [58]

Figure 12.29 shows the temperatures at various locations on a four-cylinder SI engine cylinder head. The maximum temperatures occur where the heat flux is high and access for cooling is limited. Such locations are the bridge between the valves and the region between the exhaust valves of adjacent cylinders. Figure 12.30 shows how the average heat flux and temperature vary along the length of a DI diesel engine liner. Because the lower regions of the liner are only exposed to combustion products for part of the cycle after substantial gas expansion has occurred, the heat flux and temperature decrease significantly with distance from the cylinder head. Note that the heat generated by friction between the piston and the liner, the difference between $\dot{q}_{GL}$ (the gas to liner heat flux) and $\dot{q}_L$ (the total heat flux into the liner), is a significant fraction of the liner thermal loading.

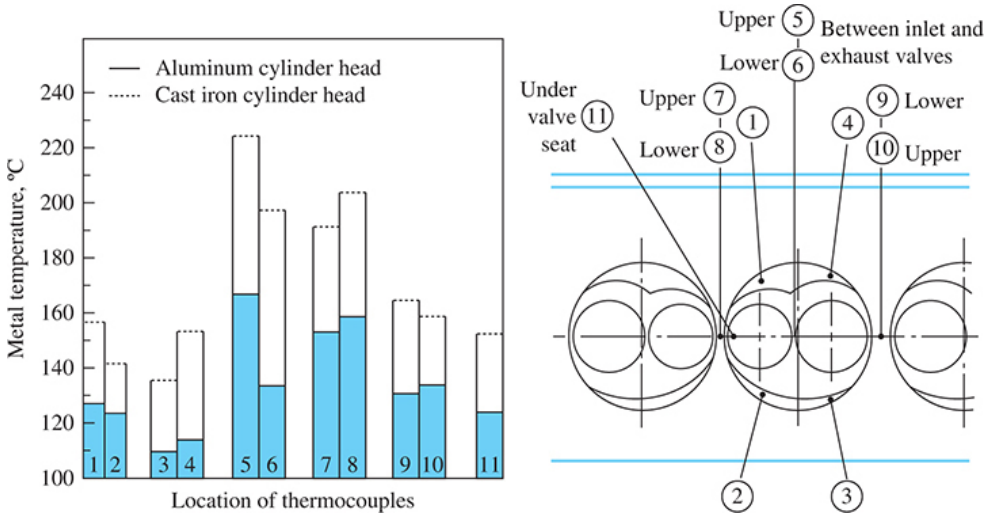


Figure 12.29 Variation of cylinder head temperature with measurement location in a spark-ignition engine operating at 2000 rev/min, wide-open throttle, with coolant water at 95°C and 2 atm.[54]

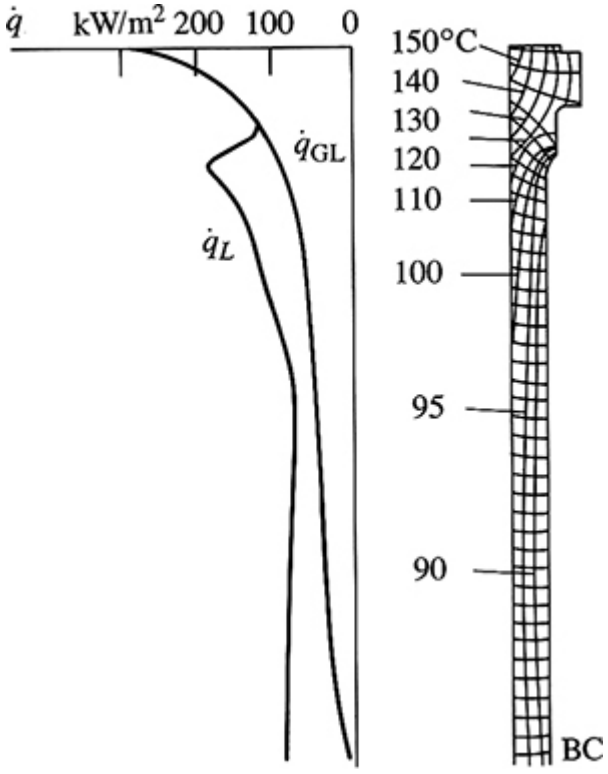


Figure 12.30 Temperature and heat flux distribution in the cylinder liner of a high-speed DI diesel engine at 1500 rev/min and bmep =11 bar. $\dot{q}_L$ is heat

flux into the liner; $\dot{q}_{GL}$ is heat flux from the gas to the liner: difference is friction-generated heat flux. [61]

The exhaust valve is cooled through the stem and the guide, and the valve seat. In small-size valves the greater part of the heat transfer occurs through the stem; with large-size valves, the valve seat carries the higher thermal load. Figure 12.31 shows the temperature distribution in an exhaust valve (one of two) in a two-stroke DI diesel at high load. Such peak exhaust valve temperatures of 650 to 700°C are typical of highly loaded engine operation. When additional exhaust valve head cooling is required, under extreme conditions, sodium-cooled valves can be used. Here the valve stem is hollow and is filled with sodium: the melting and thermally driven convective motion of the sodium in the stem transfers thermal energy from the valve head region to that portion of the stem within the valve guide.

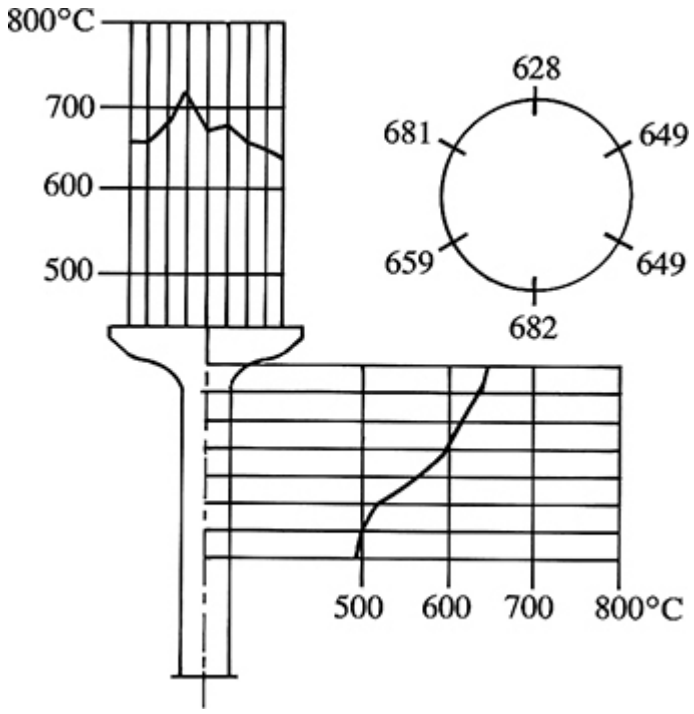


Figure 12.31 Temperature distribution in one of the four exhaust valves of a two-stroke cycle uniflow DDA 4-53 DI diesel engine. Bore = 98 mm, stroke = 114 mm. [62]

Intake valve temperatures in a warmed-up engine are higher than the coolant temperature (≈ 90°C) but significantly lower than exhaust valve temperatures. The valve head is heated by heat transfer from the high-temperature in-cylinder burned gases (and by the backflow of hot residual into the intake when the intake valve opens). It is cooled by the intake airflow and, with port fuel-injection, by the evaporation of liquid fuel off the valve head and stem surface. Figure 12.32 shows valve surface temperatures over the load range of a 2-liter four-cylinder spark-ignition engine with port fuel injection, with gasoline and propane fueling. The valve surface temperatures (at several locations on the valve head) rise modestly with increasing load—except where fuel spray impingements are direct and substantial (thermocouple location A). The head temperatures are about 40 deg. C higher with propane fuel, where there is no liquid evaporation. Fuel evaporation with gasoline cools the entire valve head. At locations with direct impingement, local evaporation of the impinging liquid steadily reduces the surface temperature (location A), as increasing amounts of liquid fuel are deposited as load increases. Under typical gasoline engine operating conditions, intake valve head temperatures are in the 150 to 200°C range. [24, 26]

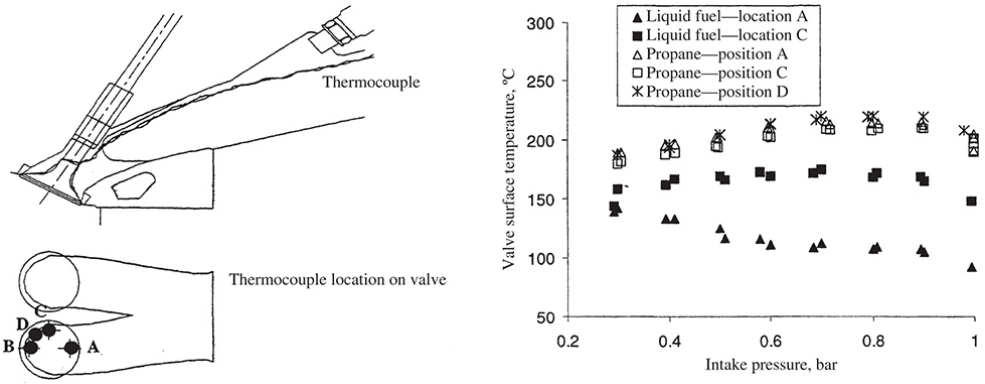


Figure 12.32 Intake valve-head surface temperatures at several locations in an operating spark-ignition engine fueled with gasoline and propane, as a function of load (range of intake manifold pressures), at 1500 rev/min.[26]

Temperature distributions in engine components can be calculated from a knowledge of the heat fluxes across the component surface using finite element analysis techniques. For steady-state engine operation, the depth

within a component to which the unsteady temperature fluctuations (caused by the variations in heat flux during the cycle) penetrate is small, so a quasi-steady solution is satisfactory. Results from such calculations for a spark-ignition engine piston illustrate the method. [63] A mean heat-transfer coefficient from the combustion chamber gases to the piston crown, and a mean chamber gas temperature, were defined (using the input from a cycle simulation of the type described in Sec. 14.4). These define the time-averaged heat flux into the piston. Heat-transfer coefficients for the different surfaces of the piston (underside of dome, ring-land areas, ring regions, skirt outer and inner surfaces, wrist pin bearings, etc.) were estimated. The actual piston shape was approximated with a three-dimensional grid for one quadrant of the piston. A standard finite-element analysis of the heat flow through the piston yields the temperature distribution within the piston. The thermal stresses can then be calculated and added to the mechanical stress field to determine the total stress distribution. Such results can be used to define the potential fatigue regions in the actual piston design. Figure 12.33 shows the temperature distribution calculated with this approach, compared with measurements (indicated by dots). The agreement is acceptable, except in the piston skirt where the heat-transfer rate between the skirt and cylinder liner has been overestimated.

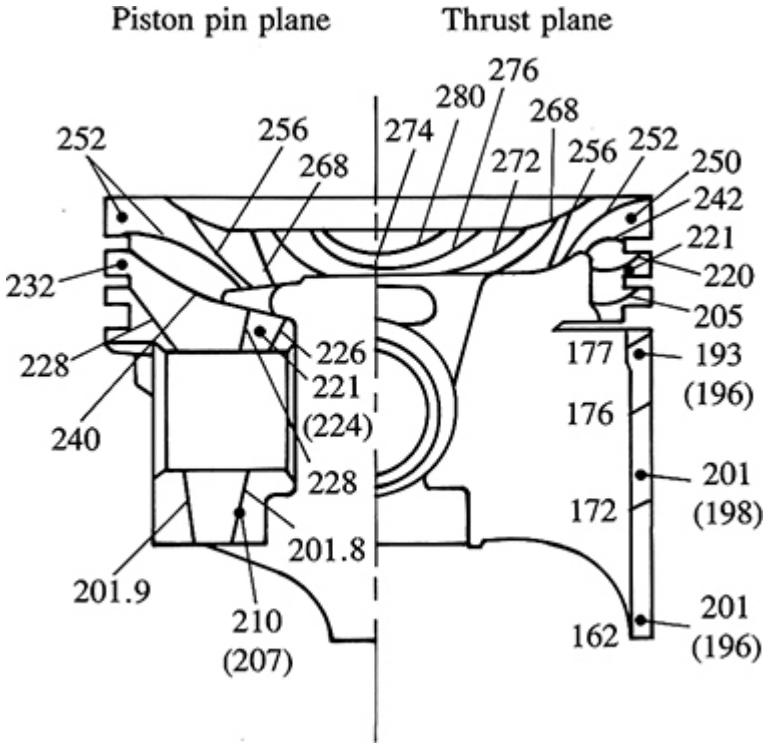


Figure 12.33 Measured (dots) and calculated temperature (°C) distributions in piston pin and thrust planes of the piston of a four-cylinder 2.5-liter spark-ignition engine at 4600 rev/min and wide-open throttle.[63]

Detailed measurements of the temperature distribution in the piston allow the relative amounts of heat which flow out of the different piston surfaces to be estimated. Figure 12.34 shows such estimates for a DI diesel engine at no-load and full-load. About 70% of the heat flows out through the ring zone, and much smaller amounts through the pin boss zone, underside of the crown and skirt. In larger diesel engines and highly loaded diesel engines, one or more cooling channels are usually incorporated into the piston crown. This reduces the heat flow out through the ring area significantly. [64]

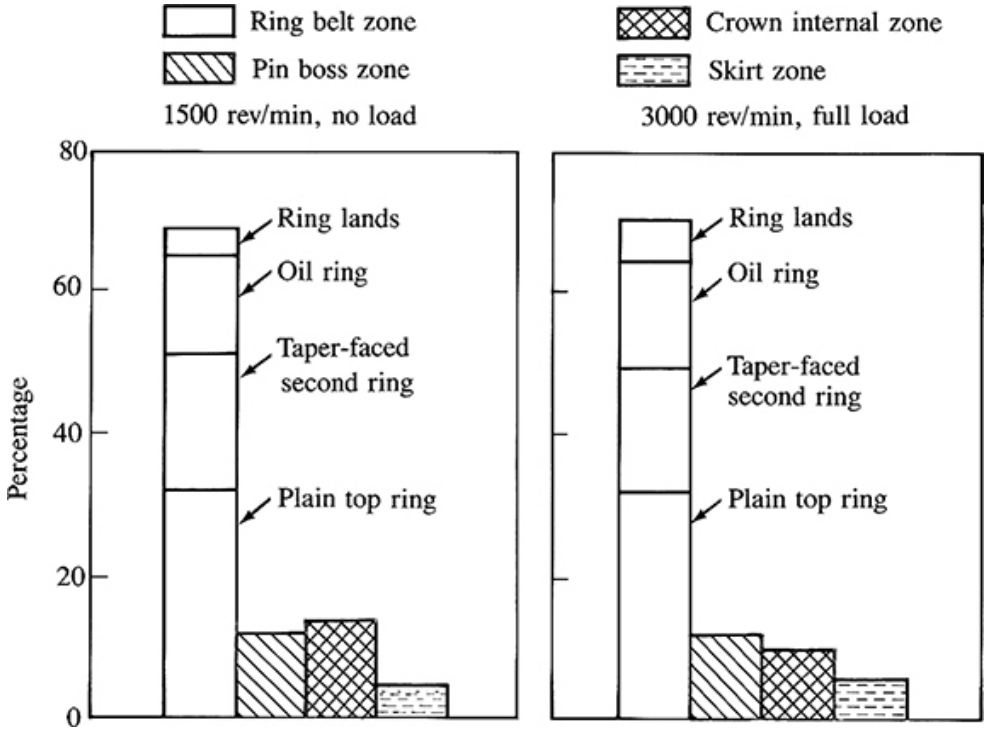
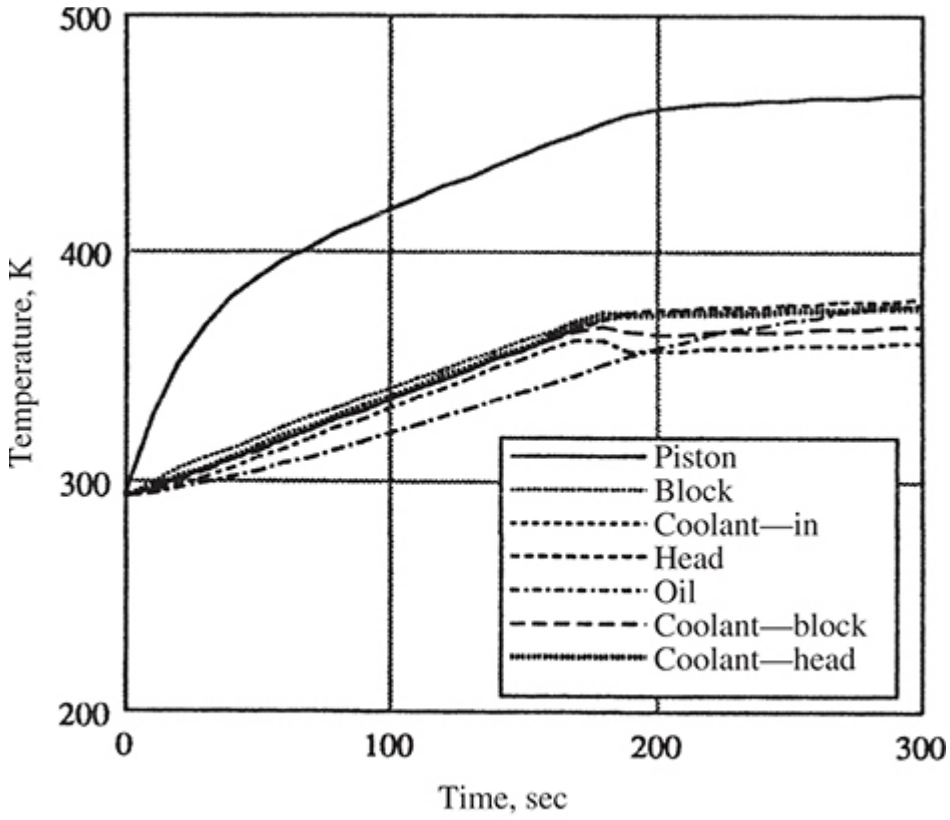


Figure 12.34 Heat outflow from various zones of a piston as percentage of heat flow in from combustion chamber. High-speed DI diesel engine, 125-mm bore, 110-mm stroke, $r_c = 17$. [58]

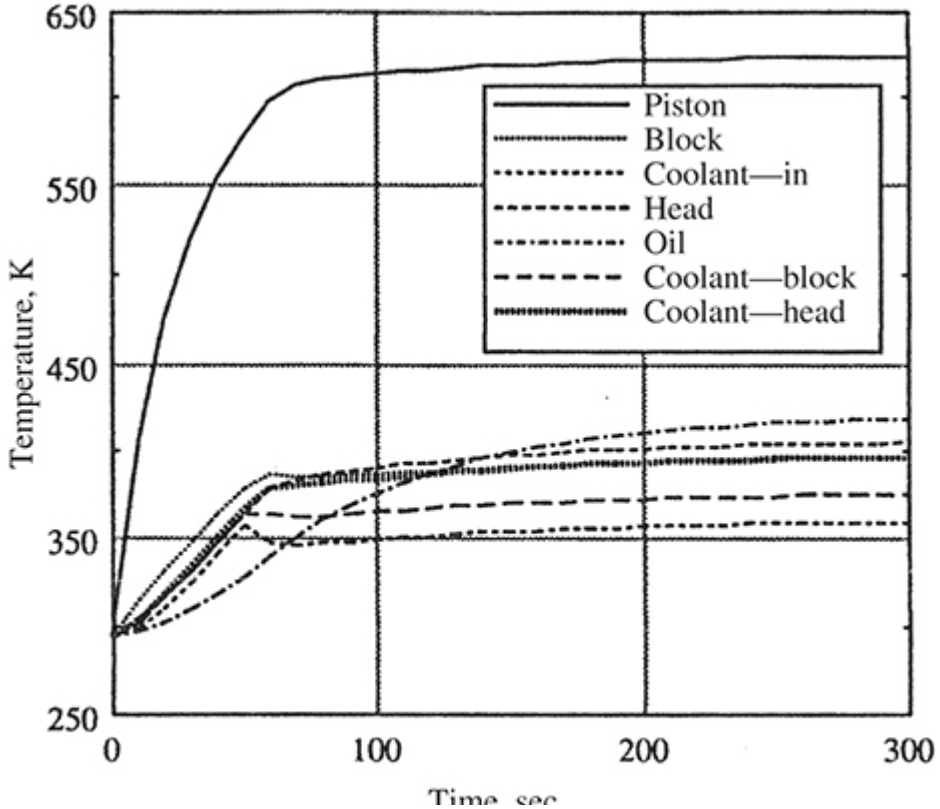
12.7.3 Engine Warm-Up

So far we have focused primarily on heat transfer processes in a warmed-up engine. When an engine is started after a substantial shut down period. The components are at close-to-ambient temperatures. Thus the wall temperature T_w in Eq. (12.2) is well below the warmed-up values discussed in the previous section, so the wall heat transfers (driven by $T_g - T_w$) are higher. It takes from one to ten or so minutes for the engine's components to reach their steady state temperatures, depending on operating conditions. The engine's performance, efficiency, and emissions depend on component (and exhaust-gas) temperatures. These are all affected by the engine's heat transfer process: an engine's warm-up behavior is important.

The engine warm-up process can be studied through measurement of component surface and coolant temperatures and through the analysis of heat flows through the components that make up the total engine structure. Figure 12.2 illustrates the layout of models used to do this type of analysis, with heat flows through thermal resistance networks, and thermal capacitances at the nodes to capture the changing thermal state (temperature) of each major component and the fluids involved. Results from such analysis are shown in Fig. 12.35, [65] which shows temperature versus time plots for the major components of a gasoline SI engine at light load, and at wide-open throttle. [Note each component in the block diagram of the engine structure is characterized by its mean temperature since the Biot number (hd/k), the ratio of the resistance to conduction within a solid to the convection heat flux resistance across the thermal boundary layer at the fluid/solid surface boundary, is less than 0.1.]


500
400
300
200
0
100
200
300
Temperature, K
Time, sec
Piston
Block
Coolant—in
Head
Oil
Coolant—block
Coolant—head

(*a*)


650
550
450
350
250
0
100
200
300
Temperature, K
Piston
Block
Coolant—in
Head
Oil
Coolant—block
Coolant—head

Time, sec

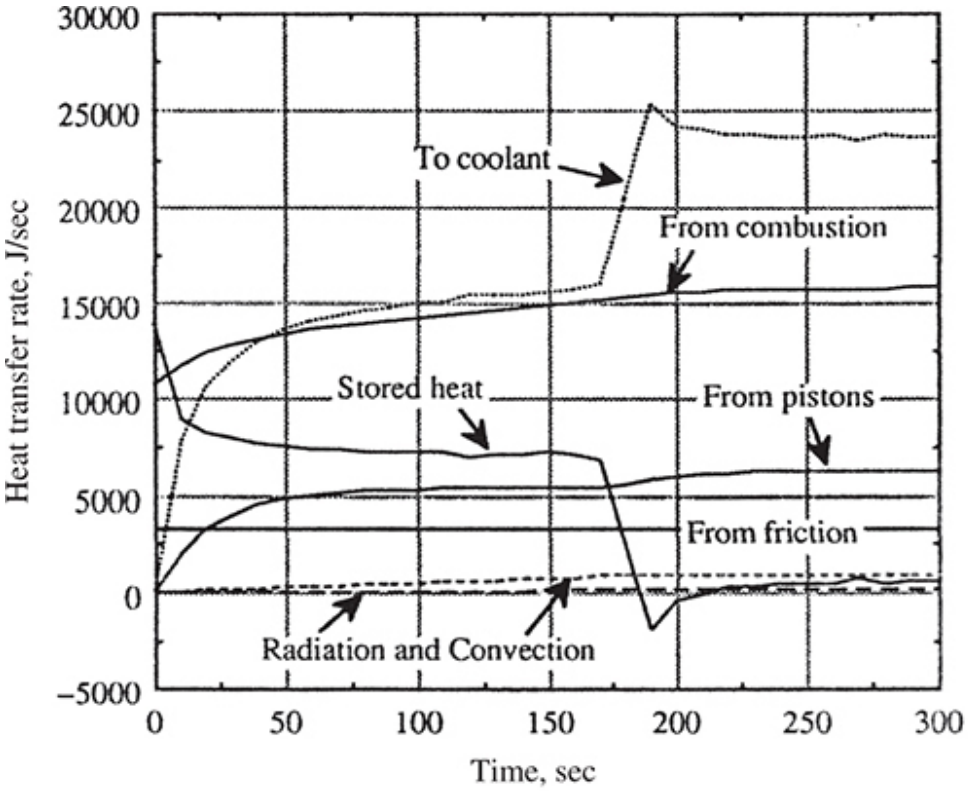
(*b*)

Figure 12.35 Predicted temperatures of key components as a function of time following a cold (20°C) engine start. 1.9-liter four-cylinder SI engine; coolant 50% ethylene glycol, flow rate 0.57 kg/s; oil flow rate, 0.08 kg/s. (*a*) 1500 rev/min, 260-kPa bmep; (*b*) 2500 rev/min, wide-open throttle. Convective heat transfer coefficients, $W/m^2 \cdot K$: head, 10; block, 25; oil pan, 70. [65]

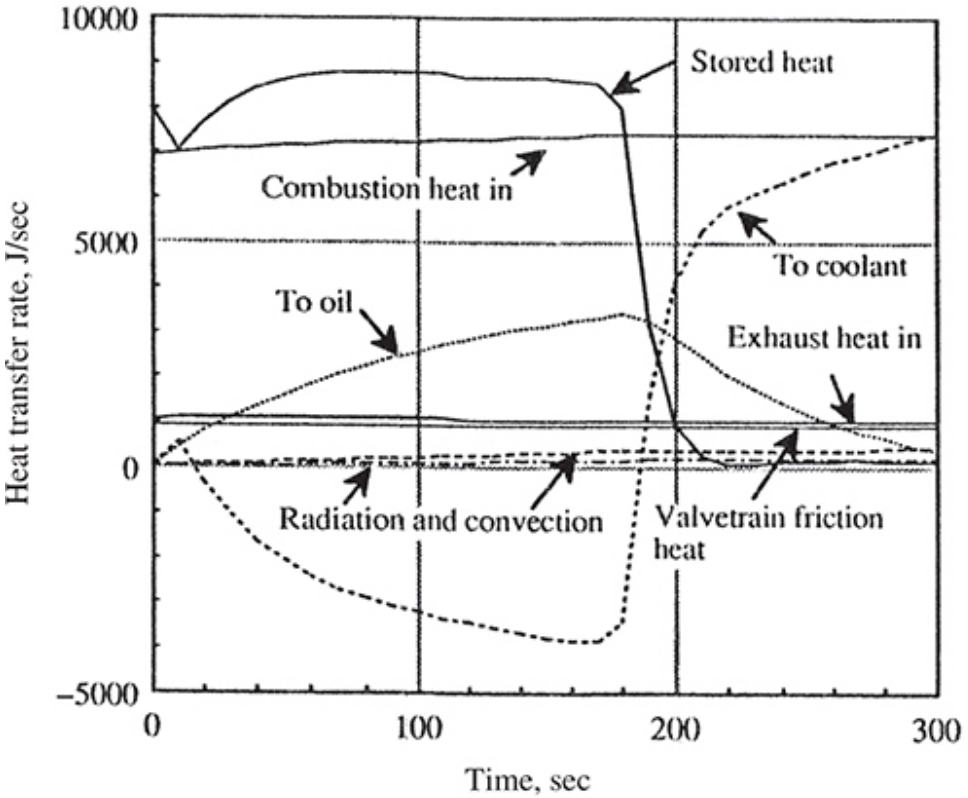
Engine operating conditions have significant effects on component temperatures, as discussed in the previous section. Figure 12.35 *a* shows light-load low-speed behavior (typical of normal passenger vehicle driving): temperatures rise over some 200 seconds. The piston (since it is not directly cooled by the coolant) has a temperature about 100 K hotter than the other components. Its temperature rises rapidly over the first 50 seconds, and then more slowly, paralleling the block, head, and coolant temperatures. The coolant circuit thermostat opens, allowing coolant flow to the radiator, at about 175 seconds. The oil temperature lags because its thermal energy is received through secondary processes. The rate of heat transfer to the oil is small until the rest of the components reach higher temperatures. At the higher load and speed, component temperatures rise more rapidly, and to higher values: the thermostat opens at about 60 seconds after start up. This accelerated behavior results from the significantly higher heat fluxes (W/m^2) at the wide-open-throttle condition, primarily due to the higher engine speed. [65] Characteristic times for component warm up are not as strongly influenced by load since, while heat fluxes are higher (see Fig. 12.23), the steady-state warmed-up temperatures are higher too. [66]

The complexity of the warm-up process in the major engine components is indicated by the results shown in Fig. 12.36, where the heat flow rates into and out of the engine's block and cylinder head, and the increase in these component's stored thermal energy, are plotted as a function of time. Figure 12.36 *a* shows the heat flow through the block. The largest heat transfer is to the coolant. The rapid increase in heat flux to the coolant at 165 seconds occurs because the thermostat has opened, admitting cool water. The heat flux form the piston to the liner plays an important role in heating this component, adding almost 40% to the direct heat flux from the hot in-cylinder

combustion gases. At this point, the rate at which the thermal energy stored in the block increases, drops to close to zero. Note that in the quasi-steady-state situation beyond this point, the heat transfer driven by the in-cylinder hot gases resulting from combustion, levels off at about 70% of the heat flux into the coolant: the additional coolant thermal load is the heat flux from the pistons through the cylinder liners to the coolant, augmented by thermal energy from the lubricating oil generated by frictional dissipation. [65]


30000
25000
20000
15000
10000
5000
0
−5000
Heat transfer rate, J/sec
To coolant
From combustion
Stored heat
From pistons
From friction
Radiation and Convection
0
50
100
150
200
250
300
Time, sec

(*a*)


10000
5000
0
−5000
Heat transfer rate, J/sec
Stored heat
Combustion heat in
To coolant
To oil
Exhaust heat in
Radiation and convection
Valvetrain friction heat
0
100
200
300
Time, sec

(*b*)

Figure 12.36 (*a*) Predictions of heat flow rates into and out of engine block of a four-cylinder spark-ignition engine, and rate of thermal energy storage in the block. (*b*) Predictions of heat flow rates and storage for the cylinder head. 1.9-liter four-cylinder gasoline engine operating at stoichiometric air/fuel ratio, MBT spark timing, 1500 rev/min, bmep 260 kPa. [65]

Figure 12.36 *b* shows similar results for the engine's cylinder head. The simulation results indicate that the coolant initially picks up thermal energy from the block and actually heats the head. The oil removes heat form the head until the radiator thermostat opens: shortly thereafter (see Fig. 12.35 *a*) it reaches a higher temperature than the head and becomes a source of thermal energy for this component. [65]

These discussions indicate the complexity of this transient engine warm-up process. Many of these details and trends are confirmed by experimental data (e.g., Ref. 67). Quantifying the component and gas temperatures during the engine warn-up is important for many reasons. The lower than steady-state temperature values increases engine friction: at 20°C the mechanical friction power loss is almost twice the warmed-up value. [67] The exhaust gas temperature is lower during warm-up which negatively impacts the turbine performance in the turbocharger. The fast light-off of the exhaust system catalyst and particulate traps is affected by the thermal energy flux in the engine exhaust. This is lower during warm-up.

PROBLEMS

12.1 If radiation in the combustion chamber is negligible, Eqs. (12.5) , (12.6), and (12.7) can be combined into the following overall equation approximating the time-averaged heat transfer from the engine:

$$\dot{q} = h_{c,o}(\bar{T}_g - \bar{T}_c)$$

Derive the expression for the overall heat transfer coefficient, $h_{c,o}$, for the flow of thermal energy from the hot gases in the cylinder, through the wall, into the coolant.

12.2 Given that the average heat flux through a particular zone in a cast iron liner 1 cm thick is 0.2 MW/m^2, the coolant temperature is 85°C, and the coolant side heat-transfer coefficient is 7500 W/m^2 · K, find the average surface temperature on the combustion chamber and coolant sides of the liner at that zone.

12.3 Figure 12.1 shows a schematic of the temperature profile from the gas inside the combustion chamber out to the coolant. Draw an equivalent figure showing schematically the temperature profiles at the following points in the engine cycle: (*a*) late in the intake stroke; (*b*) just prior to combustion; (*c*) just after combustion; (*d*) during the exhaust stroke. Your sketch should be carefully proportioned.

12.4 Using dimensional analysis, compare the relative heat losses of two geometrically similar SI engines (same bore/stroke ratio, same connecting rod/stroke ratio) operating at the same imep and power. Engine A has twice the displacement per cylinder of engine B. Assume that the wall temperature and the gas temperature for both engines are the same.

12.5 **(a)** Using Woschni's in-cylinder heat transfer correlation, evaluate the percentage increase in heat transfer expected in an engine with a mean piston speed of 10 m/s when the swirl ratio is raised from 0 to 5. Do your comparison for the intake process only. The engine bore is 0.15 m and the engine speed is 2000 rev/min.
(b) Explain how both the generation of swirl and the change in heat transfer that results affect the volumetric efficiency of the engine.

12.6 **(a)** Explain how you would estimate the thermal boundary-layer thickness on the combustion chamber wall of an internal combustion engine.
(b) Using representative data, make a rough estimate of the thickness of the thermal boundary layer in the combustion chamber of an SI engine just after the completion of combustion, and the fraction of the cylinder mass contained within the boundary layer. $B \approx 100$ mm. Transport properties given in Sec. 4.8 .

12.7 **(a)** Using the analysis found in Sec. 12.6.1 , calculate the depth below the surface where the amplitude of the temperature oscillations has attenuated

to 1% of the amplitude at the surface. The wall material is aluminum and the four-stroke cycle engine is operating at 2500 rev/min. For this estimate, consider only the temperature oscillations which have a frequency equal to the engine firing frequency ($\omega = 2\,\pi_N/\,n_R$).

(b) Repeat the calculation for the engine operating at 5000 rev/min.

(c) What is the dependence of the penetration depth on the amplitude of surface temperature fluctuations?

12.8 The instantaneous heat-transfer rate $\dot{Q}$ from the cylinder gases to the combustion chamber walls in a spark-ignition gasoline engine may be estimated approximately from the equation

$$\dot{Q} = h_c A(\bar{T}_g - \bar{T}_w)$$

where h_c is the heat-transfer coefficient, A is the surface area, $\bar{T}_g$ is the average temperature of the gas in the cylinder, and $\bar{T}_w$ is the average wall temperature. The heat-transfer coefficient can be obtained from the Nusselt, Reynolds, and Prandtl number relationship:

$$\mathrm{Nu} = C(\mathrm{Re})^m(\mathrm{Pr})^n$$

where $C = 0.4$, $m = 0.75$, $n = 0.4$. The characteristic velocity and length scales used in this relation are the mean piston speed and the cylinder bore. Assuming appropriate values for the engine geometry and operating conditions at wide-open throttle with the wall temperature at 400 K, at an engine speed of 2500 rev/min, and using the cylinder pressure versus crank angle curve of Fig. 12.22, calculate the following:

(**a**) The *average* temperature of the gas in the cylinder at $\theta = -180°$, $-90°$, $0°$, $20°$, $40°$, $90°$, $150°$.

(**b**) The instantaneous heat-transfer coefficient h_c and heat-transfer rate $\dot{Q}$ from the gas to the combustion chamber walls of one cylinder at these crank angles. Plot these results versus θ.

(**c**) Estimate the fraction of the fuel energy that is transferred to the cylinder walls during compression and during expansion.

Assume for the gas that the viscosity $\mu = 7 \times 10^{-5}$ kg/m · s, the thermal conductivity $k = 1.5 \times 10^{-1}$ J/m · s · K, the molecular weight = 28, and the

Prandtl number is 0.8. Assume that the combustion chamber is disc shaped with $B = 102$ mm, $L = 88$ mm, and $r_c = 9$. (The calculations required for this problem are straightforward; do not attempt anything elaborate.)

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[a] h_c is an empirical constant, usually determined through correlation with experimental data: see Sec. 12.4 .

[b] The term $(\overline{T}_g^4 - \overline{T}_{w,g}^4)$ in Eq. (12.5) can be rearranged as $(\overline{T}_g^2 + \overline{T}_w^2)(\overline{T}_g + \overline{T}_w)(\overline{T}_g - T_w)$ which, to a reasonable approximation, can be scaled linearly with $(T_g - T_{w,g})$.

[c] The percentages are approximate. Note that, for a diesel engine, the exhaust chemical energy is significantly lower.

[d] Taylor and Toong also found $a = 10$ a good fit. Note $n \approx 0.3$, so for gases Pr^n is close to unity.

[e] The model calculates the boundary layer thickness using a quasi-steady balance between the thermal energy flux to the wall and the additional adiabatic-core burned gas that enters the thickening thermal boundary layer region and experiences the decrease in gas temperature—adiabatic-core gas temperature to average of the wall temperature and that core temperature.

CHAPTER 13

Engine Friction and Lubrication

13.1 BACKGROUND

Not all the work transferred to the piston from the high-pressure gases contained inside the cylinder—the indicated work—is available at the drive shaft for actual use. That portion of the work transferred, which is not available, is usually termed friction work. It is dissipated in a variety of ways within the engine and engine auxiliaries. The friction work or power is a sufficiently large fraction of the indicated work or power—varying between about 10% at full load and lower speeds, through 50% at light load, to 100% at idle or no-load—for the topic to be of great practical importance. Friction losses affect the maximum brake torque and minimum brake specific fuel consumption directly. A large part of the friction losses appear as thermal energy in the coolant and oil which must be removed in the radiator and oil cooler system. Thus, friction losses influence the size of the engine coolant systems. A knowledge of friction power is required to relate the combustion characteristics of an engine—which influence the indicated power—and the useful output—the brake power, as illustrated in Fig. 13.1.

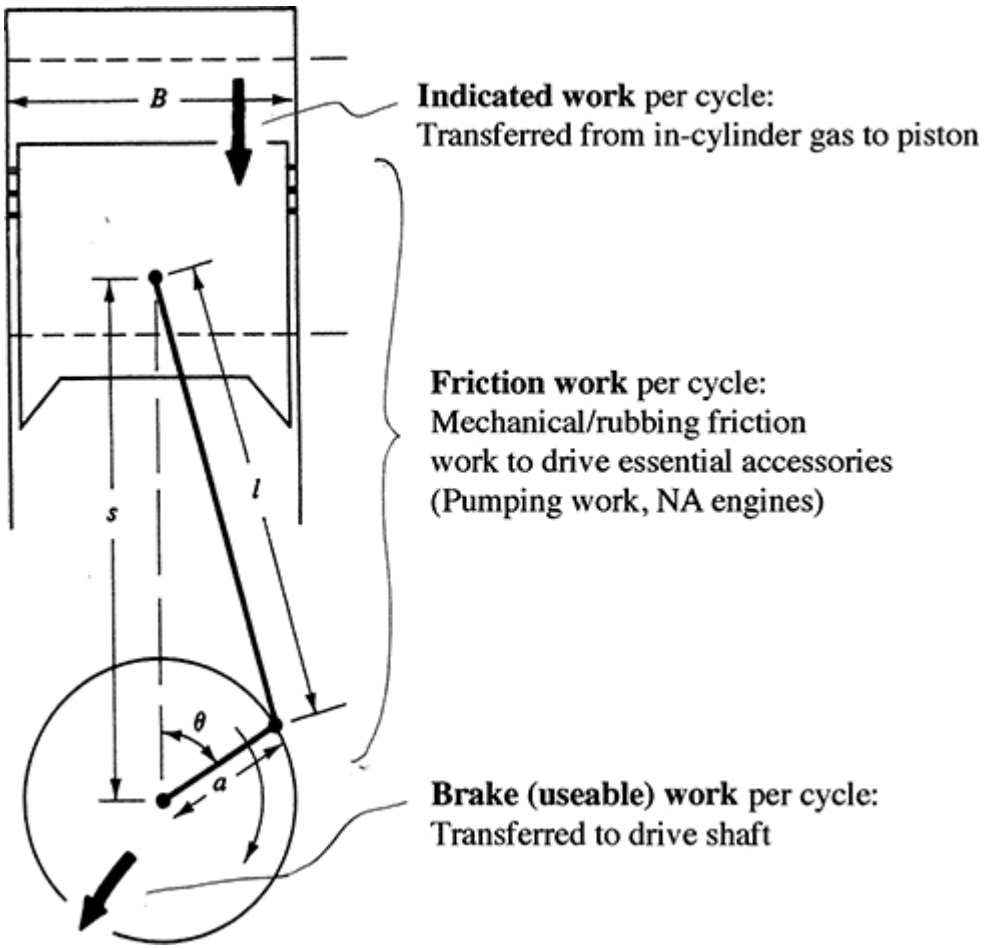


Figure 13.1 Work flow within the engine: Indicated work is transferred from the in-cylinder gases to the piston; a portion of this, the friction work, is dissipated; the brake work is then available at the drive shaft.

The friction work, defined as the difference between the work delivered to the piston from the working fluid within the cylinder, and the useable work delivered to the drive shaft, is expended as follows:

- To draw the fresh mixture through the intake system and into the cylinder, and to expel the burned gases from the cylinder and out of the exhaust system. This is usually called the pumping work.
- To overcome the resistance to relative motion of all the moving parts of the engine. This includes the friction between the piston rings, piston skirt, and cylinder wall; friction in the piston pin, connecting rod small and big ends, crankshaft and camshaft bearings and seals; friction in the

valve actuation mechanism; friction in the gears, or pulleys and belts, which drive the camshaft and engine accessories.

- To drive the engine accessories. These usually include: the water pump, the oil pump, the fuel pump, a secondary air pump (if used) for emission control, the fan, and the generator. The various pumps are defined as the *essential* engine accessories.

Pumping work is treated differently in naturally-aspirated and turbocharged engines. In naturally-aspirated engines, the pressure in the cylinder during the intake stroke is below atmospheric pressure due to flow resistances in the total intake system, and is higher than atmospheric pressure during the exhaust stroke. The pumping work per cycle (the $\int pdV$ over the exhaust and intake strokes) is therefore negative and thus dissipates part of the gross indicated work (see Fig. 6.1). In turbocharged engines the same intake and exhaust system flow resistances and pressure drops are present, but the in-cylinder intake and exhaust stroke pressures are primarily controlled by the turbocharger compressor and turbine behavior. The air pressure rise the compressor produced at higher engine loads is significantly larger than the intake flow resistance-induced pressure losses. And at higher engine loads, the pressure difference needed to drive the turbine (which then drives the compressor) is higher than the exhaust flow resistance pressure drops (see Fig. 6.2). Note that at lighter loads and lower turbocharged engine speeds, the flow-resistance pressure drops, and the impact of the turbocharger on intake and exhaust stroke in-cylinder pressures, are small. As load and speed increase, the flow resistances increase (see Sec. 6.2.3) but the compressor outlet and turbine inlet pressures increase much more. Under these more highly loaded conditions, the pumping work over the exhaust and intake strokes becomes positive and *adds* to the gross indicated work. Note that the exhaust gas temperature in spark-ignition engines is significantly higher than in diesel engines. Thus, the exhaust manifold pressures required to drive the turbines in diesel engines are higher than in spark-ignition engines. As a consequence, in diesels this higher-load (positive) pumping work is smaller.

All this work is eventually dissipated into thermal energy: the term friction work or power is therefore appropriate. In this chapter we will explore the relative importance of these components in typical automotive SI and diesel engines at different loads and speeds. The absolute value of the

total friction work varies with load, and increases as speed increases. The piston and connecting rod assembly contributes the largest friction component.

The relative importance of the friction components in a naturally-aspirated gasoline SI engine at light load and low-to-mid speed is illustrated in Fig. 13.2. Typical ranges are crankshaft 10 to 15%, reciprocating (piston and con rod) 25 to 30%, valvetrain 10 to 15%, auxiliaries 10 to 15%, and pumping 30 to 45%. In a small turbocharged diesel engine at equivalent operating conditions, the pumping work is much lower (only a few percent) and the reciprocating component is significantly larger due to higher in-cylinder pressures. The high-pressure fuel pump drive requirement (small in a gasoline engine) becomes comparable to the water and oil pump requirements.

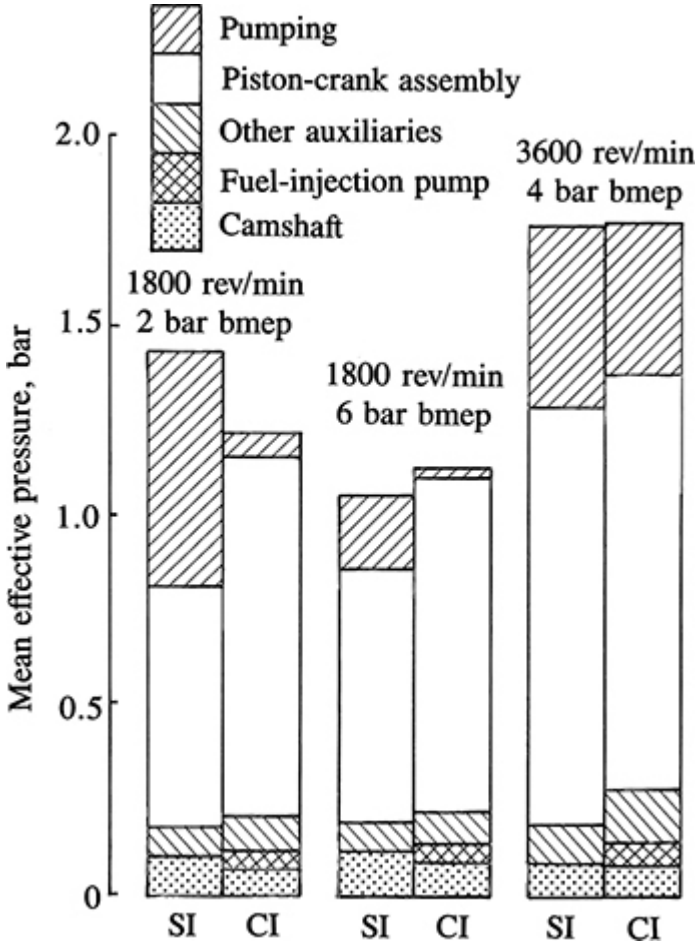


Figure 13.2 Comparison of major categories of friction losses: friction mean effective pressure at different loads and speeds for 1.6-liter four-

cylinder overhead-cam automotive spark-ignition (SI) and compression-ignition (CI) engines.[1]

Note that the *magnitude* of engine friction is important, and has been steadily reduced over the years through engineering research, development, and design improvements. The *relative* importance of friction relative to the indicated engine output is also important. As the specific output of engines has steadily increased, the relative (negative) impact of friction has been reduced. The pumping work for naturally-aspirated SI engines is larger than for equivalent CI engines and becomes comparable to rubbing friction at light loads as the engine is increasingly throttled.

13.2 DEFINITIONS

The following terminology will be used in discussing engine friction.

Pumping work W_p. The net work per cycle done by the piston on the in-cylinder gases during the inlet and exhaust strokes. W_p is only defined for four-stroke cycle engines. It is the area ($B + C$) in Fig. 2.6 *b* for a naturally-aspirated gasoline SI engine. [a]

Mechanical friction work W_{mf}. The work per cycle dissipated in overcoming the friction due to relative motion of adjacent components within the engine. This includes all the items listed in the second category of engine friction enumerated above. This is also referred to as *rubbing friction work*.

Accessory work W_a. The work per cycle required to drive the essential engine accessories; for example, water, oil, and fuel pumps, fan, generator, etc. Normally, only those accessories essential to engine operation are included.

Total friction work W_{tf}. The total friction work is the sum of these three components, that is,

$$W_{tf} = W_p + W_{mf} + W_a \tag{13.1}$$

It is often useful to discuss the difference between indicated and brake output in terms of *mean effective pressure*, mep, the work per cycle per unit displaced volume:

$$\text{mep} = \frac{W_c}{V_d}$$

and also power. Power and mep are related by

$$P = \text{mep} \times V_d \times \frac{N}{n_R}$$

where n_R (the number of revolutions per cycle) = 1 or 2 for the two-stroke or four-stroke cycles, respectively. Hence, from W_p, W_{mf}, W_a, and W_{tf} we can define pumping mean effective pressure and power (pmep and P_p), mechanical friction mean effective pressure and power (mfmep and P_{mf}), accessory mean effective pressure and power (amep and P_a), and total friction mean effective pressure and power (tfmep and P_{tf}), respectively.

Brake mean effective pressure and power (bmep and P_b), indicated mean effective pressure and power (imep and P_i), and mechanical efficiency have already been defined in Secs. 2.3 , 2.4, 2.5, and 2.7. Note that for four-stroke cycle engines, two definitions of indicated output are in common use. These have been designated as:

Gross indicated mean effective pressure, imep $_g$. The work delivered to the piston over the compression and expansion strokes, per cycle per unit displaced volume.

Net indicated mean effective pressure, imep $_n$. The work delivered to the piston over the entire four strokes of the cycle, per unit displaced volume.

From the above definitions it follows that for naturally-aspirated four-stroke cycle engines:

$$\text{imep}_g = \text{imep}_n + \text{pmep} \qquad (13.2a)$$

$$\text{tfmep} = \text{pmep} + \text{mfmep} + \text{amep} \tag{13.2b}$$

and

$$\text{bmep} = \text{imep}_g - \text{tfmep} \tag{13.2c}$$

For turbocharged four-stroke cycle engines:

$$\text{imep}_n = \text{imep}_g + \text{pmep} \tag{13.2d}$$

$$\text{tfmep} = \text{mfmep} + \text{amep} \tag{13.2e}$$

and

$$\text{bmep} = \text{imep}_n - \text{tfmep} \tag{13.2f}$$

Note that for both types of engines, pmep in these equations is a *positive* quantity: thermodynamically, the pumping work per cycle ($\int pdV$) for a naturally-aspirated engine is *negative*, and for higher loads in a turbocharged engine where $p_i > pe$, it is *positive* (and will approach zero at mid-load and become negative at the lighter engine loads).

That two different definitions of indicated output are in common use follows from two different approaches to determining friction work or power. In the standard engine test code procedures friction power is measured in a hot *motoring test*: the engine is motored with water and oil temperatures held at the firing engine values, with the throttle setting unchanged from its firing engine position (in an SI engine). This measures (approximately) the sum of pumping, mechanical or rubbing friction, and auxiliary power requirements. The sum of brake power, and friction power determined in this way, is the gross indicated power. Alternatively, when an accurate record of cylinder pressure throughout the whole cycle is available, pumping power can be determined directly: the sum of mechanical friction and accessory power is then the difference between the net indicated power —determined from $\int pdV$ over the whole cycle—and the brake power.

13.3 FRICTION FUNDAMENTALS

The friction losses outlined in Sec. 13.1 can be classified into two groups, depending on the type of dissipation which occurs. One type is friction between two metal surfaces in relative motion, with a lubricant in between. The other type is dissipation in turbulent flows.

13.3.1 Lubricated Friction

A primary problem in understanding friction between lubricated surfaces in engines is the wide variation in the magnitude of the forces involved. Thus various regimes of lubrication can occur. Figure 13.3 shows two common geometries for lubricated parts: a journal bearing and a slider bearing. The different regimes of lubricated friction are illustrated by the Stribeck diagram shown in Fig. 13.4, where the coefficient of friction f (ratio of tangential force to normal force) for a journal bearing is plotted against a dimensionless duty parameter $\mu N/\sigma$, where μ is the dynamic viscosity of the lubricant, N is the rotational speed of the shaft, and σ is the loading force per unit area. For sliding surfaces the dimensionless duty parameter becomes $\mu U/(\sigma b)$, where U is the relative velocity of the two surfaces and b is the length of the sliding pad in the direction of motion.

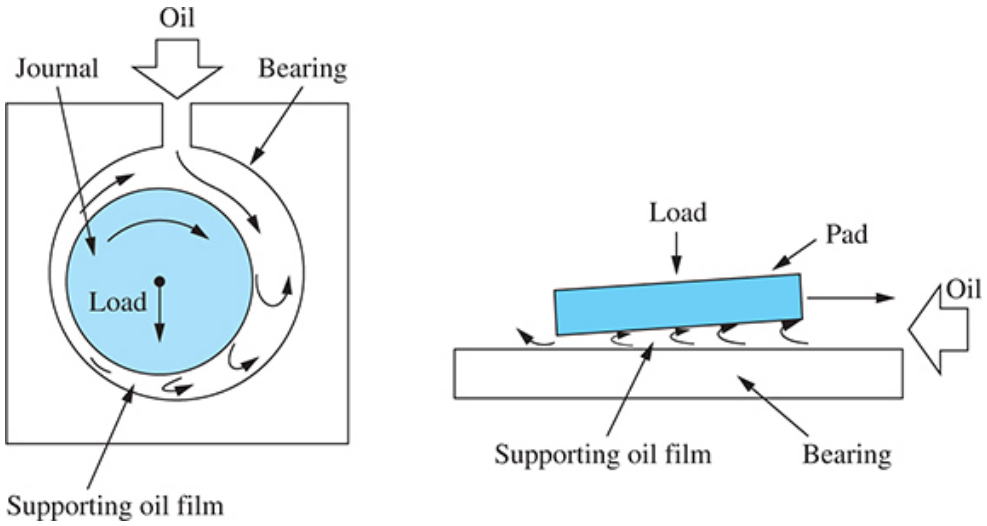


Figure 13.3 Schematic of a lubricated journal bearing, and a sliding bearing.

The coefficient of friction can be expressed as

$$f = \alpha f_s + (1-\alpha) f_L \tag{13.3}$$

where f_s is the metal-to-metal coefficient of dry friction, f_L is the hydrodynamic coefficient of friction, and α is the metal-to-metal contact constant, varying between 0 and 1. As $\alpha \to 1, f \to f_s$, and the friction is called *boundary friction*; that is, close to solid friction. The lubricating film is then sufficiently thin that it cannot prevent metal-to-metal contact between surface asperities. As $\alpha \to 0, f \to f_L$ and the friction is called *hydrodynamic* or *viscous* or *thick film* lubrication. The lubricant film is sufficiently thick that it separates completely the surfaces in relative motion. In between these regimes, there is a *mixed* or *partial* lubrication regime where the transition from boundary to hydrodynamic lubrication occurs. While Fig. 13.4 applies to journal bearings, this discussion holds for any pair of engine parts in relative motion with lubricant in between.

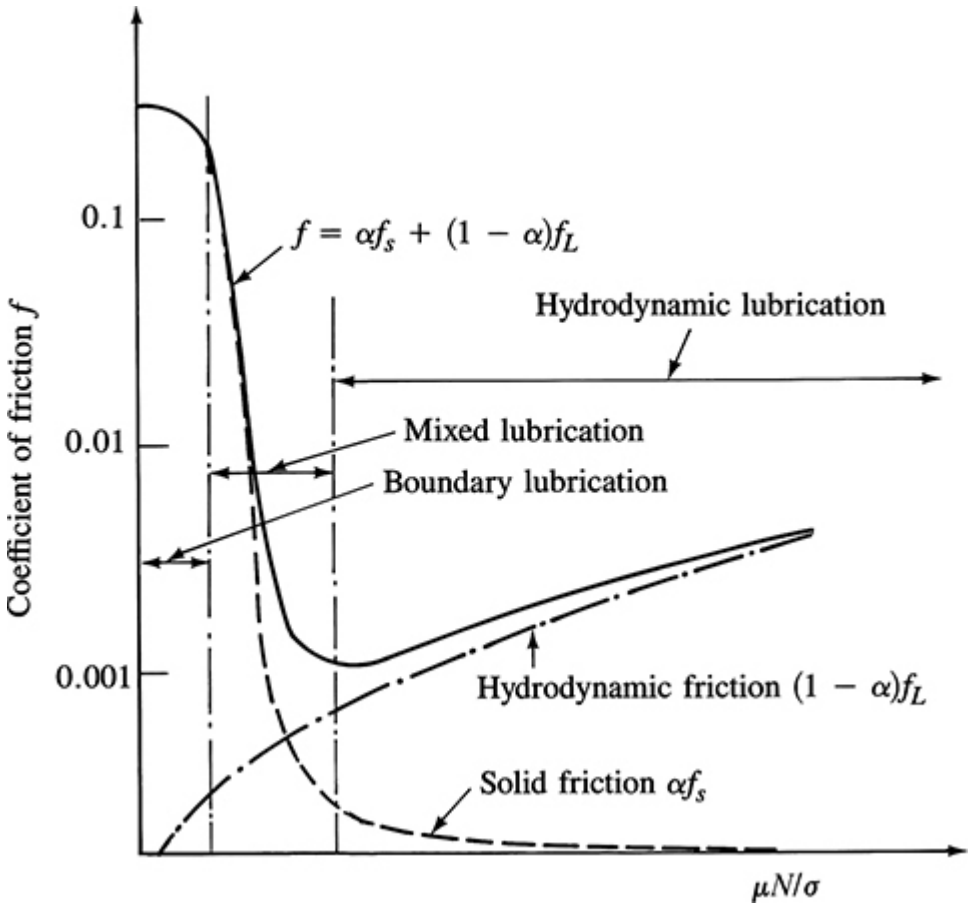


Figure 13.4 Stribeck diagram for journal bearing: coefficient of friction f versus dimensionless duty parameter $\mu N/\sigma$, where μ is the lubricant dynamic viscosity, N is rotational speed of shaft, and s is the loading force per unit area.

Under boundary lubrication conditions, the friction between the two surfaces in relative motion is determined by surface properties as well as by lubricant properties. The important surface properties are roughness, hardness, elasticity, plasticity, shearing strength, thermal conductivity, and wettability with respect to the lubricant. The important lubricant properties are mainly surface ones or chemical ones, which govern the ability of lubricant (and additive) molecules to attach themselves to the solid surfaces. Figure 13.5 shows two surfaces under boundary lubrication conditions. Due to the surface asperities, the real contact area is much less than the apparent contact area. The real contact area A_r is equal to the normal load F_n divided by the yield stress of the material σ_m:

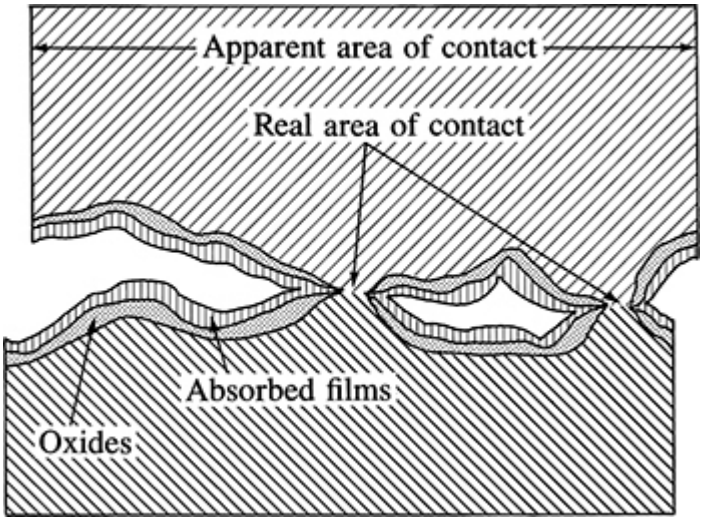


Figure 13.5 Schematic of two surfaces in relative motion under boundary lubrication conditions.[2]

$$A_r = F_n/\sigma_m$$

The force required to cause tangential motion is the product of the real contact area and the shear strength of the material τ_m:

$$F_t = A_r \tau_m$$

Thus the coefficient of friction f is

$$f = F_t / F_m = \tau_m / \sigma_m \tag{13.4}$$

For dissimilar materials, the properties of the weaker material dominate the friction behavior. Since, as shown in Fig. 13.5, the surfaces are covered by oxide films and adsorbed lubricant films, the shear strength of the material in Eq. (13.4) is effectively the shear strength of the surface film. Under boundary lubrication conditions, the coefficient of friction is essentially independent of speed. Boundary lubrication occurs between engine parts during starting and stopping, and during normal running at the piston ring/cylinder interface at top and bottom center crank positions, between heavily loaded parts, between slow moving parts such as valve stems and rocker arms, and crankshaft timing gears and chains.

Hydrodynamic lubrication conditions occur when the shape and relative motion of the (lubricated) sliding surfaces form a liquid film in which there is sufficient pressure to keep the surfaces separated. Resistance to motion results from the vicious shear forces within the liquid film, and not from the interaction between surface irregularities, as was the case under boundary lubrication. The shear stress τ in a liquid film between two surfaces in relative motion is given by

$$\tau = \mu \left(\frac{dv}{dy} \right)$$

where μ is the fluid viscosity and (dv/ dy) the velocity gradient across the film. Hence, the friction coefficient (shear stress/normal load stress) in this regime will be proportional to viscosity × speed ÷ loading, essentially a straight line on the Stribeck diagram. Full hydrodynamic lubrication or viscous friction is independent of the material or roughness of the parts, and the only property of the lubricant involved is its viscosity. Hydrodynamic lubrication is present between two converging surfaces, moving at relatively high speed in relation to each other and withstanding a limited load, whenever an oil film can be formed. This type of lubrication is encountered

in engine bearings, between piston skirt and cylinder liner, and between piston rings and liners for high sliding velocities.

Hydrodynamic lubrication breaks down when the thickness of the fluid film becomes about the same as the height of the surface asperities. To the viscous friction is then added metal-to-metal solid friction at the peaks of the asperities. Both hydrodynamic and boundary lubrication conditions coexist. The surface texture controls this transition from hydrodynamic to mixed lubrication: rougher surfaces make the transition at lower loads. Abrupt load or speed variations or mechanical vibration may cause this transition to occur. This phenomenon occurs in connecting rod and crankshaft bearings where periodic metal-to-metal contact results from breaks in the oil film. The contact area between rings and cylinders is a region where, due to sudden variations in speed, load, and temperature, lubrication is of the mixed type. Intermittent metal-to-metal contacts occur as the result of breaks in the oil film. [3]

13.3.2 Turbulent Dissipation

Part of the total friction work is spent on pumping fluids through flow restrictions. The cylinder gases, cooling water, and oil are pumped through the engine; the fan pumps air through the engine's radiator. Work is dissipated driving these turbulent viscous fluid flows. The pressure difference required to pump these fluids around their flow paths is proportional to $\rho_v{}^2$, where v is a representative fluid velocity. The proportionality constant depends primarily on the details of the flow-path geometry. Hence the friction forces associated with fluid pumping will be proportional to N^2 (or $\overline{S}_p^2$ if the piston motion forces the flow).

13.3.3 Total Friction

The work per cycle for each component i of the total friction is given by integrating the friction force $F_{f,\,i}$ times its displacement dx around the cycle:

$$W_{f,i} = \int F_{f,i}(\theta)\, dx$$

The friction force components are either independent of speed (boundary friction), proportional to speed (hydrodynamic friction) or to speed squared (turbulent dissipation), or some combination of these. It follows that the total friction work per cycle (and thus the friction mean effective pressure) for a given engine geometry engine will vary with speed according to

$$W_{tf}\,(\text{or tfmep}) = C_1 + C_2 N + C_3 N^2 \qquad (13.5)$$

Some of the components of hydrodynamic lubrication friction and turbulent dissipation will be dependent on mean piston speed rather than crankshaft rotational speed N. Examples are piston skirt and ring friction, and the pressure losses associated with gas flow through the inlet and exhaust valves. For conventional engine geometries, crankshaft rotational speed is usually used to scale the total friction data rather than mean piston speed. [4, 5]

13.4 MEASUREMENT METHODS

A true measurement of friction in a firing engine can only be obtained by subtracting the brake power from the indicated power determined from accurate measurements of cylinder pressure throughout the cycle. However, this method is not easy to use on multi-cylinder engines, both because of cylinder-to-cylinder differences in indicated power and due to the difficulties in obtaining sufficiently accurate cylinder pressure data from each cylinder. As a result, friction is often measured in a motored engine. Friction components in a firing and a motored engine are different for the reasons outlined below. First, the common measurement methods will be described.

1. *Measurement of fmep from imep*. The gross indicated mean effective pressure is obtained from $\int pdV$ over the compression and expansion strokes for a four-stroke engine, and over the whole cycle for a two-stroke engine. This requires accurate and in-phase pressure and volume data. Accurate pressure versus crank angle data must be obtained from each cylinder with a pressure transducer and crank angle indicator. Volume versus crank angle values can be calculated. Great care must be exercised if accurate imep data are to be

obtained. [4] Both imep *g* and pmep are obtained from the *p-V* data. By subtracting the brake mean effective pressure, the combined mechanical/rubbing friction plus auxiliary requirements, mfmep + amep, are obtained.

2. *Direct motoring tests*. Direct motoring of the engine, under conditions as close as possible to firing, is another method used for estimating friction losses. Engine temperatures should be maintained as close to normal operating temperatures as possible. This can be done either by heating the water and oil flows or by conducting a "grab" motoring test where the engine is switched rapidly from firing to motored operation. The power required to motor the engine includes the pumping power. In tests on SI engines at part-load, the throttle setting is left unchanged. "Motoring" tests on a progressively disassembled engine can be used to identify the contribution that each major component of the engine makes to the total friction losses.
3. *Willans line*. An approximate equivalent of the direct motoring test for diesel engines is the Willans line method. A plot of fuel consumption versus brake output obtained from engine tests at a fixed speed is extrapolated back to zero fuel consumption. An example is shown in Fig. 13.6. Generally, the plot has a slight curve, making accurate extrapolation difficult. Agreement with a motored test result is shown.

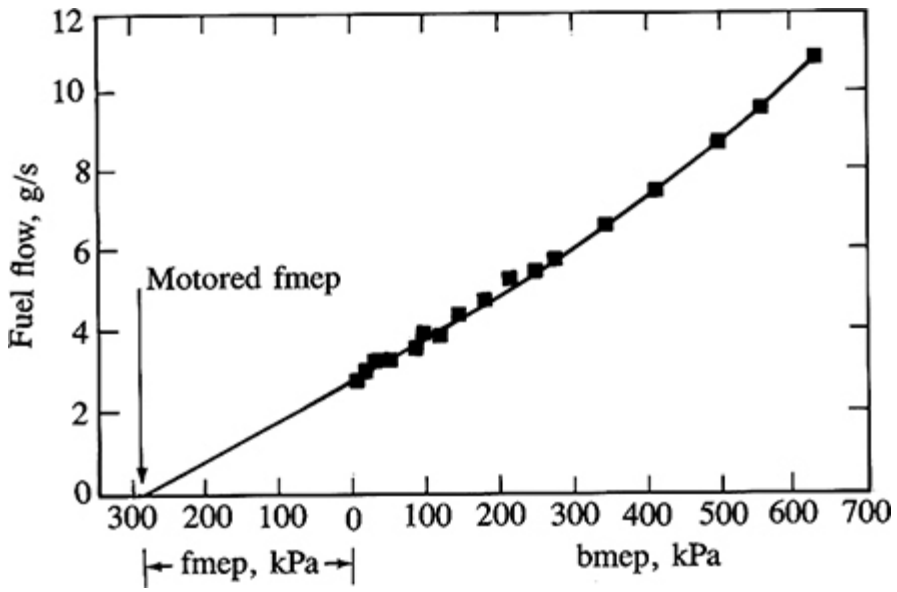


Figure 13.6 Willans line method for determining friction mean effective pressure.[6]

4. *Morse test*. In the Morse test, individual cylinders in a multi-cylinder engine are cut out from firing, and the reduction in brake torque is determined while maintaining the same engine speed. The remaining cylinders drive the cylinder cut out. Care must be taken to determine that the action of cutting out one cylinder does not significantly disturb the fuel or mixture flow to the others.

Only the first of these four methods has the potential for measuring the true friction of an operating engine. The last three methods measure the power requirements to motor the engine. The motoring losses are different from the firing losses for the following reasons:

- Only the compression pressure and not the full firing pressure acts on the piston, piston rings, and bearings. The lower gas loadings during motoring lower the rubbing friction.
- Piston and cylinder bore temperatures are lower in motored operation. This results in higher lubricant viscosity and therefore increased viscous friction. In addition, piston-cylinder clearances are greater during motoring operation, which tends to make friction lower. However, in firing operation, the lubrication of the top ring near the top of the stroke is inadequate to maintain normal hydrodynamic lubrication with the higher gas pressures behind the ring. The resulting boundary friction in this region makes friction in the firing engine higher. Overall, the net effect of lower piston and cylinder temperatures during motoring is unclear.
- In motored operation, the exhaust blowdown phase is missing and the gases discharged in the latter part of the exhaust stroke have a higher density than under firing conditions. These effects can result in different pumping work.
- When motoring, net work is done during the compression and expansion strokes because of heat loss from the gas to the walls, and because of gas loss through blowby. This work is not part of the true total friction work in a firing engine and should not be deducted from the indicated work of the firing engine to obtain the brake work; heat

losses and blowby are additional energy transfers to the indicated work, friction work, and brake work.

Overall, there are both positive and negative differences between firing and motoring engine friction and the net difference is surprisingly small. Thus motoring tests are widely used to estimate total and also individual components of friction. Note, however, that reasonable agreement for total engine friction does not imply equally good agreement for each of the friction components.

13.5 ENGINE FRICTION DATA

13.5.1 SI Engines

Figure 13.7 shows total motored friction mep for several four-stroke cycle four-cylinder SI engines between 845 and 2000 cm^3 displacement, at wide-open throttle, as a function of engine speed. These data (from older engines) indicate that, for a range of cylinder sizes and geometries, friction mean effective pressure is an appropriate normalized measure of engine friction. [7]

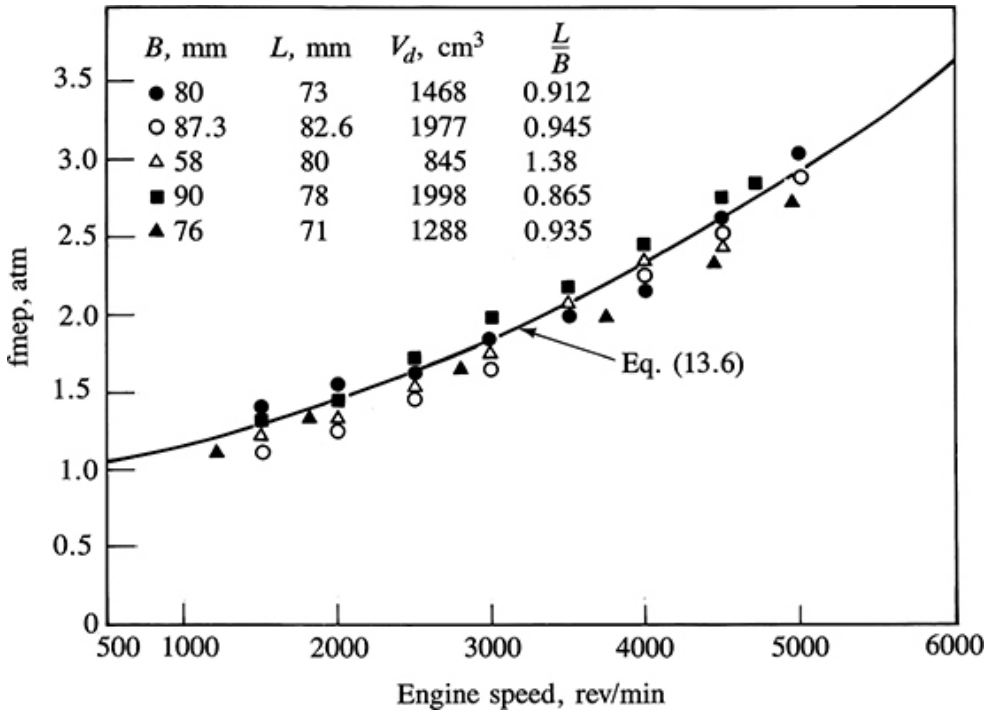


Figure 13.7 Friction mean effective pressure under motored conditions at wide-open throttle for several four-cylinder spark-ignition engines.[7]

Figure 13.8 shows total friction mean effective pressure for a more recent 5.4-liter V-8 naturally-aspirated gasoline engine (with a single-overhead-cam 2-valve-per-cylinder valvetrain) as a function of engine speed at wide-open throttle. It also shows the mechanical plus accessory friction for a similar 3.0-liter V-6 engine (four valves per cylinder, with dual-overhead-cam roller finger follower). The markedly different slopes of these two curves is due to the increasing pumping gas-exchange losses with rising engine speed at WOT. At low engine speeds, WOT pumping losses are small and the two fmep curves have similar magnitude. However, pmep rises rapidly with the square of piston (and thus engine) speed as speed increases. The importance of avoiding high engine speeds in the interests of good mechanical efficiency is evident. Under normal automobile engine operating conditions, a reduction in total friction mean effective pressure by about 10 kPa results in about a 2% improvement in vehicle fuel consumption. [7]

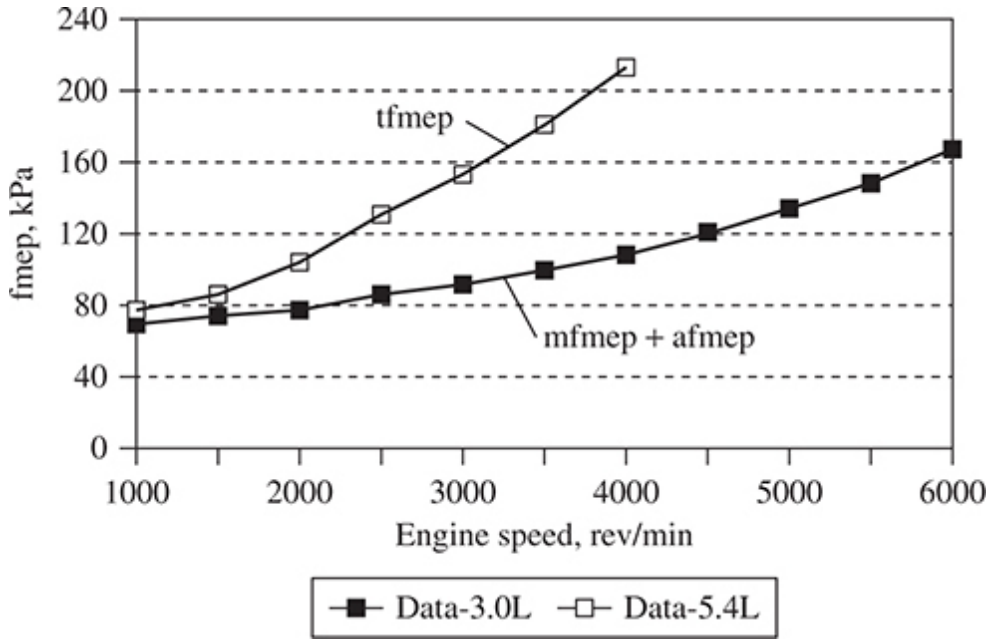


Figure 13.8 Friction mean effective pressure for two naturally-aspirated spark-ignition engines, illustrating the importance of pumping friction: 5.4-liter, V-8, two-valve-per-cylinder engine (tfmep); 3.0-liter, V-6, four-valve-per-cylinder, valvetrain roller-follower engine.[8]

The WOT total friction mean effective pressure data in Fig. 13.8 (for the 5.4-liter engine) is well fitted by the correlation.

$$\text{tfmep(kPa)} = 57 + 15\left(\frac{N}{1000}\right) + 5.5\left(\frac{N}{1000}\right)^2 \tag{13.6}$$

where N is in revolutions per minute. A comparison of this correction with an equivalent correlation from the 1980s indicates that total friction mean effective pressure has decreased substantially over the past 20 years: by 30 to 40% at low engine speeds, 20 to 25% at mid-speeds, and 10 to 15% at higher speeds. At lower speeds, valvetrain friction has been halved; at high speeds, flow resistances have been significantly reduced.

The relative importance of friction is quantified by the mechanical efficiency (η_m = bmep/imep). In SI engines, this increases from zero when the engine is idling, to a value close to 0.9 (at low to mid-engine speeds) at high loads. This behavior, and the decreasing relative importance of pumping friction (pmep) as load increases, are shown in Fig. 13.9 for a spark-ignition engine operating at a fixed (mid) speed.

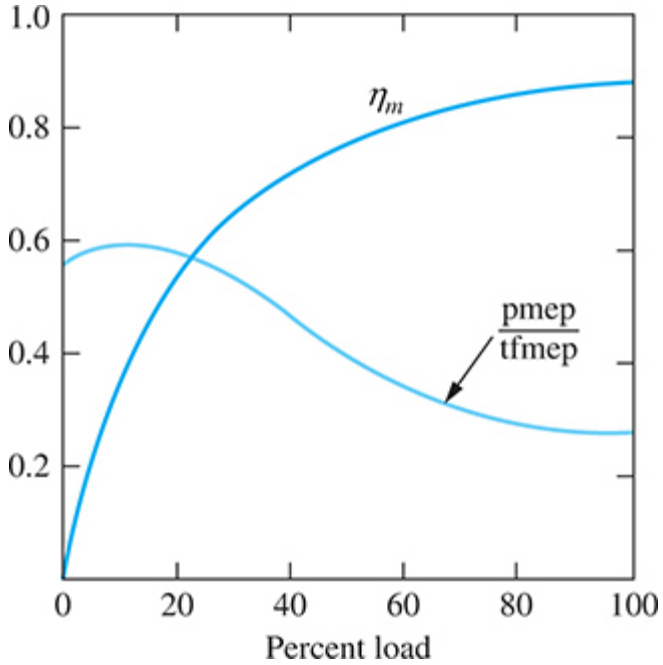


Figure 13.9 Mechanical efficiency η_m and ratio of pumping mep to total friction mep as a function of load for a typical spark-ignition engine at fixed (mid) speed.[2]

13.5.2 Diesel Engines

Figure 13.10 shows total friction as determined from motoring tests for direct-injection CI engines. Correlations for a typical diesel of each type are shown, of the form:

$$\text{Motoring mep (kPa)} = C_1 + 48\left(\frac{N}{1000}\right) + 0.4\bar{S}_p^2 \tag{13.7}$$

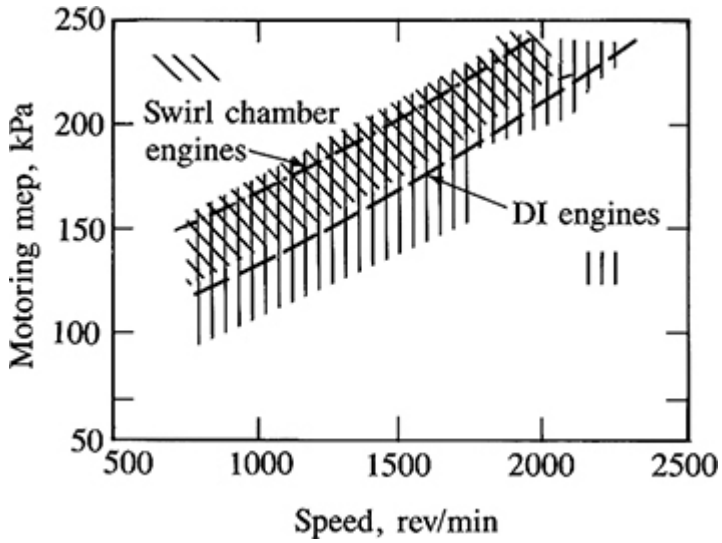


Figure 13.10 Motored total friction mean effective pressure as a function of speed for several DI diesels (bores in range 100 to 137 mm) and IDI swirl-chamber diesels (bores in range 100 to 121 mm). Correlations for r_c = 15 and L = 142 mm (DI engine) and r_c = 16 and L = 142 mm (IDI engine). [6]

where N is in revolutions per minute and $\bar{S}_p$ in meters per second. For the direct-injection engine C_1 = 75 kPa. Mean piston speed was found to give a better correlation than does engine speed (N) for the last term in Eq. (13.7) which is mainly pumping mep.

Pumping mean effective pressure data for several naturally-aspirated diesel engines under both firing and motored conditions is shown in Fig. 13.11. The solid curve is the term 0.4 $\bar{S}_p^2$ in Eq. (13.7) , with $\bar{S}_p$ in m/s which fits the data.

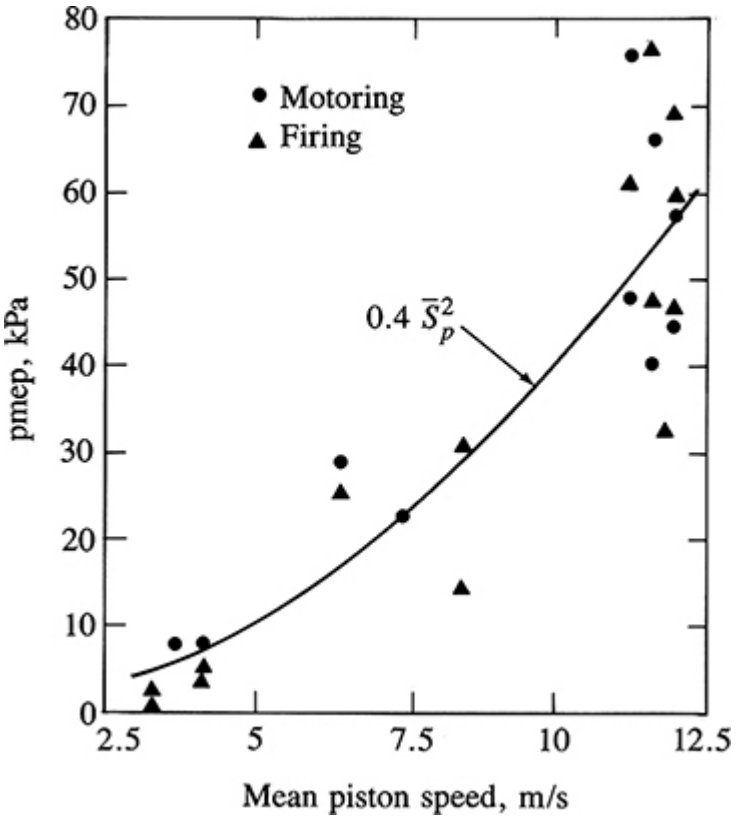


Figure 13.11 Pumping mean effective pressure as a function of mean piston speed for several naturally-aspirated diesel engines.[6]

13.6 MECHANICAL FRICTION COMPONENTS

In this section, a more detailed analysis of the major components of mechanical engine friction is presented, along with an overview of the engine's lubrication system. Pumping work and engine accessory friction are then discussed in Secs. 13.7 and 13.8.

13.6.1 Motored Engine Breakdown Tests

Motored engine tests, where the engine is disassembled or broken down in stages, can be used to determine the friction associated with each major engine assembly. While this test procedure does not duplicate the combustion-generated forces of actual engine operation, such tests are useful for assessing the relative importance of individual friction components. The

results of breakdown tests on various spark-ignition engines, separating out the contribution to mechanical friction from the piston assembly (piston, rings, connecting rod, the valvetrain, crankshaft bearings and seals), indicate a breakdown of the overall mechanical friction as: piston assembly 60%; valvetrain 25%; crankshaft 15%. Note, however, that their relative importance does vary over the engine's speed range. A typical distribution (at light load and low engine speed) of total engine friction into its various categories for an automobile gasoline naturally-aspirated engine is shown in Table 13.1. The percentage breakdown is shown for mechanical plus accessory friction, and also with pumping work included. At this light load typical of passenger car urban driving, the pumping work losses (due primarily to throttling the engine intake flow) with a manifold pressure of about 0.5 bar is some 30%. At these operating conditions, the engine brake fuel conversion efficiency would be about 20%.

TABLE 13.1 Typical friction distribution: gasoline engine at light load [9]

Crankshaft	15	11
Piston, connecting rods	33	23
Valvetrain	28	20
Oil pump	15	11
Water pump	9	6
Subtotal	100	
Pumping work		~30
Total		100

More detailed motoring friction breakdowns for a gasoline SI engine and a diesel engine are shown in Figs. 13.12 and 13.13, respectively. For the four-cylinder gasoline engine, the largest components are the (motored) pumping work, which rises significantly as speed increases, and the mechanical (rubbing) friction components of the piston, rings, connecting rod assembly.

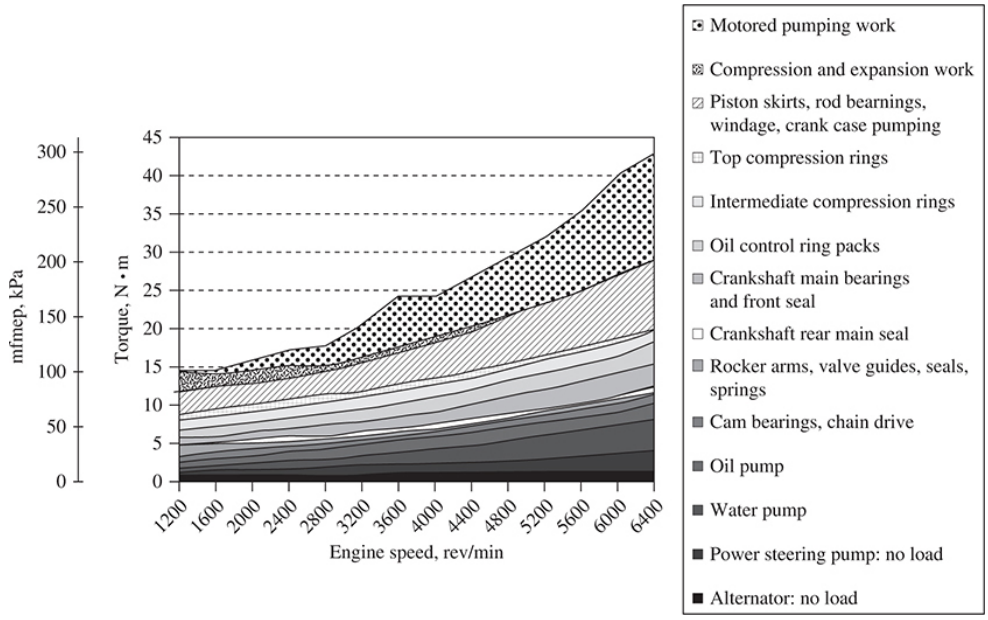


Figure 13.12 Motored friction breakdown (mfmep) for a 1.6-liter four-cylinder four-valve spark-ignition engine as a function of speed; oil and coolant temperature 88°C.[10]

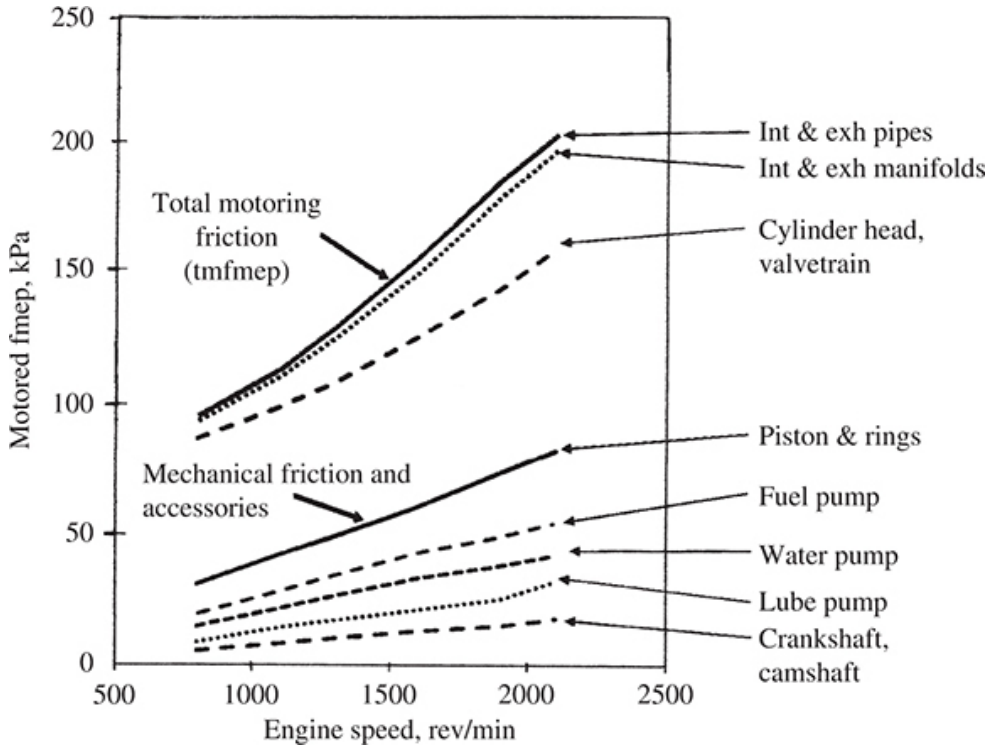


Figure 13.13 Pumping, mechanical friction, and accessory losses from a motoring tear-down test of a heavy-duty diesel engine.[11]

Figure 13.13 shows friction mean effective pressure data from a motored tear-down test of a heavy-duty diesel engine, as a function of speed. Flow losses in the total intake and exhaust systems are significant at the higher engine speeds, especially across the ports and valves in the cylinder heads, as are the mechanical friction losses in the piston, rings, and con rods.

13.6.2 Engine Lubrication System

The lubrication of the sliding surfaces in the moving components of the engine is an essential factor in determining its mechanical friction. Figure 13.14 shows the various lubricated mechanical components of the engine and identifies the sliding contact lubrication regimes under which they operate—boundary, mixed, and hydrodynamic (see Sec. 13.3.1). Boundary lubrication friction, which is much higher than fully hydrodynamic friction, occurs when rotating and sliding components come to rest, and when the (normal) forces become high. Where shafts rotate continuously in journal bearings, a fully supporting oil layer (hydrodynamic lubrication) can normally be maintained except when the engine starts or stops: examples are crankshaft, camshaft, and con rod big-end bearings, cam roller followers driven by cam lobes, teeth contact in gear-drive systems. The potential for boundary lubrication and thus surface-to-surface contact exists between piston rings, piston skirt, and the cylinder liner; also in various parts of the valve drive system. Whether or not surface contact (and thus boundary friction) occurs depends on whether a supporting oil film exists (as, say, when the piston comes to rest at TC or BC) and whether that film is squeezed out of the thin surface-to-surface gap under the loads being applied. [12]

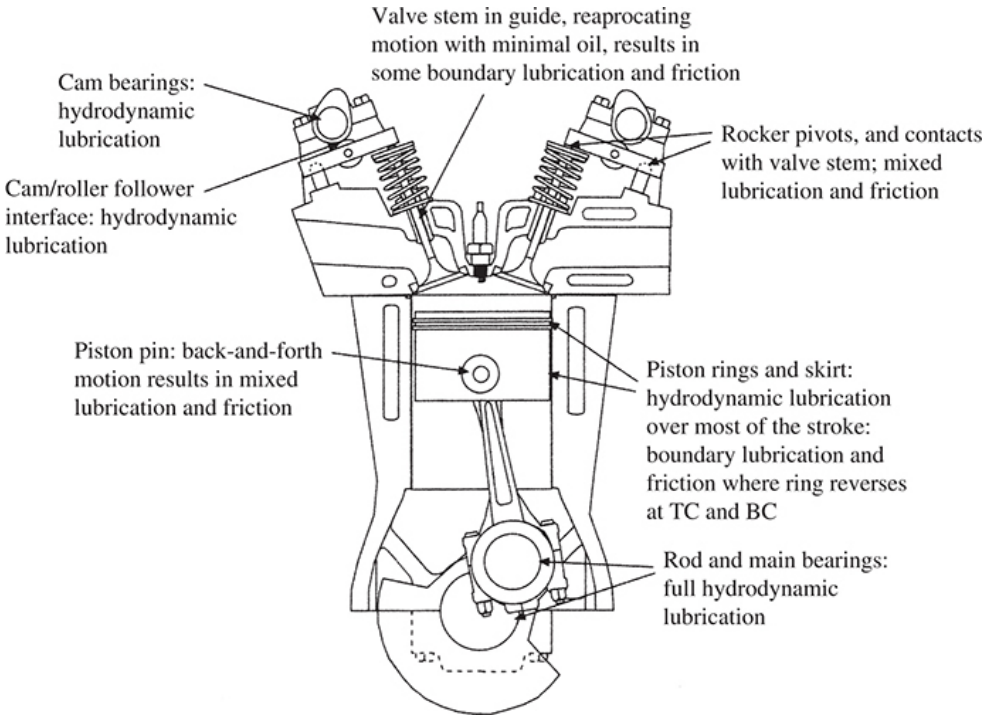


Figure 13.14 Engine components where relative motion between two surfaces occurs, and the lubrication and friction regimes which then prevail: boundary, mixed, and hydrodynamic.[12]

The principle moving parts of an engine, identified in Fig. 13.14, are positively lubricated by introducing a supply of oil from a pressurized system. An example of an engine lubrication system is shown in Fig. 13.15. The oil pump draws oil from the engine sump through a suction strainer and then delivers it through a control valve to the oil cooler (not shown in figure). The oil then passes through the filter to the main oil gallery. From the main oil gallery, it is branched to the main crankshaft bearings to the con rod big end bearings. In parallel, oil is directed to the camshaft bearings. Through a passage in these bearings, the oil flows to the valve tappets. As the oil passages of tappets and tappet bridges line up during tappet motion, rocker arms and valve stems are pulse-lubricated through the tappets (and pushrods, if present). For cooling pistons and lubricating cylinders, oil is thrown against the underside of the piston either through oil splashing from the crank or (for greater cooling) through nozzles connected to the main bearings. The gears of the main timing train are splash-lubricated. Oil, from

these lubricated engine regions, is then returned to the sump. Lubricant properties are reviewed in Sec. 13.10.2 .

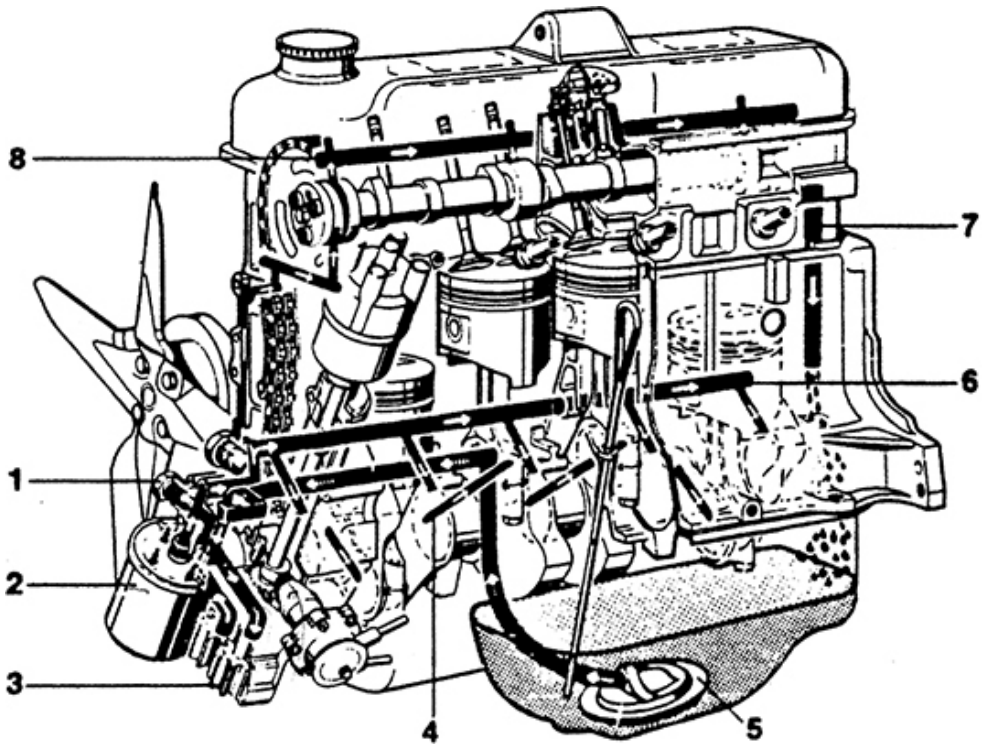


Figure 13.15 Layout for a forced-feed engine lubrication system: (1) Pressure relief valve; (2) Oil filter; (3) Gear pump; (4) Connection from main bearing to connecting-rod bearing; (5) Suction strainer; (6) Main pressurized oil line to crankshaft bearings; (7) Return flow from timing-gear case to crankcase; (8) Oil passage to camshaft bearings.[13] (*Courtesy Robert Bosch GmbH and SAE.*)

Mechanical engine friction depends on the temperature of the lubricating oil in the various lubricated regions of the engine, since the oil's viscosity is a function of temperature. When an engine is started cold, that viscosity will be significantly higher because the component and oil temperature will be close to ambient air temperature. As the engine then warms up, component, coolant, and oil temperatures rise to their steady-state values (which depend to a degree on engine load and speed). The impact of oil temperature on the engine's mechanical friction is substantial. As shown in Fig. 13.16, the total friction power loss decreases by a factor of two as the engine coolant temperature increases from 20 to about 90°C, where it then stabilizes. Fmep

was determined from the difference between imep (obtained from cylinder pressure data) and the measured bmep. [14] The coolant temperature (used as an indicator for the local lubricant temperature) is estimated from a friction model. [8] During warm-up, the engine was operated at light load and low speed. Engine warm-up at these lightly loaded (normal driving) conditions takes some 10 minutes.

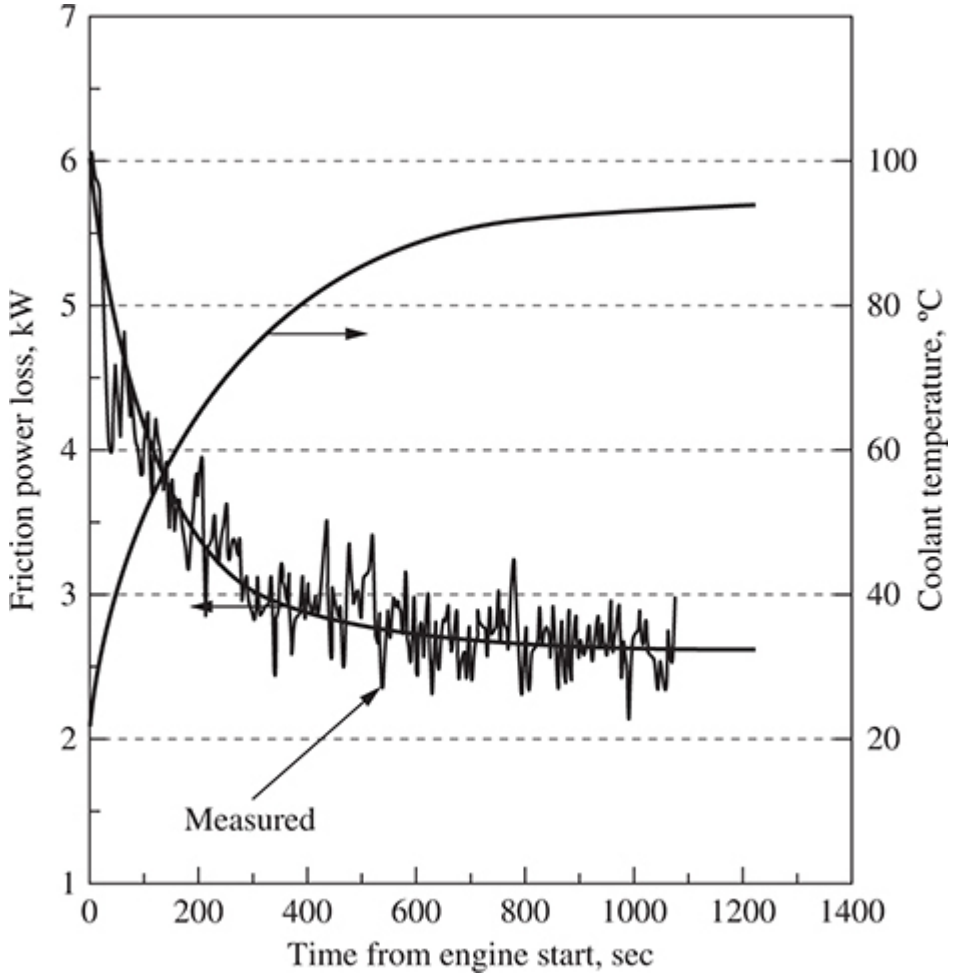


Figure 13.16 Measured coolant temperature and friction power loss as a function of time from engine start up. 1.1-liter four-cylinder gasoline engine. [14]

In the sections that follow, total engine friction will be discussed under the following headings: piston assembly friction, valvetrain friction, crankshaft bearing friction, pumping friction (in Sec. 13.7), and accessory power requirements (in Sec. 13.8).

13.6.3 Piston Assembly Friction and Lubrication

The construction and nomenclature of the piston and ring assembly for a typical automobile gasoline and truck diesel engine are shown in Fig. 13.17. The piston skirt is the load-bearing surface, which keeps the piston properly aligned within the cylinder bore. The piston skirt carries the side load, which is present when the connecting rod is at an angle to the cylinder axis. The rings control the behavior of the lubricant between the piston and ring surfaces and the liner. Automotive engines normally use three rings. These rings perform the following tasks: (1) The top ring seals the clearance between the piston and cylinder to retain in-cylinder gas pressure and minimize blowby; (2) the second ring (often called the oil scraper ring) controls the amount of oil on the liner above it so the top ring is adequately (but not excessively) lubricated; and (3) the bottom, oil control, ring controls the oil flow from the region below the ring pack into the third land region above this bottom ring. Effectively, the oil control ring does the gross oil flow management (there is plenty of oil below this ring) and the oil scraper ring does the fine control of oil left on the liner as the piston and its rings move down the cylinder. The back of the oil control ring groove has holes in it so excess oil can be returned to the crankcase.

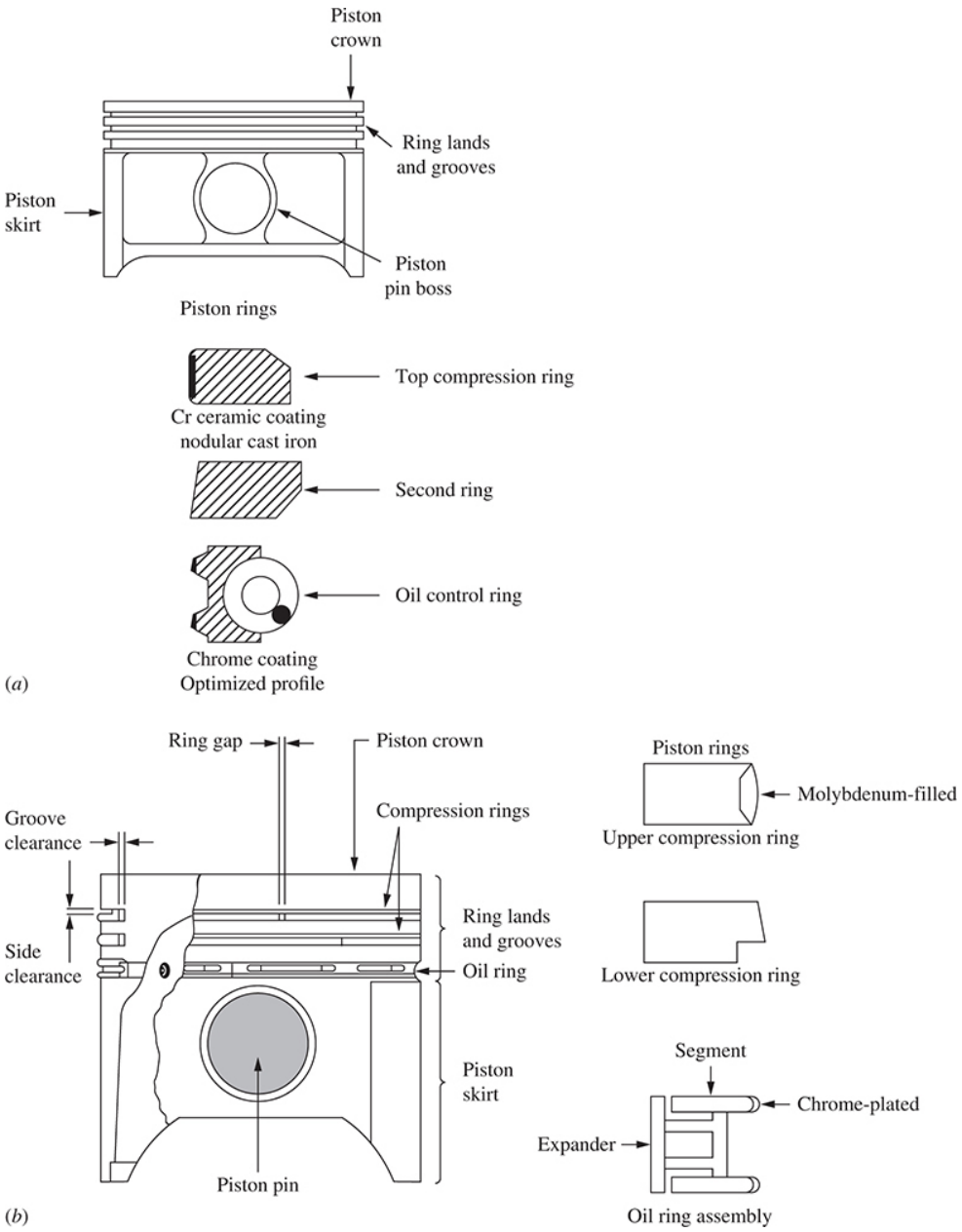


Figure 13.17 Construction and nomenclature of typical piston and ring assemblies: (*a*) spark-ignition gasoline engine; (*b*) diesel engine (various sources).

Many designs of compression ring are employed, the differences between them being in the cross-sectional shapes (and hence relative flexibility) and in their use of wear-resistant surface treatments. Top compression rings are usually made of nodular cast iron (or steel). The ring profiles are chosen to facilitate hydrodynamic lubrication. Common shapes are a rectangular cross-

section with inner and outer edges chamfered to prevent sticking in the groove, with a barrel-shaped working surface which can accommodate the rotation of the piston which occurs with short piston skirts (such as that shown in Fig. 13.17 *a*). Wear-resistant coatings (either a hard chromium-plated overlay or a molybdenum-filled inlay) are usually applied to the outer ring surface. The rings are forced outward onto the liner surface through the ring tension produced in the ring as the ring gap is largely closed when the ring is inserted into the circular cross-section liner. The cut-off corner in the automobile top and second ring profiles in Fig. 13.17 *a* causes the ring to *twist*. For the top compression ring shown, the chamfer at the top right-hand corner reduces the radial width of the top of this ring and thus makes its *stiffness* at the top less than at the bottom. A positive (anti-clockwise for the top ring profile in Fig. 13.17 *a*) ring twist results. The second ring serves principally to fine tune the amount of oil left on the liner (though it also provides additional pressure sealing). Since the operating environment is less arduous, the second ring can be made more flexible to give better oil control through its "oil scraping" profile, and its chamfer that produces negative twist. Note these ring twists are small—fractions of a degree. Ring bottom-edge contact provides the desired oil scraping action on the down-stroke. The oil control ring meters and distributes the oil directed onto the cylinder liner by the crankshaft system, returning excess oil to the crankcase sump. It must exert sufficient pressure against the cylinder liner, possess suitably shaped wiping edges (usually two thin steel rings), and provide adequate oil drainage. Slotted or composite rings are normally used.

The tension in all the piston rings holds them out against the cylinder wall, and cylinder bore deformations require high ring conformability. The gas pressure behind the compression rings increases this radial force. All these factors contribute to piston assembly friction. The gas pressures behind the second ring are substantially lower than behind the first ring. The gas pressures behind the rings are a function of engine load. All three rings contribute significantly to piston assembly friction. The oil control ring, due to its two rail design and higher ring tension, may operate partly under boundary lubrication and contribute about twice the friction of each compression ring. [15] Ring pack friction mep is in the range of 4 to 12 kPa.

The piston assembly is the dominant source of engine rubbing friction. The components that contribute to friction are: the rings (as noted above), piston skirt, and piston pin. The forces acting on the piston assembly include:

static ring tension (which depends on ring design and materials); the gas pressure forces (which depend on engine load); the inertia forces (which are related to component mass and engine speed). The major design factors which influence piston assembly friction are the following: ring width, ring face profile, ring tension, ring gap (which governs inter-ring gas pressure), liner temperature, ring-land width and clearances, skirt geometry, skirt-bore clearance.

Piston assembly friction is dominated by the ring friction. The forces acting on a compression ring, lubricated by a thin oil film, as it moves down the cylinder liner during the expansion stroke are indicated in Fig. 13.18. The analysis of this hydrodynamic contact is complex because the forces acting on the ring vary with time and slight changes in ring face geometry can have large effects on the computed results. Cylinder pressure p_c normally acts on the top and back of the ring. The inter-ring gas pressure p_{ir}, which depends on cylinder pressure and the geometry of the lands, ring grooves, and ring (especially the ring gap), acts on the oil film and bottom part of the ring. The character of the gas flow into and out of the inter-ring regions and its effect on ring motion were discussed in Sec. 8.6 . Late in the expansion stroke, pressure reversals can occur which may cause the ring to move to the upper surface of the groove or to flutter in between. Ring tension t_r acts to force the ring against the liner. The pressure in the lubricating oil film generated by the downward ring motion is as shown by the surface *A*-*B*. The oil film can cavitate between *B* and *C*; if so, the pressure decreases to a low value and then increases to p_c. When the direction of motion is reversed, *C*-*B* becomes the pressure-generating surface. The pressure generated in the lubricant film holds the ring surface away from the liner.

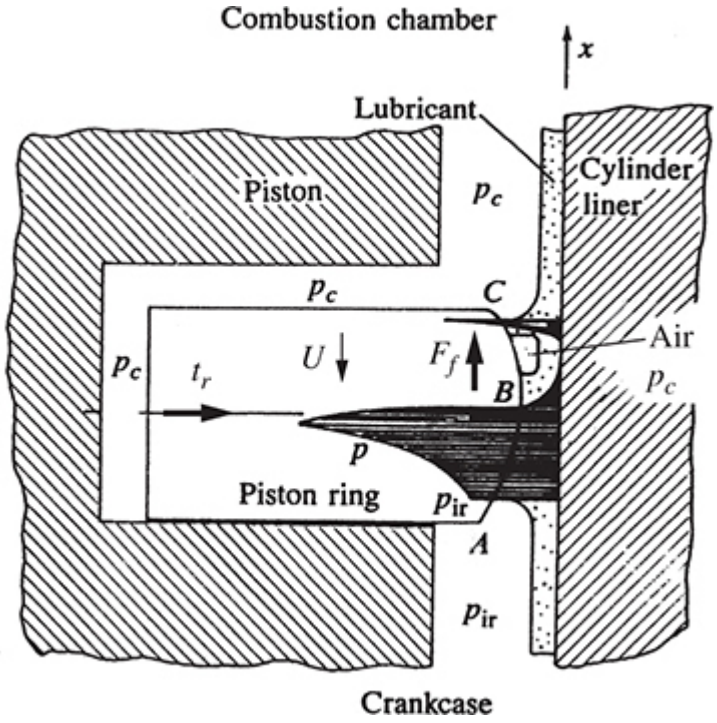


Figure 13.18 Schematic of pressure distribution in the lubricating oil film and around a compression ring during expansion stroke. Pressure profile in the oil film indicated by horizontal shading. [2]

Models for the ring and oil film behavior have been developed. For the practical case where the oil film thickness h is much less than the ring width, the Navier-Stokes equation for the liquid film motion reduces to a one-dimensional Reynolds equation of the form:

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) = 6U\mu \frac{\partial h}{\partial x} + 12\mu \frac{\partial h}{\partial t} \tag{13.8}$$

where h is the local film thickness, m the liquid viscosity, and U the relative velocity between the two surfaces. This equation, along with the appropriate force balances on the ring, can then be solved for the coupled film and ring behavior.

Oil film thickness measurements within the piston assembly illustrate many of the above-described phenomena. Figure 13.19 shows results obtained with a point laser-induced-fluorescence (LIF) measurement technique (where a fluorescing dye is added to the lubricating oil) of the oil

film thickness between the piston rings and the liner, and of the oil accumulation on the piston lands and skirt, in an automobile gasoline engine. The piston profile is shown along the bottom of the graph, with the locations of the top ring groove, the scraper ring groove, and the oil control ring groove marked. The piston skirt is on the left-hand side. Two 25-μm-deep grooves used for measurement calibration are positioned just above the piston skirt. Oil film thicknesses of 1 to 2 μm are observed above the top of the piston (between 0 and +5 mm on the right-hand end of the horizontal axis). This film, left on the liner by the descending piston, lubricates the top rings on the piston upstroke. Oil film thicknesses between the liner and the two top rings have a minimum of some 1.5 μm. Elsewhere, primarily on the piston lands, the descending two rails of the oil control ring carry substantial oil from the piston skirt region downward, as does the second oil scraper ring. The liner surface temperature, and thus the oil temperature, does not affect this oil distribution pattern significantly. These oil distributions within the piston and ring assembly have a major impact on oil consumption (Sec. 13.10.1).[16]

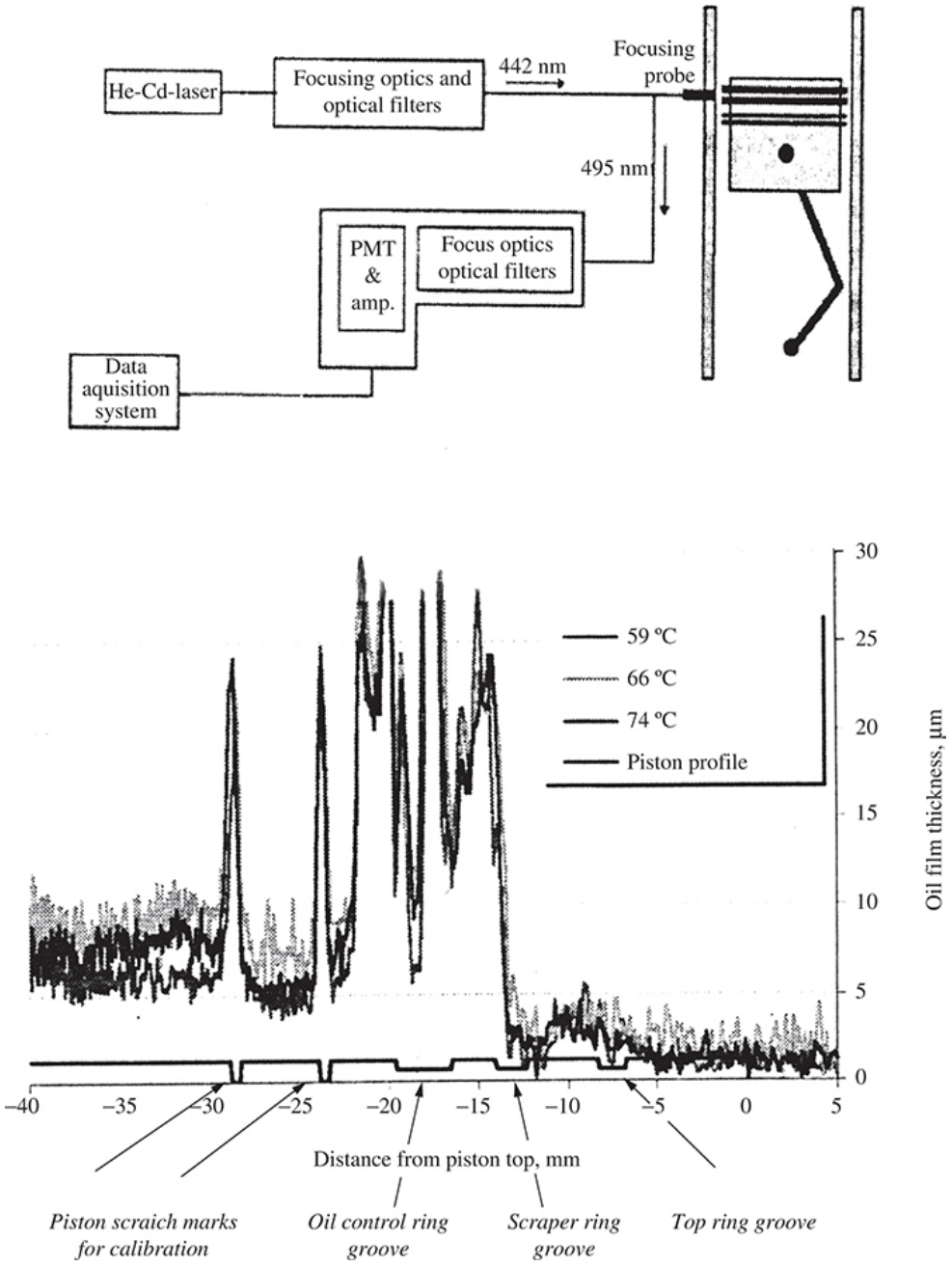


Figure 13.19 Oil film thickness measurements between the piston, piston rings, and the liner made using a laser-induced fluorescence technique illustrated at top. Piston profile is along the bottom. Curves show the oil film thickness in microns at different liner temperatures in an operating gasoline spark-ignition engine. [16]

Figure 13.20 illustrates key aspects of piston ring behavior during the engine's operating cycle. The engine is a heavy-duty 8.9-liter diesel with a bore of 114 mm, stroke 144.5 mm, and compression ratio of 16.6 operating at its rated power. [17] The top graph (Fig. 13.20) shows the gas pressures above the top ring (which equals the cylinder pressure), between the top and second rings, and below the second ring. Note that during the latter half of the expansion stroke, the second land pressure is essentially the same as the pressure above the top ring. This is generally undesirable since there is no longer a significant pressure force holding the top ring on the bottom of its groove (see Fig. 13.18). During this period, as shown in Fig. 13.20, the top ring can oscillate up and down in its groove as the sketches and the simulation results for relative ring lift indicate. A consequence of this top ring *flutter* and the higher gas pressure below the ring is an oscillating (reverse) gas flow around the ring which can carry any oil available below the ring upward toward the combustion chamber, contributing to oil consumption (Sec. 13.10.1).

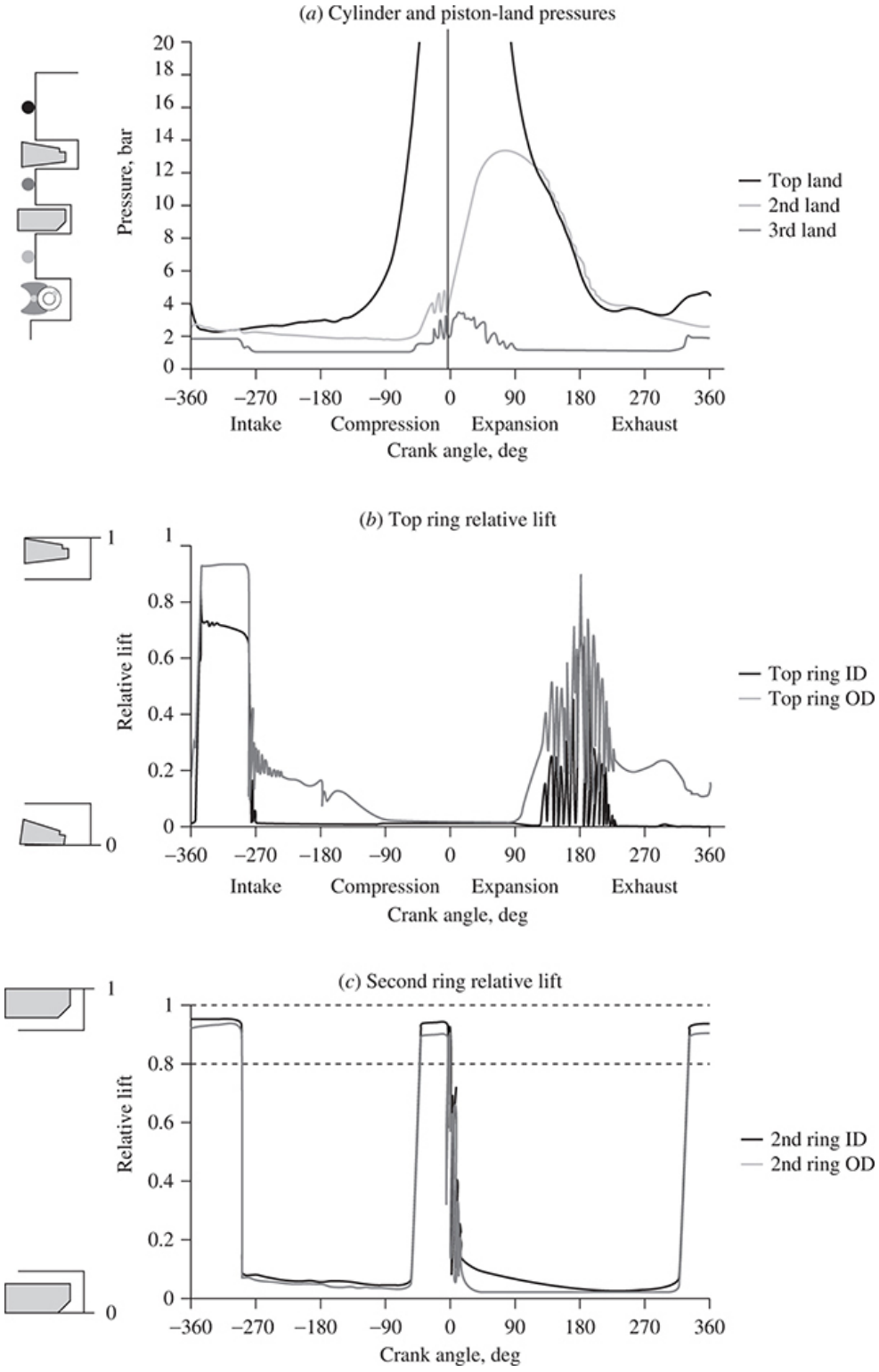


Figure 13.20 Piston ring motion, within the ring grooves, during the operating cycle in one cylinder of a 8.9-liter heavy-duty diesel engine: (*a*)

cylinder and piston land pressures; (*b*) top ring relative lift; (*c*) second ring relative lift.[17]

The second ring lift behavior shows upward axial movement in its groove around the top center firing piston location. This can be helpful in reducing the second land pressure by enabling downward gas flow behind the second ring. Note that this gas flow resulting from top and second ring motion and flutter in their respective grooves, is significantly larger than the flows through the (relatively small area) ring gaps. [17]

The piston rings also rotate, circumferentially, within their grooves. The top and second rings usually move independently of each other, and arbitrarily. Changing engine operating conditions generally alters the location of the ring gap. The rotation speed of the rings is in the 0.2 to 0.4 rev/min range for the top ring, and can be twice that for the second ring. Ring rotation is easier at low engine speeds. Note that usually the ring gaps of the top two rings are in different circumferential positions: the blowby gases that pass that flow axially downward through the top ring gap must therefore flow circumferentially around the second land region before they can flow through the second ring gap. When the ring gaps line up (as they at times may) the blowby flow path is more direct. See Sec. 8.8 for additional discussion of ring-pack gas flows. Flow through the ring gaps impacts oil consumption (Sec. 13.10.1).[18]

The desired ring behavior as the piston moves up and down the cylinder during the four-stroke operating cycle can be summarized as follows. [12] As the piston accelerates down the cylinder during the intake stroke, ring friction and inertia forces move the rings to the top of their grooves. During this downward stroke, the oil control and oil scraper rings “scrape” oil which has accumulated on the liner back into the crankcase. As the piston decelerates approaching BC, the inertia of the rings dominates: rings move to the bottom of their grooves. As the piston moves up during the compression, friction and inertia forces on the rings hold the rings on the bottom of their grooves. As the piston approaches TC, the increasing gas pressure above the top ring keeps it on the bottom of its groove: lower inter-ring pressures allow the second ring, in the latter part of the compression stroke, to move to the top of its groove as it decelerates. During this upward stroke, the rings are lubricated by the oil left on the liner during the last downward stroke.

During the expansion (power) stroke, the high cylinder gas pressure holds the top ring on the bottom of its groove: rising inter-ring gas pressure part way into the stroke forces the second ring to the bottom of its groove. Again, on downward strokes, the descending oil control and scraper rings "scrape" excess oil off the liner and direct it back to the crankcase. Early in the exhaust stroke the ring friction dominates, and the top and second rings ride on the bottom of their grooves. As the piston slows in the latter part of the stroke and gas pressure are low, inertia moves both rings to the top of their grooves. [12]

The cylinder liner and piston ring surface finish impacts the piston assembly oil behavior and friction. The liner is *honed* to improve the liner surface's capacity to retain oil: that is, a closely spaced cross-hatched pattern of grooves some 1.5 to 3 μm deep are machined into the liner to retain oil and also wear debris. A finer honing operation smooths the surface. Combined, these surface modifications reduce piston assembly friction, especially during engine wear-in operation.

An analysis of the forces involved in the piston, connecting rod, crankshaft mechanism of Fig. 2.1 leads to the following. An axial force balance relates the piston mass m and acceleration to the net axial force:

$$m\frac{dS_p}{dt} = -F_r\cos\phi + \frac{\pi B^2}{4}p \mp F_f \tag{13.9}$$

where ϕ is the angle between the cylinder axis and connecting rod, and p is the cylinder *gauge* pressure. The piston acceleration $a_p(= dS_p/dt)$ is given by Eq. (2.14) .

A transverse force balance gives

$$F_t = F_r\sin\phi = \left(-m\frac{dS_p}{dt} + \frac{\pi B^2}{4}p \mp F_f\right)\tan\phi \tag{13.10}$$

Here F_r is the force in the connecting rod (positive when in compression) and F_f is the friction force on the piston assembly (- when piston is moving toward the crank shaft; + when piston moves away from the crank shaft).

The side thrust F_t given by Eq. (13.10) is transmitted to the liner largely via the piston skirt. It changes direction as the piston passes through top- and

bottom-center positions. Since the friction force changes sign at these locations, and the gas pressure during expansion is greater than during compression, the side thrust during the expansion or power stroke is greater. The side of the liner resisting that greater force (the left side in Fig. 2.1) is termed the *major thrust side*. The other (right) side is termed the *minor thrust side*. The variation of this piston side force through the operating cycle in a turbocharged gasoline engine (peak cylinder pressure 115 bar, specific output 87 kW/liter) is shown in Fig. 13.21. [19] Data for a standard looser fitting piston are shown, as well as for a tighter-fitting low-friction lighter piston where both piston skirts exert some side force on the two sides of the liner. The high side skirt loading due to the combustion-generated high in-cylinder gas pressures starting at (firing) TC piston position are apparent.

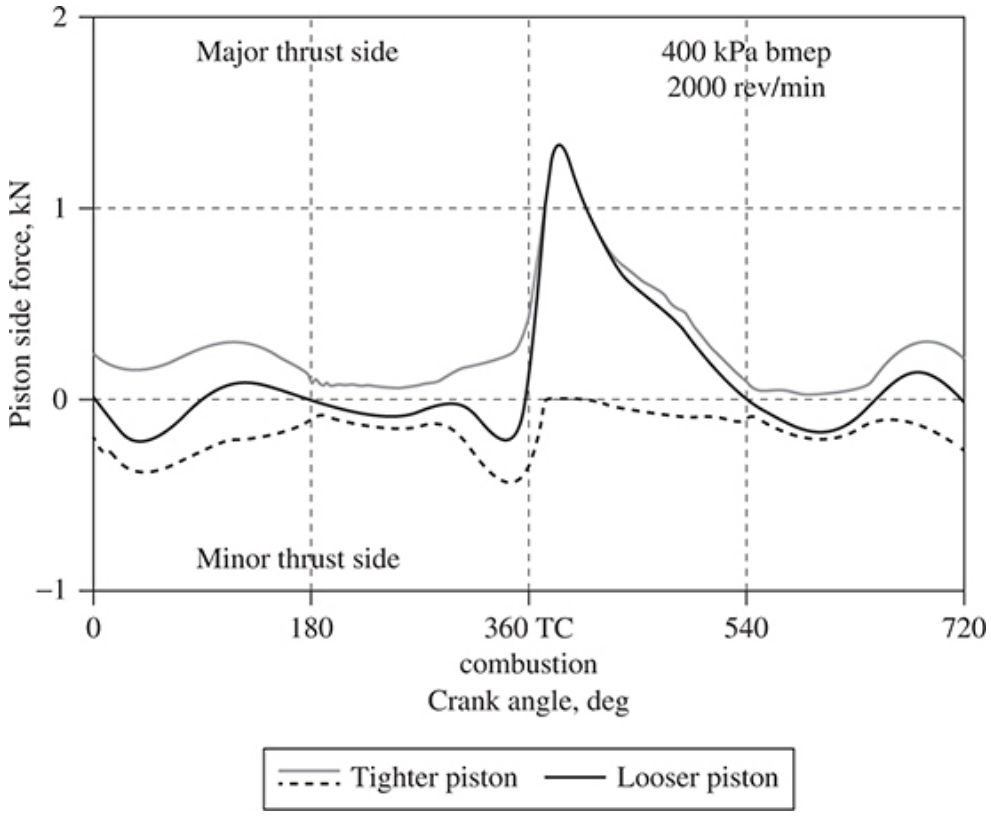


Figure 13.21 Piston side force through the operating cycle of a turbocharged gasoline engine: 400-kPa bmep, 2000 rev/min.[19]

The piston skirt carries most of this side thrust so it contributes to piston assembly friction. The large contact area between the skirt and liner, relative

to the ring contact area, results in lighter loading (force/area) and promotes hydrodynamic lubrication. Piston skirt areas have been reduced substantially in recent years to reduce piston height and mass (which reduces side thrust) and contact area. An additional reduction in side thrust leading to reduced skirt friction has been standard design practice through use of an offset wrist-pin. By offsetting the pin axis by about 1 mm toward the major thrust side without changing its vertical location, the crank angle at which the piston traverses the bore and "slaps" the other side of the cylinder is advanced to before TC, so it occurs before combustion has increased the cylinder gas pressure significantly.

Offsetting the crankshaft axis from the cylinder axis toward the major thrust side of the cylinder by some 15% of the bore can modestly reduce the piston side force and thus piston friction. This is done in some production gasoline engines. Also, changes in R (the crank ratio, $R = a/\ell$ see Sec. 2.2) change the piston side force, and friction. It has been noted that if the piston side force is below a certain level, then piston assembly friction is not significantly affected by the side force. So while decreasing the crank ratio below about 3.5 increases expansion stroke friction due to the increased side force, increasing R above 3.5 has little impact. Crankshaft axis offset does reduce piston fmep during the first portion of the expansion stroke (from about 30° to 90° ATC). A crank offset of about the optimum (0.1 to 0.15, crankshaft offset: connecting rod length) which is about 15 mm with a stroke of 80 mm, resulted in a 10% reduction in piston assembly friction. This reduction occurs not only from the reduced piston side force, but also from the changed timing of the piston movement from the minor thrust side to major thrust side of the liner, relative to TC. [20]

An additional factor in piston assembly friction and oil consumption is cylinder bore distortion. Mechanical forces, gas pressures, and acting on the engine block and cylinder head, and thermal expansion, can cause the cylinder liner to be "out of round." Various block design and manufacturing approaches are used to minimize the mechanical distortions that occur during engine assembly. The previous discussion of the intended ring function and behavior makes clear that significant distortion of the liners in multi-cylinder engines will increase friction and oil consumption.

Figure 13.22 illustrates how cylinder distortion is categorized: whether it is primarily second, third, or fourth order. Figure 13.22 *a* shows the block layout of a hypothetical engine where the cylinder liners are secured in

different ways. Cylinder liners 2 and 3 are constrained on either side, and thus are prone to second-order distortion. Cylinder 4 is effectively attached to the block at three circumferential locations and is susceptible to third-order distortions. Cylinder 1 has four clamping points and thus fourth-order distortions are possible. Figure 13.22 *b* shows these three orders of bore distortion.[21]

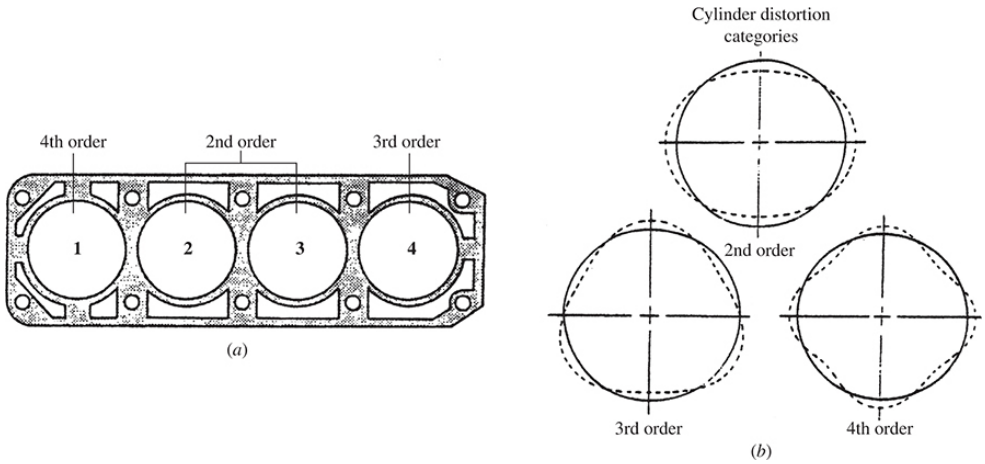


Figure 13.22 Definition of second-, third-, and fourth-order bore distortion, and illustration of block designs that are susceptible to these distortions.[21]

Maximum radial bore distortions in practice can be up to some ±30 μm Piston rings are designed to be flexible, and with adequate ring tension, to conform to liner distortions to a degree. Note second-order distortions are likely to be largest in amplitude; fourth-order distortions are significantly smaller. Since piston rings have a ring gap, their conformity to distorted bores is not circumferentially uniform.

A consequence of bore distortion and incomplete ring conformity is that the oil film thickness between the top ring (with its barrel-shaped outer surface) and the liner varies around the ring. Liner distortions can occur along the length of the liner. Ring conformity would be expected to vary with load (higher gas pressures force better conformity). Limited measurements of oil film thickness between the top ring and the liner show it can vary substantially (e.g., from less than 1 μm to 10, even 20, μm, depending on where the piston is in the 720° engine operating cycle). The oil film can vary

from normal behavior (covering the majority but not all the top ring surface) to “fully flooded” behavior, or to significantly “starved” behavior. [22]

Measurements of the friction force associated with the piston assembly have been made. The most common technique involves the use of a special engine where the axial force on the cylinder liner is measured directly with a load transducer. Piston assembly friction can also be determined from cylinder pressure and connecting rod reaction force variations (see Fig. 13.1). Figure 13.23 shows the friction force (fmep) measured in small-displacement turbocharged gasoline engine through the engine’s operating cycle. This data is from a lightly loaded lower speed operating point. Friction forces are highest just before and after top-center at the end of the compression stroke. The high values at the start of the expansion stroke under firing conditions are caused by the piston slap impulse, [21] and then by the piston skirt high side-thrust force as well as the combustion gas pressure loading on the rings, due to the high cylinder gas pressure (which peaks at 15 to 20° ATC). The figure also shows the piston friction mean effective pressure (friction work per cycle/displaced volume). When fmep is multiplied by N/2 (cycles per second) this gives friction power. When the piston comes to rest at TC and BC positions, the power lost (rate of doing work to overcome friction) goes to zero. The first half of the expansion stroke with its higher friction forces consumes more power than do the other strokes. [19]

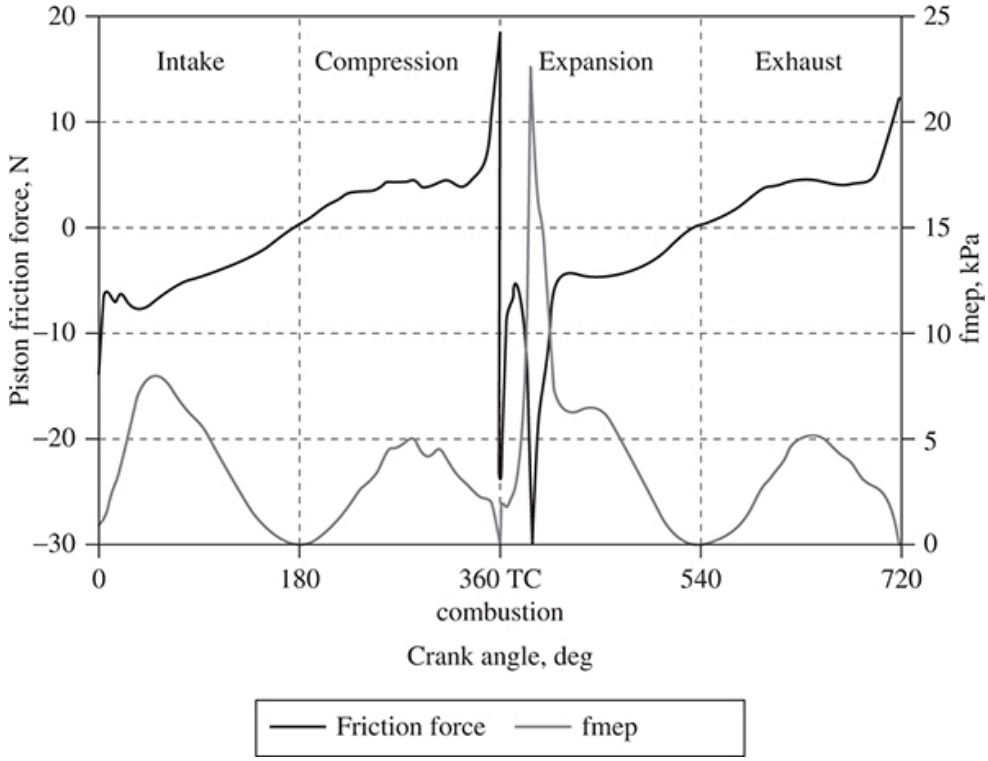


Figure 13.23 Variation in piston friction force, and resulting fmep, through the operating cycle of a gasoline spark-ignition engine. Lightweight low-friction piston design: 400- kPa bmep, 2000 rev/min. [19]

13.6.4 Crankshaft Friction

Crankshaft friction contributions come from journal bearings (connecting-rod big end, main and accessory or balance shaft) bearings, and their associated seals. A schematic of a journal bearing operating under hydrodynamic lubrication is shown in Fig. 13.24. Large loads can be carried by journal bearings with low energy losses under normal operating conditions due to the complete separation of the two surfaces in relative motion by the lubricant film. Loads on crankshaft journal bearings vary in magnitude and direction because they result primarily from the inertial loads of the piston/connecting rod mechanism and the cylinder gas loads [see Eq. (13.15)]. Typical loads and the resulting journal eccentricity diagram for a connecting rod big-end bearing are shown in Fig. 13.25. From the journal eccentricity diagram, the minimum oil film thickness can be determined. This quantity, the minimum separation distance between the journal and bearing surfaces, is a critical

bearing design parameter. If the film thickness is too low, asperities will break through the oil film and substantially increase the friction and wear. Journal bearings are usually designed to provide minimum film thicknesses of about 2 μm.

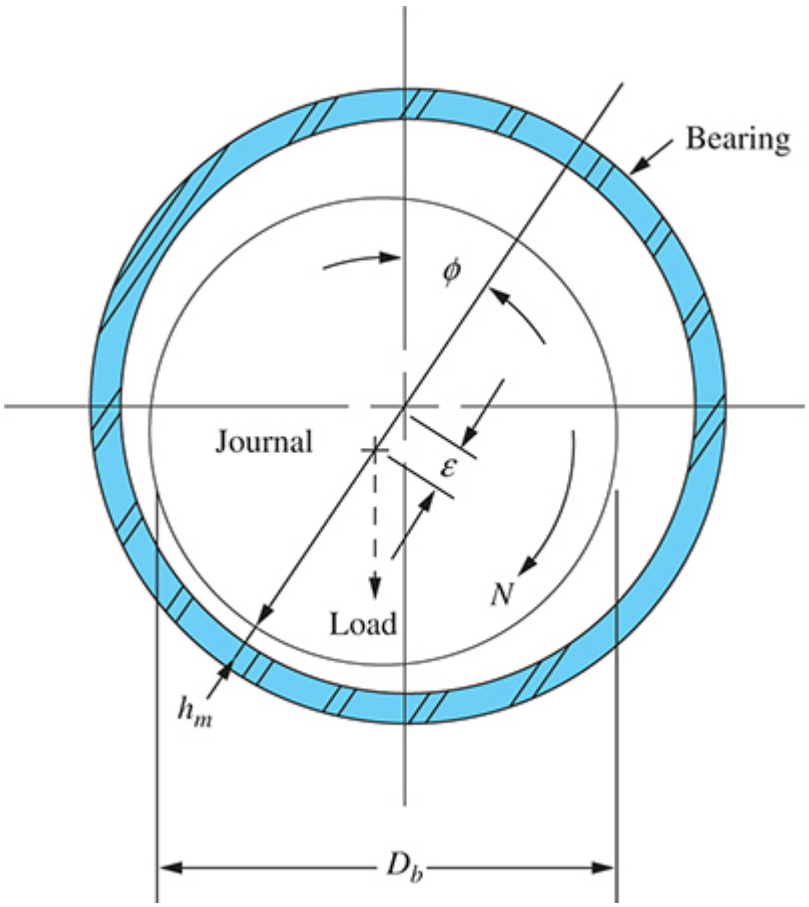


Figure 13.24 Schematic of hydrodynamically lubricated journal bearing.[3]

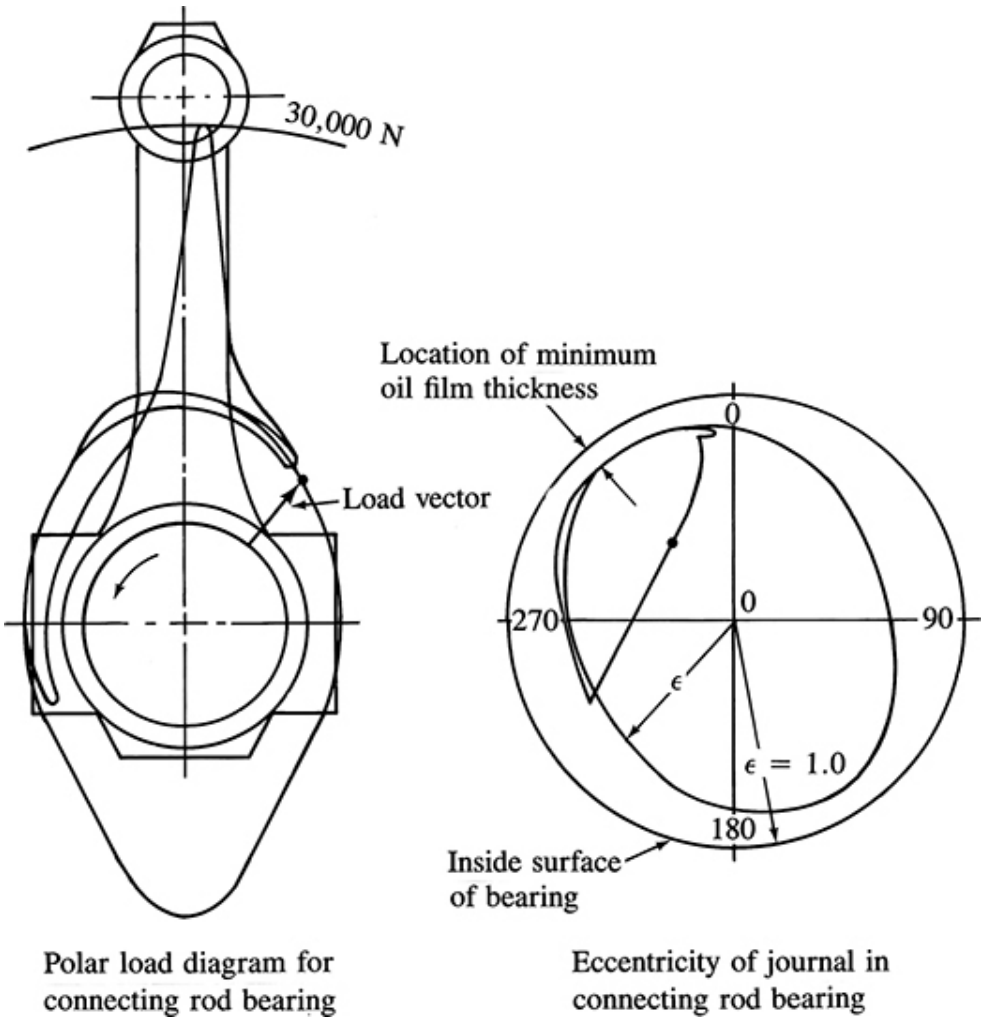


Figure 13.25 Typical engine journal bearing load and eccentricity diagrams.[3]

The friction force F_f in a bearing is given approximately by the product of the bearing area, the oil viscosity, and the mean velocity gradient in the oil:

$$F_f \approx (\pi D_b L_b)\mu\left(\frac{\pi D_b N}{\bar{h}}\right) = \frac{\pi^2 \mu D_b^2 L_b N}{\bar{h}} \tag{13.11}$$

where D_b and L_b are the bearing diameter and length, $\bar{h}$ is the mean radial clearance, and N is the shaft rotational speed. A more sophisticated analysis of the friction in a hydrodynamically lubricated bearing yields the relation [20]

$$F_f = \frac{\pi^2 \mu D_b^2 L_b N}{(1-\varepsilon^2)^{1/2} \bar{h}} + \frac{\bar{h} \varepsilon W}{D_b} \sin\phi \tag{13.12}$$

where e is the eccentricity ratio ($\bar{h} - h_m$)/ $\bar{h}$ and h_m is the minimum clearance. The first term closely matches the approximation given in Eq. (13.11) . The factor 1/(1 - ε [2]) [1/2] and the second term correct for the offset of the journal center relative to the bearing center: W is the bearing load and ϕ the attitude angle. To first order, with hydrodynamic lubrication, the friction power does not depend significantly on the bearing load. If σ is the loading per unit projected area of the bearing [W/(L_b D_b)] then the coefficient of friction/is given by

$$f = \frac{F_f}{W} \approx \frac{\pi^2 \mu D_b^2 L_b N}{\sigma L_b D_b \bar{h}} = \frac{\pi^2 D_b}{\bar{h}} \frac{\mu N}{\sigma} \tag{13.13}$$

For a given bearing, or series of geometrically similar bearings, the friction coefficient is proportional to μ_N/ σ. However, at low values of μ_N/ σ the hydrodynamic pressure in an actual bearing will be insufficient to support the shaft load and the oil film becomes incomplete. The friction coefficient increases rapidly as mixed lubrication then occurs.

Figure 13.26 shows two examples of crankshaft friction measurements, for a 3.0-liter and 5.4-liter naturally-aspirated gasoline engines.[8] The two engines show comparable crankshaft friction mep values which rise steadily with increasing engine speed. Note that the front and rear main bearing seals also contribute to crankshaft assembly friction. At 1500 rev/min, they are responsible for about 20% of the friction attributable to the crankshaft. Hoag [12] provides a useful summary of bearing and crankshaft design requirements.

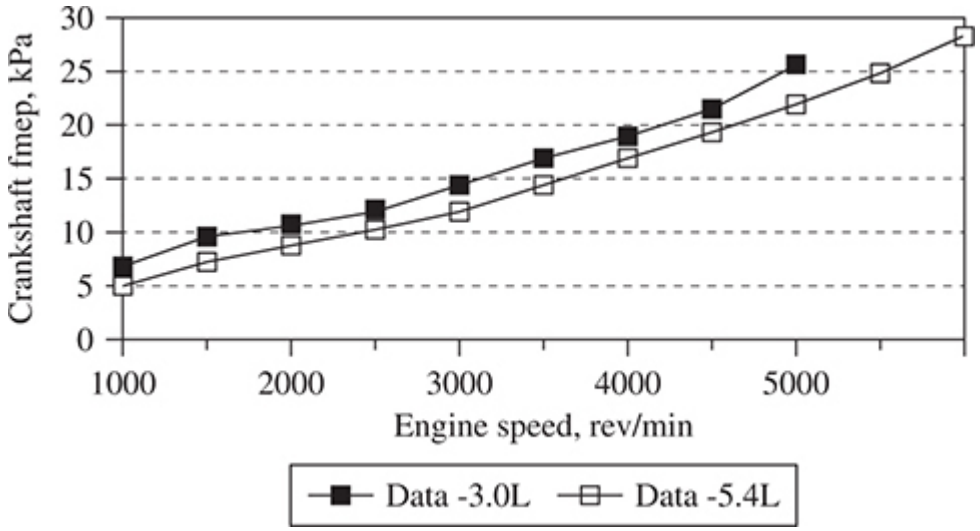


Figure 13.26 Crankshaft friction mean effective pressure data from a 3.0-liter V-6 and a 5.4-liter V-8 naturally-aspirated gasoline spark-ignition engine; from motored engine teardown tests.[8]

13.6.5 Valvetrain Friction

The valve train carries high loads over the entire speed range of the engine. Loads acting on the valve train at lower speeds are due primarily to the spring forces, while at higher speeds the inertia forces of the component masses dominate. Valve train designs can be classified by type of configuration. Examples of common valvetrain configurations are shown in Fig. 13.27. The objectives are to provide a compact, sufficiently "stiff," low mass, valve opening and closing mechanism at low cost. (Valve airflow and exhaust flow characteristics are discussed in Sec. 6.3 .) The arrangements shown in Fig. 13.27 *a* have overhead cam drives; those in (*b*) have cam-driven push-rod arrangements. The latter are the traditional valvetrain design for larger engines, where the camshaft can be positioned in the "vee" of the engine block: they have lower cost but higher mass and are less stiff. Overhead cam systems act more directly on the valves, have lower mass, and are preferred in smaller higher-speed engines. The direct-acting system shown is the simplest configuration, but since the cam lobe must provide the full valve lift, requires a larger cam and camshaft. Center- and side-pivot systems provide valve lift greater than the cam lobe height (by some 1.2 to 1.5 times). The direct-acting system thus increases engine height the most.

Valvetrain friction can be broken down by critical contact regions: camshaft journal bearings, rocker-arm fulcrum, and cam/tappet interface. A flat tappet in sliding contact with a cam mostly experiences hydrodynamic lubrication. The contact between rocker arm and valve stem, and in the rocker arm fulcrum/pivot, experiences mixed-film lubrication. The use of roller cam followers (shown in the center pivot and roller-follower push rod drawings in Fig. 13.27) allows the load at this interface to be about twice that with a flat sliding tappet. If hydrodynamic lubrication between the roller and cam surface is maintained, use of roller followers significantly reduces valvetrain friction—especially at lower engine speeds. [12]

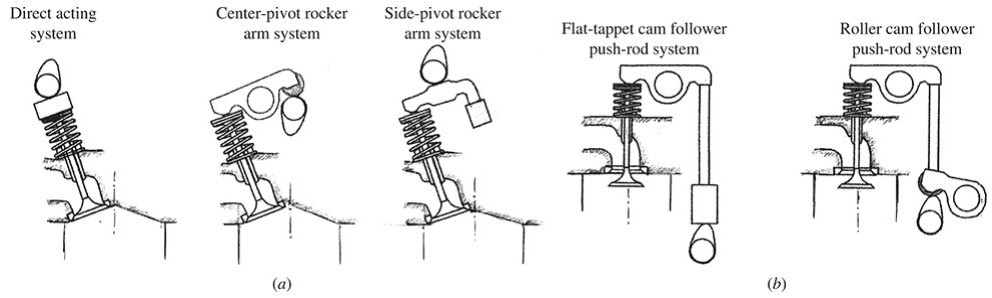


Figure 13.27 Different valve train arrangements: (*a*) overhead cam configurations; (*b*) push-rod configurations: roller cam-follower also illustrated.[12]

As the valves are opened and closed, the inertia force (effective mass × valve acceleration) as well as the valvetrain friction, determines the required driving force. Valve lift, velocity, and acceleration profiles through an engine operating cycle are shown in Fig. 13.28. Typical exhaust and intake valve lift profiles are shown on the left of the figure. The valve velocity profile, L_v', is obtained by differentiating the valve lift with respect to time. A second time differentiation yields the valve acceleration, L_v''. As the valves lift off (and later) close onto the seat, the velocity increases (or decreases) slowly to minimize the acceleration from impact to rest. The velocity then increases rapidly (on opening) to reach a large valve open area quickly. At maximum lift, the valve comes to rest, and the process reverses. Maximum accelerations occur as the valve velocity increases (later, decreases) rapidly as the valve reaches about half its maximum lift.

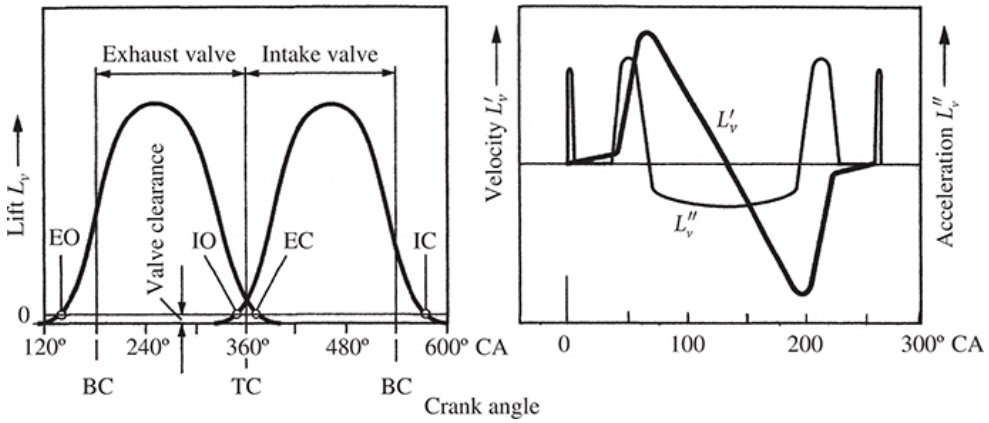


Figure 13.28 Left: Exhaust and intake valve lift L_v profiles versus crank angle; opening and closing angles (EO, IO, EC, and IC) are defined at a modest lift about equal to the slack in the valvetrain system. Right figure shows valve velocity L'_v and acceleration L''_v during the valve open period. [13]

Friction differences between these systems are difficult to quantify. Generally, measurements of valvetrain friction for each of these valve train types show significant variations, especially at mid- and high-engine speeds.

Figure 13.29 shows two examples of valvetrain friction mean effective pressure for naturally-aspirated gasoline engine. The 5.4-liter V-8 engine had a two valve per cylinder, single overhead cam, with direct acting valve mechanism. The 3.0-liter V-6 engine had four valves per cylinder, double overhead cam with roller finger followers. The fmep values are significantly different. Older engines typically showed valvetrain fmep *decreasing* with increasing engine speed, believed due to the occurrence of boundary friction at low speed and the extent of that boundary friction period decreasing as speed increases. Significant reductions in valvetrain friction have been realized, and the valvetrain fmep is now almost independent of speed. The amount of boundary friction occurring has been substantially decreased. [8]

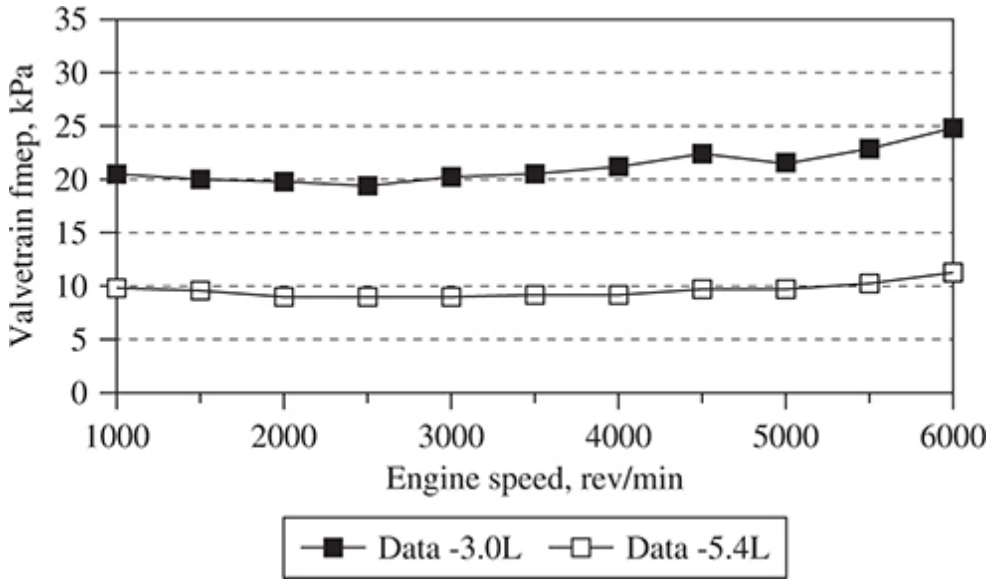


Figure 13.29 Valvetrain friction mean effective pressure (motored engine teardown tests) of two naturally-aspirated gasoline engines: a 3.0-liter (four-valve) V-6 and a 5.4-liter (two-valve) V-8.[8]

Effective methods of decreasing valve train friction are: (1) spring load and valve mass reduction; (2) use of tappet roller cam followers; (3) use of rocker arm fulcrum needle bearings. Studies of low-friction valve train designs indicate that roller cam-followers provide the largest benefit: reductions of order 50% in valve train friction can be achieved. Reducing valve, spring, and tappet mass also reduces valvetrain friction: for example, reducing this mass by 30% reduced valvetrain friction torque by 40% at speeds of 2000 rev/min and above. [23]

13.7 PUMPING FRICTION

Pumping air into the engine during the intake process, and burned gases out during the exhaust process, during the gas exchange process (in the four-stroke engine cycle), requires a work transfer between the in-cylinder gases and the piston. Several flow restrictions outside the cylinder head, in the inlet and exhaust systems—air filter, ducting, throttle, intake manifold (on the inlet side), exhaust manifold and tail pipe, catalytic converter, and muffler (on the exhaust side)—cause pressure drops which require work to

overcome. Equally important are pressure losses in the inlet and exhaust ports and valves. As load is reduced in a spark-ignition engine, the airflow is reduced by partly closing the throttle increasing its flow restriction. The increase in throttling work is more rapid than the decrease in flow work. Both throttling work and flow work increase as engine speed increases at constant load.

The above summary applies to both naturally-aspirated engines and turbocharged engines, and gasoline and diesel engines (though the latter do not normally embody a throttle). However, incorporating a turbocharger significantly changes the intake and exhaust pressures at mid and higher loads. Section 13.2 explains the different ways that pumping work is defined in naturally-aspirated and turbocharged engines. The primary reason for this difference is that in naturally-aspirated engines, the pumping work is always negative—a work loss—so is counted as an engine friction component. In turbocharged engines at high outputs, pumping work is positive and adds to the engine's output. It is not then counted as an engine friction component. But at light loads, when the impact of the turbocharger is modest, pumping work becomes a loss. Thus in assessing pumping work as a loss or gain, these three factors—the innate flow resistances, the extent the engine is throttled, and engine boosting via a turbocharger—must be considered.

The classical definition of pumping work, and pumping mean effective pressure, is given in Sec. 2.5 (Fig. 2.6) and Sec. 5.3 [Eqs. (5.22) and (5.23)]. It is the $\int pdV$ over the expansion and exhaust strokes, which in simple terms is

$$\text{pmep} = \frac{W_p}{V_d} = \overline{p}_e - \overline{p}_i \tag{13.14}$$

where $\overline{p}_e$ is the average in-cylinder pressure during the exhaust stroke, and $\overline{p}_i$ the average in-cylinder pressure during the intake stroke. In naturally-aspirated (and throttled turbocharged) engines, $\overline{p}_i$ is less than $\overline{p}_e$ and W_p and pmep constitute a loss. When $\overline{p}_i$ is greater than $\overline{p}_e$ (i.e., when the engine is boosted), the pumping work is a gain. In a highly throttled naturally-aspirated gasoline engine, $\overline{p}_e - \overline{p}_i$ might be 0.6 bar when the brake mep is 2 bar. At high loads and lower speeds, $\overline{p}_e - \overline{p}_i$ would be 0.2 to 0.3 bar and bmep 10 bar. In a turbocharged engine at high load, $\overline{p}_i - \overline{p}_e$ could be up to about 1 bar with a bmep of some 20 bar. With turbocharged diesel engines under boost

conditions, $\overline{p}_i$ is usually higher than with SI engines, but so also is $\overline{p}_e$ due to the lower diesel exhaust gas temperatures and energy.

Pumping losses have been correlated in various studies with variables such as cylinder-head valve geometries, engine mass flow, and engine speed (usually squared). The pumping mep equation below comes from an overall engine friction model for naturally-aspirated gasoline engines which adds together sub-models for each friction component, each based on fundamental scaling laws and empirical data: [8]

$$\text{pmep (kPa)} = (p_a - p_i) + 3.0\times10^{-3}\left(\frac{p_i}{p_a}\right)^2\left(\frac{\overline{S}_p^2}{n_v^2 r_i^4}\right) + 0.178\left(\overline{S}_p^2\frac{p_i}{p_a}\right) + 3.0\times10^{-3}\left(\frac{p_i}{p_a}\right)^2\left(\frac{\overline{S}_p^2}{n_v^2 r_e^4}\right) \quad (13.15)$$

Here, p_a is atmospheric pressure, p_i the intake manifold pressure, $\overline{S}_p$ the mean piston (m/s) speed, n_v number of valves per cylinder, r_i the intake valve diameter/bore, r_e the exhaust valve diameter/bore; all pressures are in kPa. The first term represents the intake system pressure losses (including any throttling), the second term is the intake port and valve losses, the third term the exhaust system losses, and the fourth term the exhaust valve and port losses. [8]

Figure 13.30 shows pumping mep data and model [Eq. (13.15)] results for a naturally-aspirated 6.4-liter V-10 gasoline engine, as a function of engine speed and load (manifold air pressure, MAP). At typical light-load engine operation (MAP = 0.5 bar, 50 kPa), the pmep is about 0.5 bar and modestly dependent on engine speed. Even at high MAP values (above 0.8 bar) pmep is not negligible, and increases significantly with increasing engine speed.

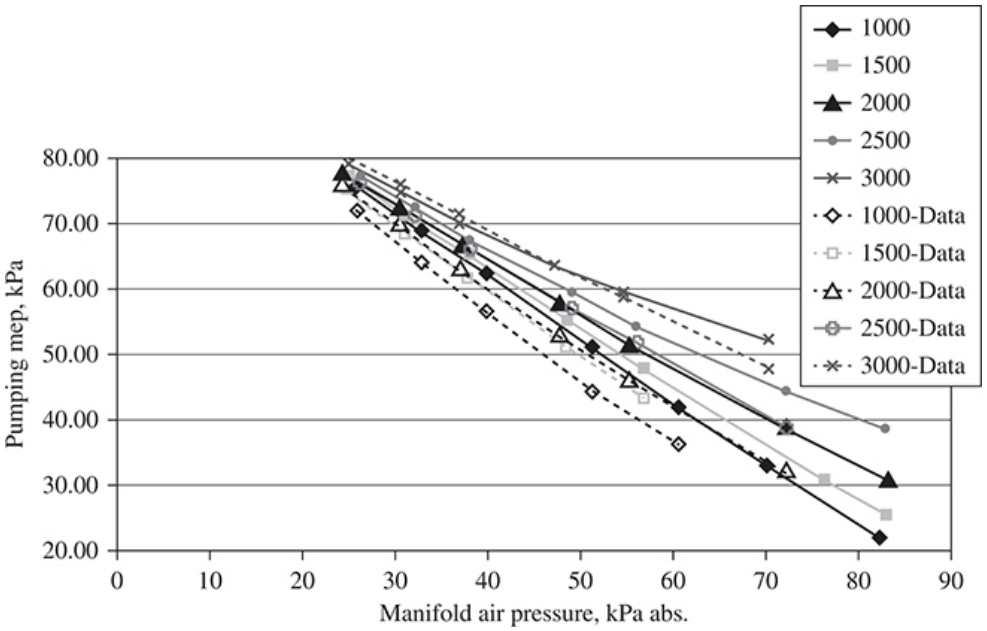


Figure 13.30 Pumping mean effective pressure (pmep) as a function of (absolute) manifold air pressure (MAP): engine speeds, 1000–3000 rev/min. 6.4-liter V-10 NA gasoline engine. Data, open symbols and dashed lines; model results, Eq. (13.15) , fitted symbols, and solid lines.[8]

The advent of variable valve control has led to the introduction of a modified approach to pumping and indicated work. The classical approach discussed above neglects valve-timing impacts that occur during the expansion and compression strokes. With variable valve timing, the valve events can have a significant effect on all four strokes of the engine's operating cycle. As a consequence, changes in the net indicated performance cannot readily be separated into energy conversion efficiency effects and pumping work changes. The following approach calculates the work associated with the gas exchange process more completely by making adjustments to the pumping mep and gross indicated mep classical definitions. [24]

For an engine with fixed valve timing, the exhaust valve opening angle (EVO) is a trade-off between high-speed pumping work and low-speed expansion work. [25] This results in some loss of expansion stroke work at lower speeds as indicated on the p-V diagram (Fig. 13.31*a*, upper right). The amount of expansion work lost can be calculated by extrapolating the expansion stroke prior to EVO using the pV^n = constant relationship and

integrating the area between that extrapolation and the measured in-cylinder pressure, as shown.

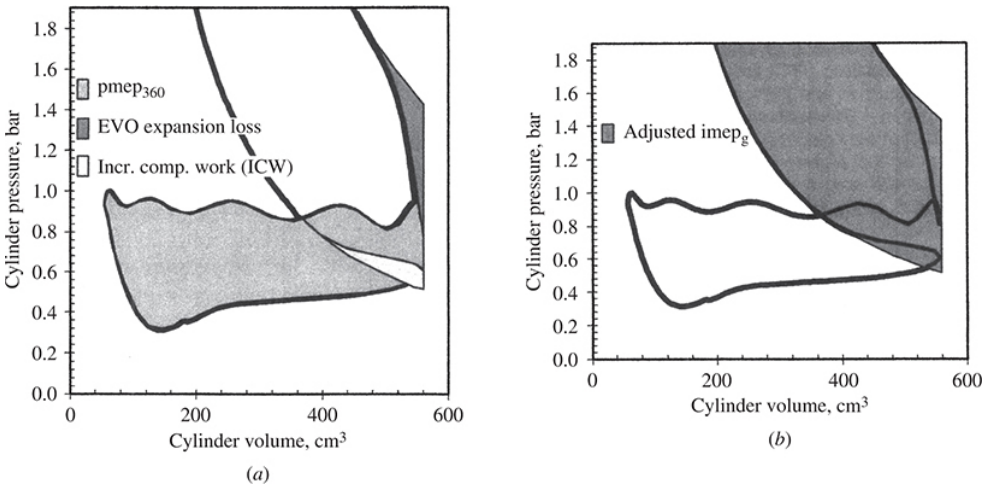


Figure 13.31 Cylinder pressure vs. cylinder volume diagram from EVO into the compression stroke: (*a*) Shaded areas represent all of the gas exchange work, defined as the adjusted pumping work; (*b*) shaded area represents the gross indicated work per cycle (Sec. 2.5) adjusted for valve timing effects. Engine data at 2.6-bar bmep, 1500 rev/min. [24]

With fixed intake cam timing, the intake valve is typically closed about 50° ABC of the compression stroke. Thus, as also shown in Fig. 13.27 *a*, the compression stroke work is slightly higher due to the actual in-cylinder gas pressure being higher than the extrapolated pV^n = constant compression stroke line from IVC back to BC. The *adjusted* definitions for imep and pmep thus become:

$$\text{imep(g)}_{\text{adj}} = \text{imep(g)} + \text{ICW} + \text{EVO}_{\text{exp loss}} \tag{13.16}$$

$$\text{pmep}_{\text{adj}} = \text{pmep}_{360} + \text{ICW} + \text{EVO}_{\text{exp loss}} \tag{13.17}$$

Thus, the net imep (which is the 720° full cycle $\int pdV$) becomes

$$\text{imep(n)} = \text{imep(g)}_{\text{adj}} - \text{pmep}_{\text{adj}} \tag{13.18}$$

which is an equivalent relationship to the classical approach equation

$$\text{imep(n)} = \text{imep(g)} - \text{pmep}_{360} \tag{13.19}$$

The gross and net imep definitions are the standard ones based on gross and net indicated work per cycle (Sec. 2.5). ICW is *increased compression work*, EVO $_{\text{exp loss}}$ is the expansion work lost due to early exhaust valve opening (both converted to mep by dividing by V_d), as described above: pmep $_{360}$ is the standard exhaust plus intake stroke (360°) pumping work definition.

Note that Eqs. (13.18) and (13.19) give identical values for net imep (as they should). However, with the adjustments to imep and pmep summarized here, the valve timing effects on the gas exchange process work/losses, and the effects on the in-cylinder fuel chemical energy to gas pressure energy to work transfer to-the-piston sequence, which determine the engine's inherent energy conversion effectiveness, have been more completely separated.

Figure 13.32 shows a load sweep for a standard naturally-aspirated gasoline engine, which illustrates the roles of this adjusted pmep approach (and that of mechanical friction mep) in the engine's imep and bmep at 1500 rev/min. Figure 13.32 *a* shows how the adjusted pmep reduces the engine's imep(g), especially in relative terms at lower loads. It then shows the additional impact of mechanical friction mep, which increases with load due to higher component force levels due to higher cylinder pressures, decreasing imep(n) to the (usable) bmep. Figure 13.32 *b* shows how that the adjusted pmep becomes steadily higher than pmep$_{360}$ as load increases. At high load (in this NA engine) the real pumping mean effective pressure, even at this low speed, is significant. [24]

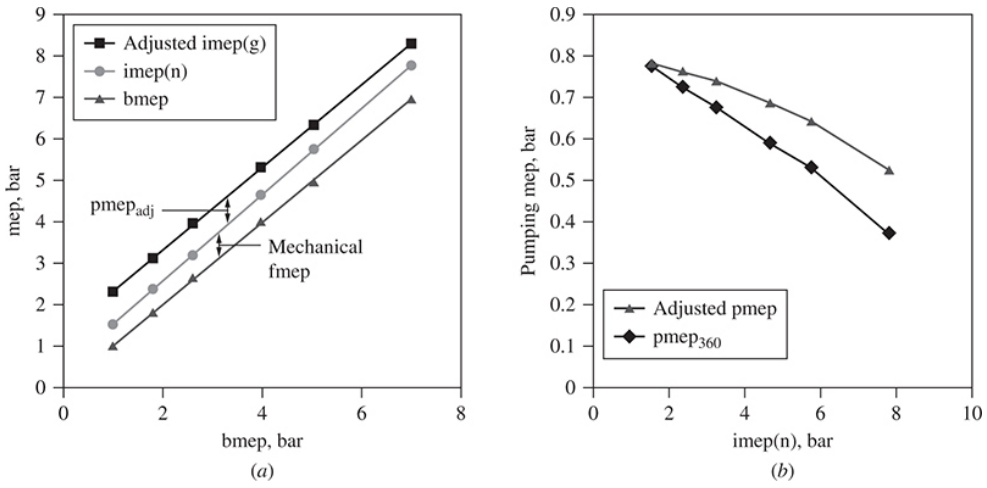


Figure 13.32 (*a*) Mean effective pressure parameters for a bmep sweep at 1500 rev/min; (*b*) comparison of pumping work mep parameters for a bmep sweep, plotted as imep(n), at 1500 rev/min.[24]

Variable valve control (see Sec. 6.3.3) allows the pumping work to be reduced. This is especially important at lighter loads and lower engine speeds, engine conditions important in automobile applications. Under these conditions, the objective is to increase $\overline{p}_i$ so the pumping mep ($\overline{p}_e - \overline{p}_i$) is reduced. At IVC, under these quasi-static conditions,

$$p_{\mathrm{IVC}} V_{\mathrm{IVC}} = mRT_{\mathrm{IVC}} \tag{13.20}$$

A specific mass of air and fuel is needed to generate the required torque; thus m is essentially constant, [b] as also are R and T_{IVC} (in kelvins). Equation (13.20) thus indicates that closing the intake valve when the cylinder volume is smaller requires a higher intake manifold pressure.

The potential for pumping work reduction is illustrated in Fig. 13.33. Reduced V_{IVC} in Eq. (13.20) can be achieved by either closing the IV part way through the *intake* stroke; or closing it later in the compression stroke. These two strategies are shown on the *p-V* diagrams that result, Fig. 13.33 *a* and *b*, respectively. The engine tested used a *camless valvetrain*—an electrical valve actuation (EVA) mechanism—to provide extensive valve control flexibility. Early valve closing (during the intake stroke) reduces the compression stroke work through gas expansion during the second half of the intake stroke followed by re-compression. The fixed-valve-timing engine pumping work is greatly reduced. With late IVC, more than halfway through the compression stoke, the pumping work is essentially eliminated, though the compression stroke work is increased. [24]

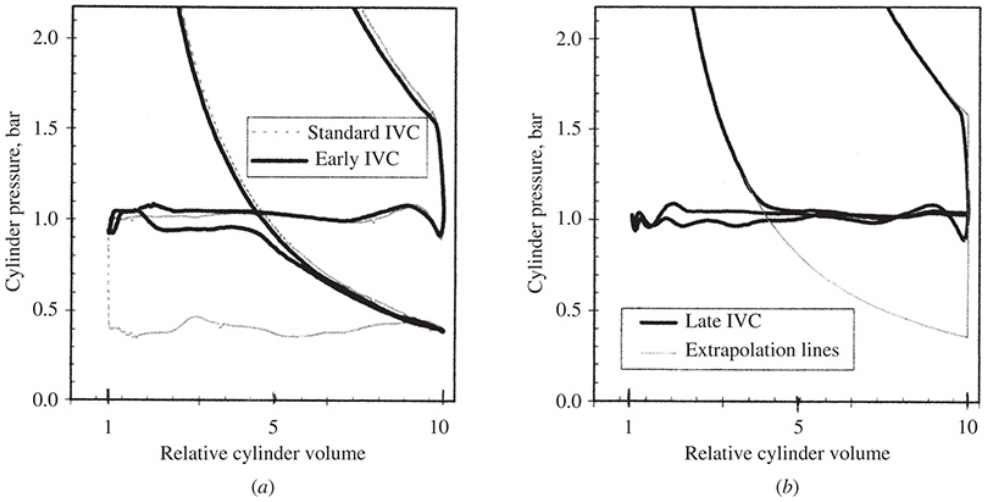


Figure 13.33 Cylinder pressure vs. cylinder volume diagram from EVO into the compression stroke: (*a*) early intake valve closing (IVC) compared to standard IVC; (*b*) late IVC, with compression line extrapolated to BC. Engine data at 2.6-bar bmep, 1500 rev/min. [24]

An alternative approach to reducing pumping used in some production gasoline engines is to reduce the number of firing cylinders (often called cylinder deactivation): for example, a V-8 engine firing only four cylinders; a V-6 firing only three or two cylinders. This is a part-load fuel consumption improvement strategy. Since, with reduced number of cylinders firing, the engine must still deliver the required torque, each firing cylinder must operate at higher bmep (half the cylinders firing, roughly twice the bmep). Thus the firing cylinders operate at significantly higher intake manifold pressure, and pumping work is reduced as is bsfc. Note that the mechanical and accessary friction of all the cylinders must be offset with part of the indicated work even when half (or one-third) of the cylinders are firing. Cylinder deactivation is implemented by decoupling the valve actuation mechanism of the cylinders that are not firing. In these cylinders, the mass of gas trapped is compressed and expanded, cycle after cycle. [26]

Finally, one of the benefits of turbocharging (*and* downsizing engines because the output for a given engine displacement goes up) is that pumping work is reduced. In automobile applications, typical vehicle driving is centered at a torque (or bmep) level that is about one-quarter of the maximum torque/bmep for that engine. Turbocharging increases the maximum bmep by some 50 to 100%. This typical part load bmep thus increases to 5 bar or

more from a bmep of 2 to 3 bar in the corresponding naturally-aspirated engine. Pumping mep is correspondingly reduced.

The above discussion of pumping work (as well as of the other components of friction) emphasizes the two complementary paths available for reducing the impact of engine friction: (1) reducing the *absolute magnitude* of engine friction components and (2) reducing the *relative importance* of friction. This latter path is a major reason why turbocharging an engine not only increases its output per unit engine displacement, it also improves the engine's mechanical efficiency η_m [Eq. (2.24)], which quantifies the relative importance of engine friction.

13.8 ACCESSORY POWER REQUIREMENTS

The coolant/water pump, oil pump, and fuel pump are built-in accessories, essential to engine operation, and are normally considered part of the base engine. A fully-equipped engine usually includes additional accessories—for example, a fan and generator. In automobile use, it often includes a power-steering pump, an air conditioner, and even an air pump for emission control. These have been driven off the engine crankshaft, though some of these accessories may now be electrically driven to decouple their performance (and thus power dissipated) from scaling with engine speed. Friction belt drives (e.g., serpentine drives with ribbed V-belts and a tensioning pulley) are commonly used to drive the camshaft, water pump, alternator, power-steering pump, and air-conditioning compressor. Chain drives may be used to drive camshafts, oil pumps, and fuel pumps in diesel engines where (due to the high fuel-injection pressure) the fuel pump power requirements are significant, [27] since they provide more robust and maintenance free operation of these critical accessories.

The power requirements for the accessories fan, generator, and power-steering pump, are not normally included in the term "accessory friction" Of these, the fan requirement is the largest, and with a direct drive increases with the cube of the speed. Alternative couplings such as a viscous drive reduce the fan speed at high engine speed and thereby reduce its power significantly. The power-steering pump is only required to provide high pressures intermittently. With hydraulic power-assisted systems, only the

fluid pumping losses are charged against the engine. Electric power-assisted steering systems are also used: these are driven by an electric motor powered by the electrical system. [13]

The power delivered by the fully equipped engine (the net power) is lower than the power delivered by the base engine due to the power requirements of these accessories. The friction mean effective pressures associated with driving the water pump, alternator, oil pump, and fuel pump (diesels) together comprise up to about 20% of total engine friction. The water pump fmep is typically less than about 7 kPa at 1500 rev/min; the oil pump 4 to 10 kPa at this speed; the alternator requires 7 to 10 kPa. These numbers vary significantly with component design details. The generator power depends on the electrical load to be met and the generator blower design. Average generator power is about two-thirds of the peak power.

Air-conditioning is standard on a majority of U.S. cars; additional power is required for the air-conditioning compressor. Also, since the compressed refrigerant is condensed in a second radiator, a larger-than-standard fan is required to pull additional air through the combined radiator systems. An air pump which pumps air into the engine exhaust ports may be part of an SI engine emission control system (see Sec. 11.6). Its power requirements (~1 kW at normal engine speeds) must then be added to the accessory friction requirements.

The accessory friction mep as a function of speed for a 3.0-liter and a 5.4-liter naturally-aspirated gasoline engine is shown in Fig. 13.34. Both motored engine data and predictions from Eq. (13.21) , below, are shown, and match well:

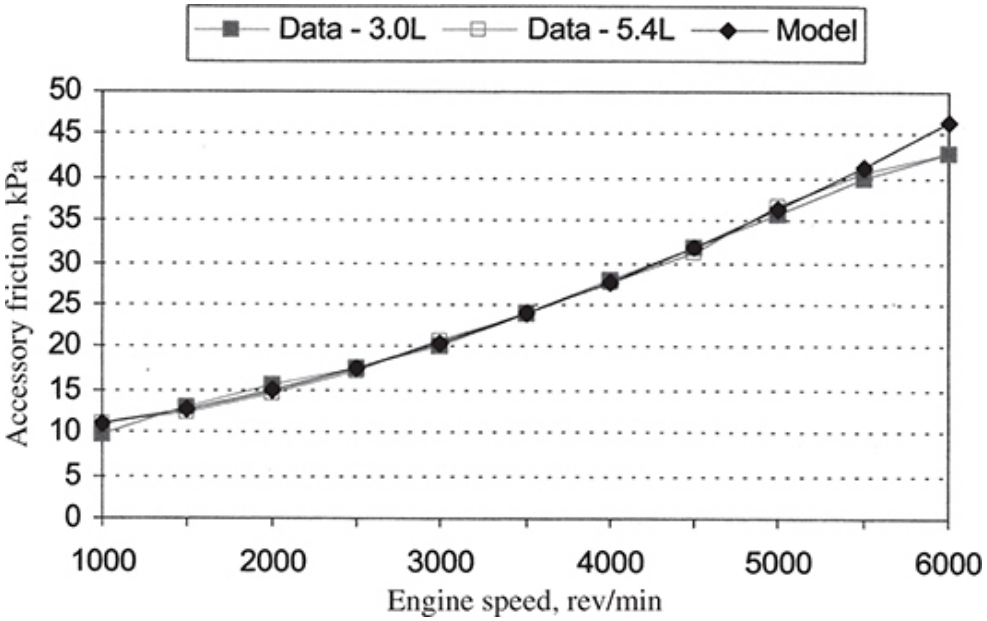


Figure 13.34 Accessory friction mean effective pressure (motored engine teardown data) for 3.0-liter V-6 and 5.4-liter V-8 naturally-aspirated gasoline engines as a function of speed. Model is Eq. (13.21) .[8]

$$\text{afmep (kPa)} = 8.3 + 1.86\left(\frac{N}{1000}\right) + 0.75\left(\frac{N}{1000}\right)^2 \qquad (13.21)$$

where N is engine speed in rev/min. Accessory friction mep here includes water pump, oil pump, and (unloaded) alternator. For diesels, the fuel pump drive requirements become significant. Note that accessory drive requirements are being reduced over time. [8]

13.9 ENGINE FRICTION MODELING

As this chapter has made clear, engine friction is a multi-component process with substantial and different details important in each friction component. Detailed models for these components, and models which combine these components, have been and continue to be developed. Integrated friction models are needed to convert from indicated engine performance and efficiency to brake values. They are a necessary component in widely used engine operating cycle simulations which though their flow, combustion, and

heat transfer models essentially predict *indicated* engine operation. Detailed component models (e.g., for the piston ring pack) develop our understanding of critical friction and lubrication phenomena and identify from a fundamental perspective opportunities for improvement. Note that engine friction is steadily being reduced so friction models, which usually require some calibration against measured data, need to be periodically updated.

Integrated models are usually based on engine component friction data obtained from motored engine teardown tests (Sec. 13.4). Models for each major friction component (piston assembly, crankshaft, valvetrain, engine accessories, pumping work) are developed, sometimes at the subcomponent level, using terms based on the scaling laws for hydrodynamic and mixed (combined boundary and hydrodynamic modes) lubrication, and turbulent dissipation (see, for example, Patton et al., [28] expanded and updated by Sandoval and Heywood [8]). These scaling law terms are then calibrated against engine teardown test data. Thus, total friction mean effective pressure is given by

$$\text{tfmep} = \sum_i (\text{component fmep})_i \tag{13.22}$$

where each component fmep term is the sum of sub-component terms with (as appropriate) hydrodynamic and mixed lubrication terms. These component models involve variables such as: lubricant viscosity (and its scaling with oil temperature); cylinder bore, stroke, compression ratio, number of cylinders; number of valves per cylinder, valve lift, valve diameter; engine speed, mean piston speed; ring tension; intake and exhaust manifold pressures relative to atmospheric; valvetrain constants for the different valve drive mechanisms; with, of course, several calibration constants. [c]

An important question is the effect of oil, coolant, and liner temperature on engine mechanical friction. The tribology literature shows that the oil viscosity scaling in the hydrodynamic friction terms should have the form: [29]

$$\mu_{\text{scaling}} = \sqrt{\frac{\mu(T)}{\mu_0(T_0)}} \tag{13.23}$$

where $\mu(T)$ is the viscosity of the oil in the engine for which friction predictions are being made, and $\mu_0(T_0)$ is the reference viscosity of the oil used in the engines that provided the data to calibrate the model. Viscosity values for several different oil grades can be obtained from Eq. (13.25) below, and Table 13.3 (in Sec. 13.11). Since the density of the oil is essentially constant, the dynamic viscosity μ is assumed to scale with the kinematic viscosity ν. Thus

$$\mu_{\text{scaling}} = \sqrt{\frac{\nu(T)}{\nu_0(T_0)}} \tag{13.24}$$

where ν_0 is the kinematic viscosity at reference temperature T_0.

The low shear rate viscosity ν_0 of the oil (in centistokes) is related to temperature via the Vogel equation:

$$\nu_0 = k\exp\left(\frac{\theta_1}{\theta_2 + T}\right) \tag{13.25}$$

k is a correlation constant (in cSt), as are θ_1 and θ_2 (°C) for an oil, and T is the oil temperature (°C). This low-shear viscosity [Eq. (13.25)] is then converted to the high-shear-rate viscosity by multiplying by the ratio $\mu\infty/\mu_0$ (a standard ratio, of magnitude one or less, for a given oil) because most lubricated engine components operate in the high shear regime where multi-grade oils exhibit "shear thinning." (See Sec. 13.11 , Table 13.3, for specific numbers.) Using Eqs. (13.23) , (13.24), and (13.25), the effects of lubricant temperature on mechanical friction can be modeled.[8]

Figure 13.35 *a* shows the effect of oil temperature (assumed to scale with the coolant temperature) on the components of mechanical friction for a 15W-40 oil for a 5.4-liter naturally-aspirated gasoline engine at 2000 rev/min. Note the substantial effect of oil temperature: friction in a just-started engine at 20°C is twice that in a warmed-up engine at 90 to 100°C coolant temperature (see Fig. 13.16 and accompanying text).

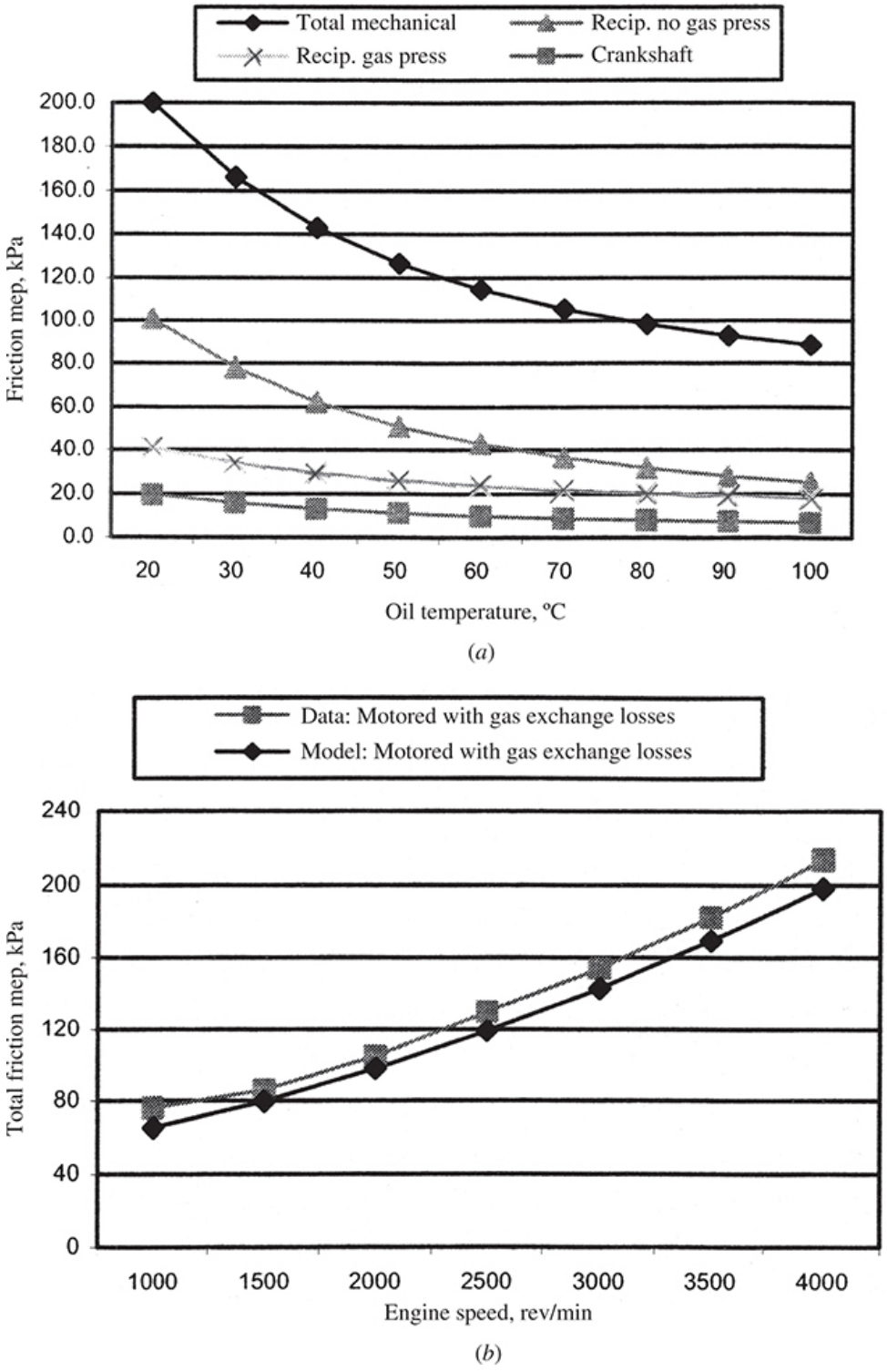


Figure 13.35 (*a*) Total mechanical friction mep as a function of lubricant (15W-40) temperature within the bearings and valvetrain, on the piston rings, and skirt (assumed to scale with coolant temperature). Also showing reciprocating fmep (without in-cylinder gas pressure, and the additional fmep

impact from gas pressure), and crankshaft fmep. 5.4-liter V-8 naturally-aspirated engine at 2000 rev/min. (*b*) Total friction mep data (motored), with gas exchange losses added, and friction model results. [8]

Figure 13.35 *b* shows comparison of the Patton, Sandoval, and Heywood integrated friction model with total engine friction under motored conditions. The magnitude of motored friction and its dependence on engine speed are modeled accurately. Earlier figures in this chapter (Figs. 13.30 and 13.34) have compared results of pumping and accessory friction components from this model with measured data. [8]

13.10 OIL CONSUMPTION

The engine's lubricating system and its functions are summarized in Sec. 13.6.2 . Here, we discuss the processes that result in engine oil consumption, a critical combined engine and lubricant problem due to its impacts on operating costs, its contribution to engine emissions, and the role of ash from the lubricant that degrades exhaust catalyst and particulate trap performance (Secs. 11.6.3 and 11.6.4).

13.10.1 Oil Consumption Context

The magnitude of engine oil consumption is typically about 2 mg per second, per liter of engine displaced volume. This oil loss is roughly one-third of 1% of the fuel used in vehicular applications. Oil consumption rates vary substantially with type of engine, engine design, and with engine operating conditions (oil consumption is generally lowest at light load and low speed). Oil is consumed under both steady-state engine operating conditions and during rapid engine load transients when the oil distribution in the piston and ring assembly at the beginning of the transient is markedly different from the steady-state oil distribution at the end of the transient. As Sec. 13.6.3 indicates, the detailed behavior of the piston rings and the piston during the engine's operating cycle is complex. Oil transport processes within this region, which depend on the behavior of the piston rings, also affect the oil consumption mechanisms. Oil consumption is thus only partly understood.

The several sources of oil consumption are shown in Fig. 13.36: [30] each schematic illustrates a different oil transport process into the cylinder that leads to oil being consumed. Liquid oil can be thrown from the piston top land region into the combustion chamber due to the inertia forces generated by the acceleration and deceleration of the piston assembly (Fig. 13.36*a*). The ring-pack gas flows are known to transport oil within the piston and ring region (see Secs. 8.8 and 13.6.3). The gas pressure in the second land region can rise above the cylinder pressure at certain points in the engine's operating cycle (see Fig. 13.20). This causes a reverse gas flow through the top ring gap (Fig. 8.33), which can carry oil, in liquid or mist form, into the combustion chamber (Fig. 13.36*b*). This oil transport process requires oil build up on the second land below the first ring gap and is more likely to occur at lower engine loads.

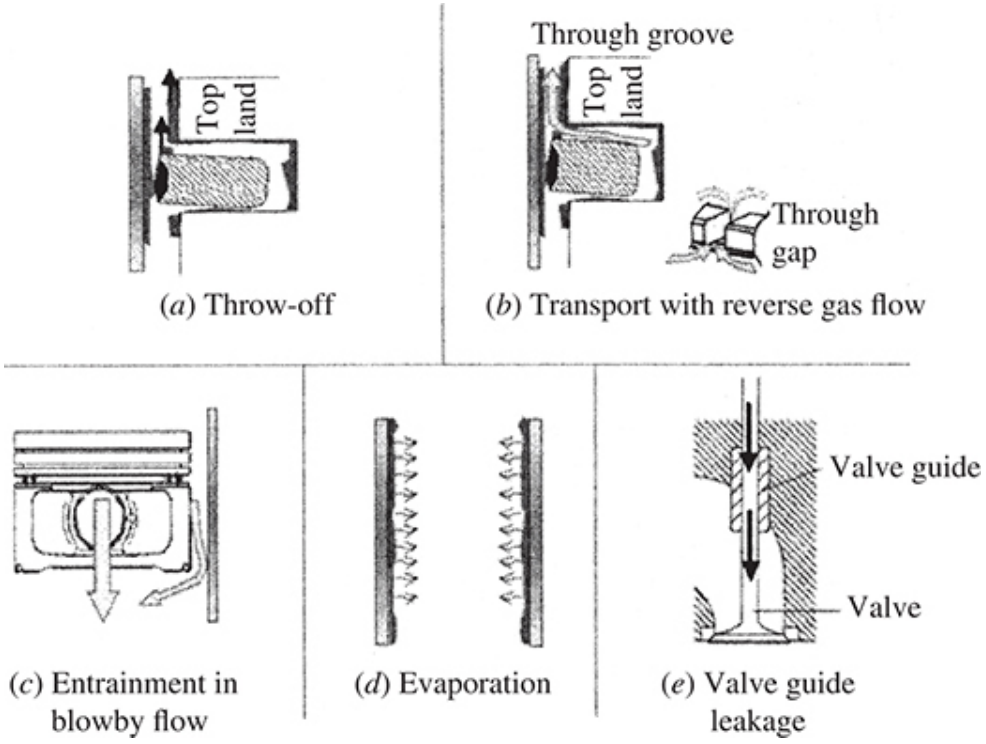


Figure 13.36 Schematic illustrating the different oil transport and consumption mechanisms: see text.[30]

The in-cylinder gases which "blow by" the piston assembly into the engine's crankcase are also a source of oil consumption. Blowby gases carry some oil back to the crankcase and oil sump. The rotating crank "splashes"

oil, and through windage, can generate oil drops and oil mist. In higher-power engines (e.g., diesels), oil jets under the piston provide additional piston cooling. The gases flowing into the crankcase are recycled to the engine's cylinders through a positive crankcase ventilation (PCV) system which incorporates an oil separator at the crankcase exit. Despite the oil separator, the recycled blowby gases contain some oil which then enters the combustion chamber via the intake manifold and burns. In turbocharged engines, turbocharger blowby—leakage of air at high boost levels from the compressor to the turbine—driven past the turbocharger bearings by the higher compressor than turbine pressure, is also an oil consumption path.

Oil evaporation, primarily from the oil film on the cylinder liner, also contributes to oil consumption (Fig. 13.36*d*). The amount of oil evaporated depends on the oil film/liner temperature: higher engine thermal loadings at higher power levels lead to higher liner temperatures and thus greater oil evaporation.

In earlier spark-ignition engine designs, oil transport from the cylinder head through the valve guide into the intake port (Fig. 13.36*e*) contributed to oil consumption during part-load operation when the intake manifold pressure is well below atmospheric. This oil leakage path is largely prevented in modern engines by employing effective valve stem seals. Valve stem leakage contributes little to oil consumption in current engine designs. [30]

The oil consumption behavior in a typical four-cylinder 1.8-liter naturally-aspirated gasoline engine is illustrated by the oil consumption map in Fig. 13.37. The oil consumption data (in µg/cycle) show a clear dependence on engine speed and load increasing as these variables increase. Oil consumption was large at high speed (>4500 rev/min) and load (>75%) at least ten times (in µg/cycle) the oil consumption at low speed and load. [30]

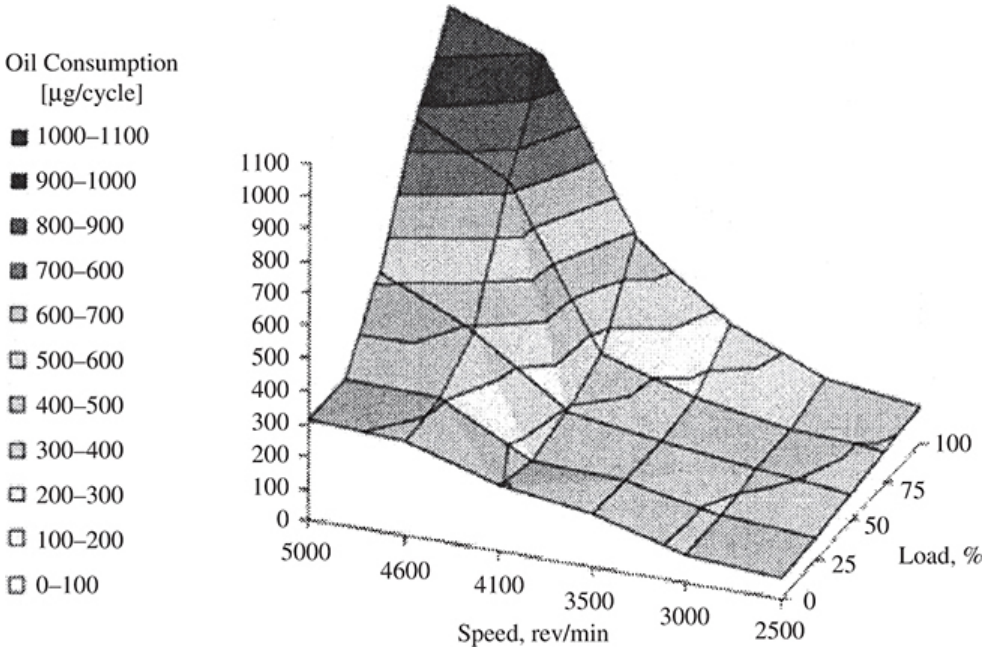


Figure 13.37 Oil consumption map for a 1.8-liter four-cylinder naturally-aspirated gasoline engine, as a function of engine speed and load (%).[30]

13.10.2 Oil Transport into the Cylinder

The question addressed here is how oil is transported from the piston skirt region below the oil control ring (where there is plenty of oil) up to the piston top-land region above the top (compression) ring. From there, inertia forces and reverse blowby gas flows could carry oil into the combustion chamber where it would burn.

Prior discussion (Secs. 8.8 and 13.6.3) has explained the behavior of the piston rings in their grooves and the gas flows into and out of these various piston land regions (Fig. 8.33). Here, we focus on the oil flows that occur in this piston ring and gas flow context. [d] Figure 13.38 shows schematically regions of the piston/ring/skirt system where oil could accumulate and from where it could be transported. The liner oil film does the primary ring and skirt lubrication task. Oil may be on the piston lands and within the ring grooves; oil is also "scraped" from the liner to the lands and vice-versa. Oil can flow through the ring gaps, up or down, directly driven by pressure differences across the ring when oil accumulates below or above the ring gap location; or as oil mist carried by the ring-gap gas flows, up or down. If the ring gaps line up, the gas flows are increased. If they are not lined up, then

circumferential oil flow on the piston lands may occur, dragged by the circumferential gas flows from one ring gap to the next. When the rings move up or down in their grooves, or flutter, then oil will be transported around the back of the ring if it has accumulated within the groove or at the groove entry.

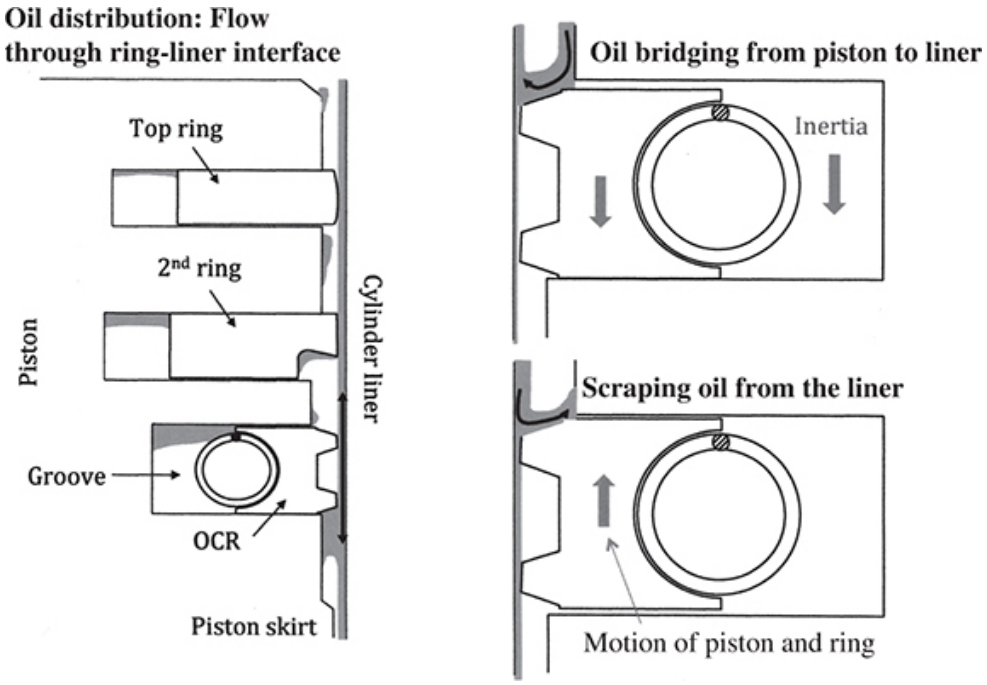


Figure 13.38 Schematics showing oil distribution in the piston, piston ring, skirt, cylinder liner, region illustrating the major oil flow paths.[31]

Cylinder bore distortion (Sec. 13.6.3) can increase the oil flow at the ring/liner interface (see Fig. 13.38*a*), depending on the magnitude of bore distortion and the extent to which the rings conform to that distortion. Larger ring-face to liner gaps will result in larger oil flows past the ring.

The several factors driving these oil transport flows in the piston and ring assembly are: pressure differences across the rings, ring-gap gas flows that entrain oil, shear forces between gas flows and oil films on the lands, inertia forces acting on the oil as the piston accelerates and decelerates.

Of greatest importance is the amount of oil on the upper part of the piston: the region above the second ring appears to be critical, and the reverse blowby top-ring gap flow is an important oil consumption contributor. Direct "throw off" of oil into the combustion chamber and evaporation of oil from any additional oil on the liner (see next section) are also major sources.

Figure 13.39 shows measured oil film thicknesses on the top and second lands as a function of speed and load, along with an example of the oil film thickness distribution along the piston (Fig. 13.39*a*). The film thickness profile indicates oil film, some 2 μm thick between the ring and liner surfaces, and the substantial accumulation of oil on the lands in between. [32]

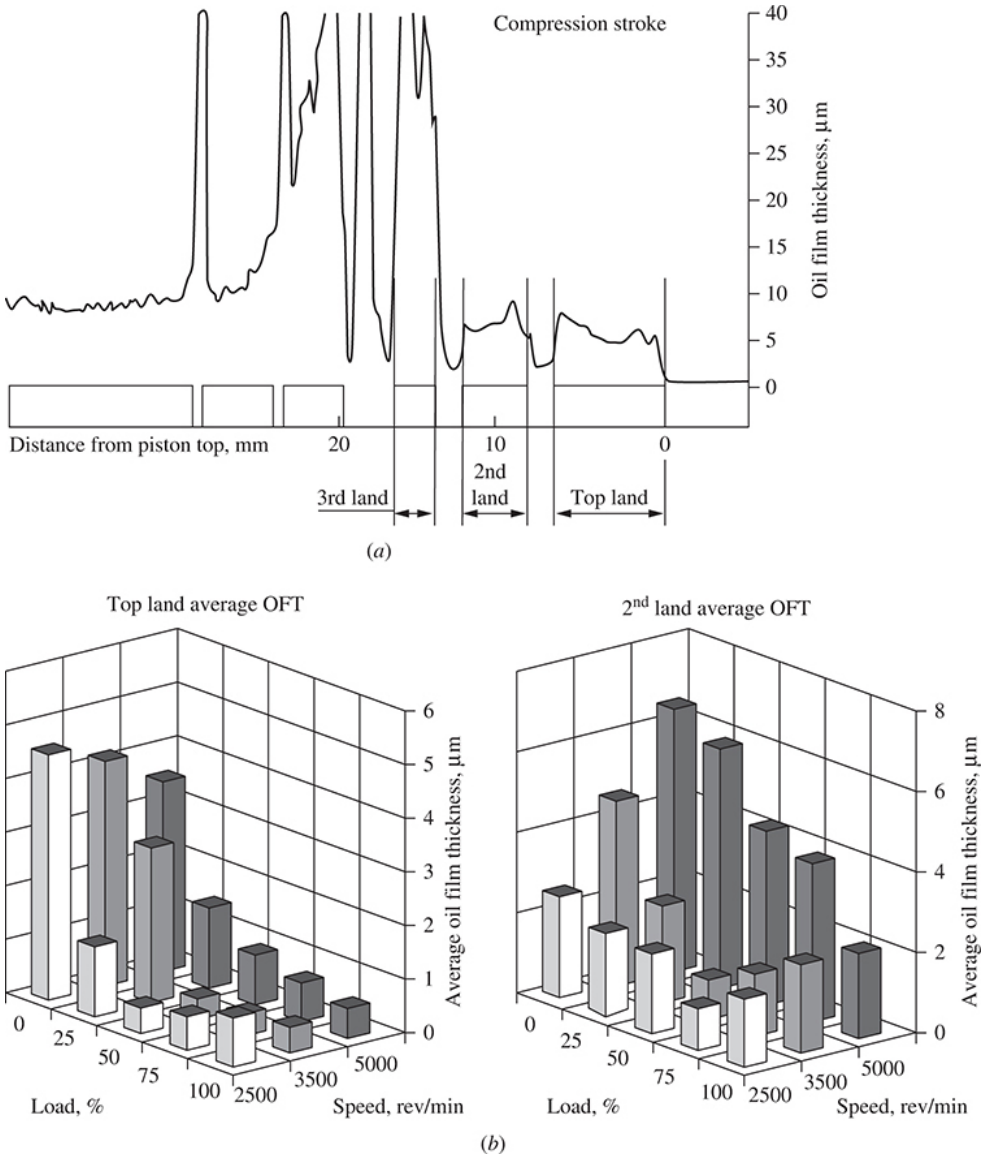


Figure 13.39 (*a*) Example of oil film thickness distribution between the piston and liner, along the piston and ring profile. (*b*) Average oil film thickness on the piston top land and second land, as a function of engine speed and load. Oil film thicknesses determined in a firing 1.8-liter four-

cylinder gasoline engine using a laser-induced fluorescence optical measurement technique illustrated in Fig. 13.20. [30]

13.10.3 Oil Evaporation

Oil evaporating from the piston-ring-liner system is thought to contribute significantly to total oil consumption. The oil evaporation process is influenced by oil volatility, thermal conditions in the region where evaporation occurs, and gas convection from the oil film surface. The physical situation where evaporation occurs is shown in Fig. 13.40: the oil concentration (ξ) and temperature profiles between the oil layer (on the liner) and adjacent gas are indicated. Distillation curves for a standard 10W-40 oil (see Sec. 13.11) and a synthetic oil are also shown. The more volatile oil components obviously vaporize more easily. [32]

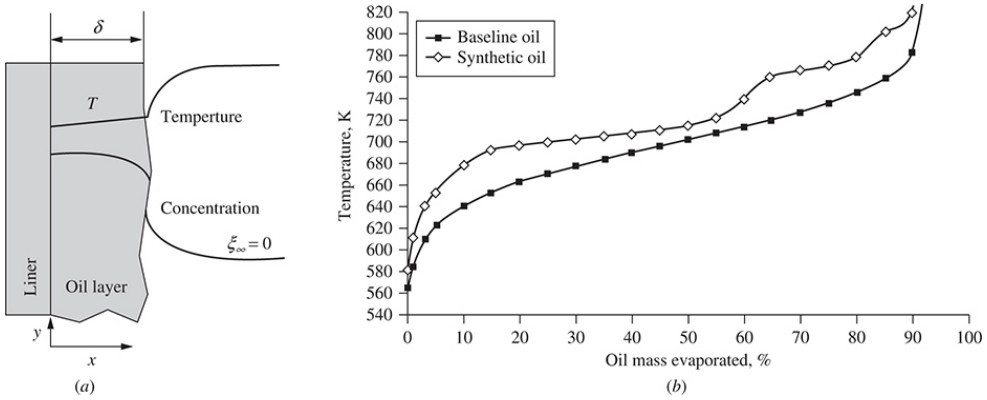


Figure 13.40 (*a*) Physical situation of oil film on the cylinder liner that leads to oil evaporation. (*b*) Distillation curves (percent oil evaporated versus temperature) for a standard 10W-40 oil and a synthetic oil. [30]

Evaporation primarily occurs from the oil film on the liner, and especially from above the TC location of the oil control ring. This portion of the liner is the hottest region, and it is well supplied with oil. Use of oils of different volatility changes the oil consumption in ways that support this upper-liner evaporative-oil-consumption source. Oil evaporation increases as engine load increases largely because of the increased thermal loading and thus higher liner temperatures. It also increases with increasing engine speed,

again due to increasing liner temperatures, and also due to higher mass transfer convection rates.

Figure 13.41 shows the three major oil consumption sources as a function of load and speed in a 1.8-liter four-cylinder NA gasoline engine.[30] As summarized above, oil evaporation (largely from the exposed oil film on the liner) rises significantly with increasing load and speed. The oil transport oil-consumption source dominates at low engine loads and becomes comparable to the evaporative source at about 80% load. Evaporation and transport sources grow together with increasing engine speed. Blowby, as a mechanism for oil consumption, is much smaller: some 10% of the total oil consumption (see Sec. 3.10.3).

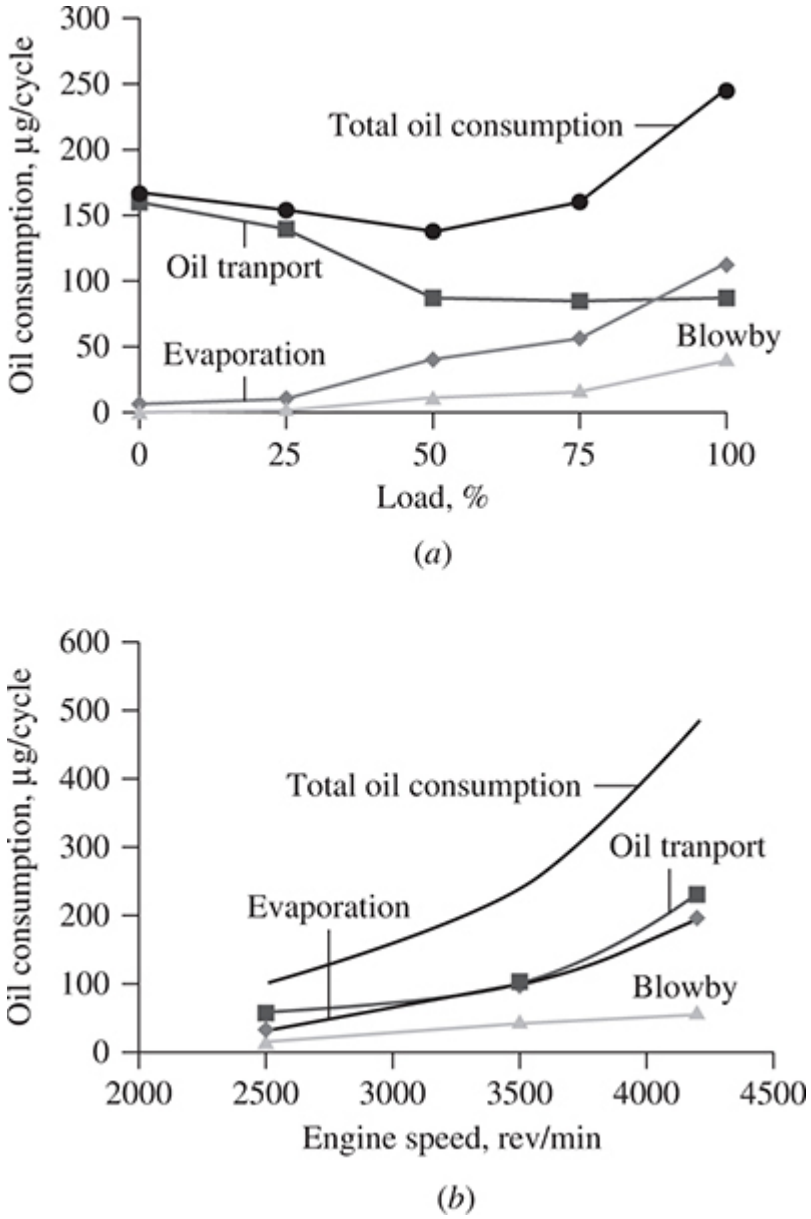


Figure 13.41 (*a*) Effect of engine load on the different oil consumption sources, and total oil consumption, at 3500 rev/min; (*b*) effect of engine speed on the different oil consumption sources, and total oil consumption, at 100% load for the baseline oil: four-cylinder 1.8-liter gasoline engine.[30]

13.10.4 Blowby and Oil Entrainment

The gases that "blow by" the piston and rings enter the crankcase, and are recycled via the engine intake manifold to the combustion chamber. Any hydrocarbons in vapor form, and oil "mist" that passes through the inertial

oil separator that is part of the positive crankcase ventilation system, will get burned. Oil thus consumed contributes about 10% to total oil consumption. The blowby flow is primary the flow through the ring gaps. Blowby oil consumption increases with increasing engine load and speed (see Fig. 13.41). Figure 13.42 indicates why: the blowby gas volume (cm^3/cycle)—in a 1.8-liter four-cylinder NA gasoline engine—increases at lower engine speeds with increasing load from about 7 to 23 cm^3/cycle (about 1% of the gas in the cylinder) due to the increasing in-cylinder pressures. Blowby decreases (in cm^3/cycle) with increasing speed at all loads primarily because, as speed rises, there is less time per cycle for blowby gas to flow through the ring gap flow restrictions. Blowby contributes little to oil consumption at the lighter loads where oil transport from below the piston which enters the combustion chamber is the dominant (more than 90%) mechanism. At high loads (and almost independent of speed) blowby is responsible for some 20%, and oil transport and oil evaporation contribute about 40%, each. [30]

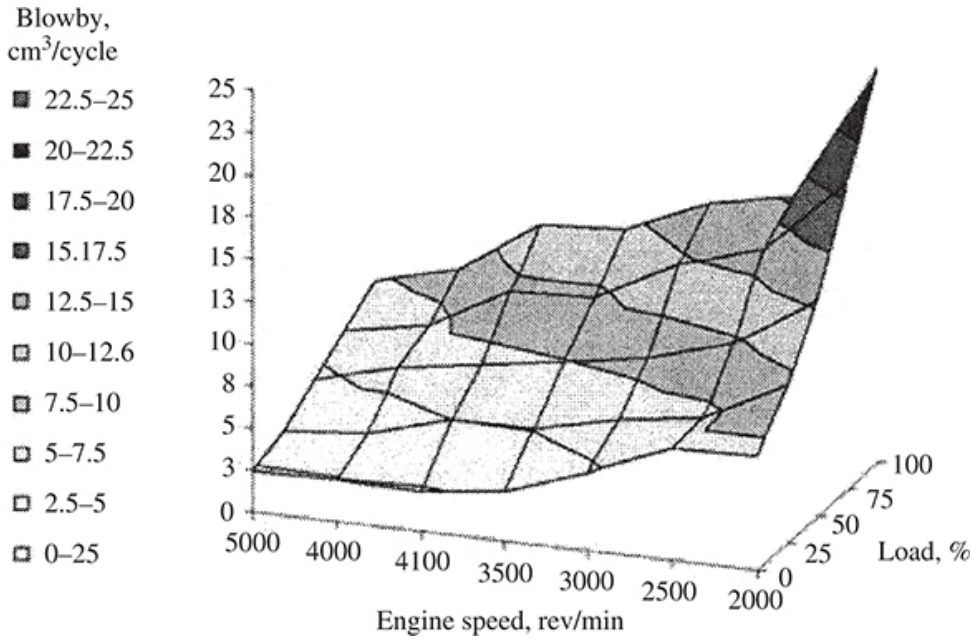


Figure 13.42 Blowby gas flow past the piston ring pack (at standard atmospheric pressure and temperature) into the crankcase: dependency on engine speed and load.[30]

13.11 LUBRICANTS

Lubricants separate two surfaces in relative motion. By forcing the surfaces apart through the pressure generated in this thin lubricant layer, direct contact between the two solid surfaces is prevented (or a least substantially reduced) thus minimizing wear and reducing friction. Lubricants also act as coolants, friction-surface sealants, and corrosion inhibitors.

The lubricant and the lubricating system perform the following functions: [3]

- Reduce the frictional resistance between moving parts of the engine to a minimum to ensure low friction and high mechanical efficiency.
- Protect the engine against wear in the above engine regions.
- Contribute to cooling the piston and regions of the engine where friction work is dissipated.
- Remove all injurious impurities and particles from lubricated regions.
- Hold gas and oil leakage (especially in the ring region) at an acceptable minimum level.

The base stock for engine lubricants (some 75 to 85% of the oil by volume) consists of a blend of higher molecular weight hydrocarbon compounds primarily selected for their viscosity characteristics. These hydrocarbons in the base stock may be refined from crude oil or, in synthetic lubricants, created through chemical processing. The remainder of the lubricant is the additive package containing a variety of chemical compounds selected to improve the lubricant's performance. Usually, these additive packages include dispersants and detergents, over-base (acid neutralizing) additives, anti-wear and anti-corrosion additives, viscosity improvers, and anti-oxidants. [12]

Note that during spark-ignition engine starting, especially in cold ambient conditions, gasoline fuel can flow past the piston rings and mix with the oil. As the engine and oil warm-up, the gasoline in the oil will vaporize. The presence of significant amounts of gasoline (or other liquid SI engine fuels) in the lubricant degrades many of its important characteristics.

Table 13.2 lists the qualities required of engine oils to perform the main lubrication system functions. These qualities can be summarized under the following headings. [33]

TABLE 13.2 Functions and qualities required of engine oils

Main functions required	Where and when	Qualities required
Reduce frictional resistance	During cold-starting Between con-rod/crankshaft bearings, and journals Between pistons, rings, and cylinders	Low enough viscosity to provide good pumping and avoid undue cranking resistance Minimum viscosity without risk of metal-to-metal contact under the varying conditions of temperature, speed, and load Sufficiently high viscosity at high temperatures; good lubrication property outside the hydrodynamic condition, especially at top-center Anti-seizure properties, especially during the run-in period
Protect against corrosion and wear	During shut-down or when running at low temperature	Must protect metallic surfaces against corrosive action of fuel decomposition products (water, SO_2, HBr, HC1, etc.) Must resist degradation (resist oxidation, have good thermal stability)
	In normal running	Must counteract action of fuel and lubricant decomposition products at high temperatures, especially on non-ferrous metals. By intervention in the friction mechanism must reduce the consequences of unavoidable metal-to-metal contact Must resist deposit formations which would affect lubrication (detergency or dispersive action). Must contribute to the elimination of dust and other contaminants (dispersive action)
Assist sealing	In the ring zone, especially at TC	Must have sufficient viscosity at high temperatures and low volatility Must limit ring and liner wear Must not contribute to formation of deposits in ring grooves and must prevent such formation
Contribute to cooling	Chiefly of pistons, rings, and connecting rod bearings	Must have good thermal stability and oxidation. Must have low volatility Viscosity must not be too high
Facilitate the elimination of undesirable materials	During oil drains to eliminate atmospheric dust, soot from diesel engines, salt, wear debris, organic products from burned fuel and lubricants, and other contaminants which promote deposits or accelerate wear	Must be able to maintain in fine suspension all solid material (dispersivity) whatever the temperature and physical and chemical conditions (water) Must be able to solubilize certain organic compounds, particularly heavy oxidation products

Source: From Schilling.[33]

Oxidation Stability

The temperature of the oil and engine parts it contacts, the presence of oxygen, the nature of the metal surfaces and debris, and the products of the fuel combustion, all influence the oxidation of the hydrocarbon components in lubricating oil. High temperatures are the primary factor, and the top piston ring groove and the crankcase are the critical regions. The temperature of the top ring groove can easily reach 250°C. The lubricating oil when subject to these conditions must not, through oxidation, contribute to deposit formation, even after long periods of running. These deposits would eventually lead to ring sticking which results in excessive blowby. At high temperatures, deposits are related to the oxidized fraction of the oil.

The oil temperature in the crankcase is about 120 to 130°C, though oil can be exposed to higher temperatures. Oil at this temperature should neither form significant acid products capable of attacking the bearing alloys, nor form insoluble products which form deposits. However, to ensure that quality mineral oils stand up well over time to the range of temperatures the oil

experiences, anti-oxidant and anti-corrosive additives are used. While anti-oxidants help to reduce deposit formation, detergent/dispersant additives are required to maintain any insoluble materials formed through oxidation in suspension.

Detergency/Dispersion

Deposits in the oil are controlled by its detergency. The amount of deposits formed depends on the fuel used, the quality of combustion, the temperature of the lubricating oil and coolant, and on the effectiveness of gas sealing at the piston rings. The detergency property is given to straight mineral oils by additives; the function of the detergent additive is to reduce the amount of deposits formed and make their removal easier.

At low temperatures, deposits are mainly byproducts of fuel combustion, and the detergency function is to keep them in suspension or solution in the oil. At high temperatures, deposits come from the oxidized fractions of the oil. The detergency function here is both to keep these products in suspension and to inhibit the reactions that lead to the formation of varnishes and lacquers. In addition, in diesel engines, the detergency helps to neutralize the acidic reaction products from the sulfur compounds in the fuel.

Wear Reduction

Wear is due to the individual and combined effects of corrosion, adhesion (i.e., metal-to-metal contact), and abrasion.

Corrosive attack by acidic products of combustion is one of the chief causes of cylinder and ring wear. The effect is worst at low cylinder wall temperatures. In diesel engines, any sulfur in the fuel increases the corrosive wear. Corrosive wear is effectively prevented by the use of detergent oils which neutralize the corrosive acids as they form, and by designing the cooling system to control metal temperatures appropriately.

Adhesive wear affects certain parts of the engine. In the upper cylinder, metal-to-metal contact between piston, rings, and cylinder walls takes place each time the engine is started (most significant during cold-starts) because there is insufficient oil in the top portion of the engine. Oils with anti-wear additives and low viscosity at low temperatures provide a partial remedy. Adhesive wear also occurs on components such as cams, tappets, drive gears, rocker arm ends, and valve stems. A commonly used class of anti-wear inhibitors is zinc-dialkyl-dithio-phosphates (ZDDPs).

Abrasion results from the presence of atmospheric dust, and metallic debris from corrosive and adhesive wear, in the lubricating oil. Efficient air filtration is therefore most important (see Ref. 34 for a discussion of air filters). Elimination of abrasive particle impurities from the oil system by filtration and periodic oil change is essential.

Viscosity

For low resistance to cranking and ease of starting, and rapid distribution of the oil while the engine is cold, a low oil viscosity at low ambient temperatures is required. When the engine (and oil) is fully warmed up, viscosity in the proper range is important for adequate sealing of the piston, acceptable oil consumption, and low hydrodynamic friction losses. The viscosity of the oil at both low and normal engine temperatures (a spread of some 200 K) is, therefore, important. The viscosity of lubricating oils decreases with increasing temperature. The *pour point*, *viscosity*, and *viscosity index* are used to characterize the behavior of a lubricating oil for these aspects of engine operation.

The *pour point* is determined by cooling a sample of oil in a test jar until, when the jar is rotated from the vertical to the horizontal, no perceptible movement of the oil will occur within 5 seconds; 5°F above this temperature is the pour point.

The viscosity of lubricating oils is determined by measuring the time required for a specified volume of oil to flow through a capillary tube or orifice, contained in a constant temperature water bath. The *kinematic viscosity*, ν ($\nu = \mu/\rho$), is determined by this method. Use of a Saybolt tube with an orifice of specified diameter is the standard U.S. measurement practice. The viscosity is then given by the time t (in seconds) required to flow 60 cm^3 of oil, and is expressed as Saybolt universal seconds, SUS. Approximate conversion to centistokes (1 centistoke = 10^{-6} m^2/s) can be obtained via

$$\nu = at - \frac{b}{t}$$

where for $115 > t > 34$ s, $a = 0.224$, and $b = 185$; for $215 > t > 115$ s, $a = 0.223$, and $b = 155$; and for $t > 215$ s, $a = 0.2158$ and $b = 0$. [35]

The viscosity of lubricating oils decreases with increasing temperature. Since engine oils must operate over a range of temperatures, a measure of the rate of decrease is important. The *viscosity index*, an empirical number indicating the effect of temperature changes on viscosity, is used for this purpose; [36] a low viscosity index indicates a relatively large change of viscosity with temperature. To increase the viscosity index, lubricating oils incorporate additives called "viscosity-index improvers." These are high molecular weight compounds (molecular weight $\approx 10^3$ to 10^4) whose primary function is to reduce the viscosity variation with temperature.

A discussion of lubricant viscosity, and its dependence on temperature can be found in Sec. 13.9 . Equations (13.23) to (13.25) define the relationship between ν and lubricant temperature T. Table 13.3 gives the constants for relating the low shear viscosity ν_0 values and the scaling ratio ($\mu\infty/\mu_0$) to obtain values relevant to lubricants in engine operation.[33]

TABLE 13.3 Lubricant's viscosity data for various engine oil grades

Oil grade	k (cSt)	θ_1 (°C)	θ_2 (°C)	μ_∞/μ_0	Viscosity, ν (cSt)	
					At 40°C	At 100°C
0W-40	0.01341	1986.4	189.7	0.67		
5W-20	0.4576	1224	134.1	0.94	49	8.6
5W-30					66	11
5W-40	0.15	1018.74	125.91	0.8		
10W-30	0.1403	869.72	104.4	0.76	75	10.8
10W-40					99	14.4
20W-50	0.0639	1255.46	117.7	0.84	174	19
SAE10	0.0258	1345.42	144.58	1		
SAE30	0.0246	1432.29	132.94	1		
SAE50	0.0384	1349.94	115.16	1		

Sources: Sandoval and Heywood,[8] Yilmaz,[32] Taylor et al.[37]

The lubricating oil classification used most extensively is the SAE classification. [38] It depends solely on the viscosity of the oil. The seven different classification numbers 5W, 10W, 20W, 20, 30, 40, and 50

correspond to viscosity ranges; increasing numbers correspond to increasing viscosity, as shown in Fig. 13.43. SAE numbers followed by W (abbreviation for winter) refer to oils for use in cold climates, and viscosity is determined at –18°C (0°F). SAE numbers without W are applied to engine oils commonly used under warmer conditions; they are based on viscosity at 99°C (210°F). Multi-grade oils (e.g., 10W-40) satisfy service requirements at low as well as high temperatures in terms of the SAE classification. The first number indicates the viscosity at −18°C; the second number at 99°C. Examples are shown in Fig. 13.29. Multi-grade oils have a higher viscosity index than single-grade oils, which make them more attractive for engine use.

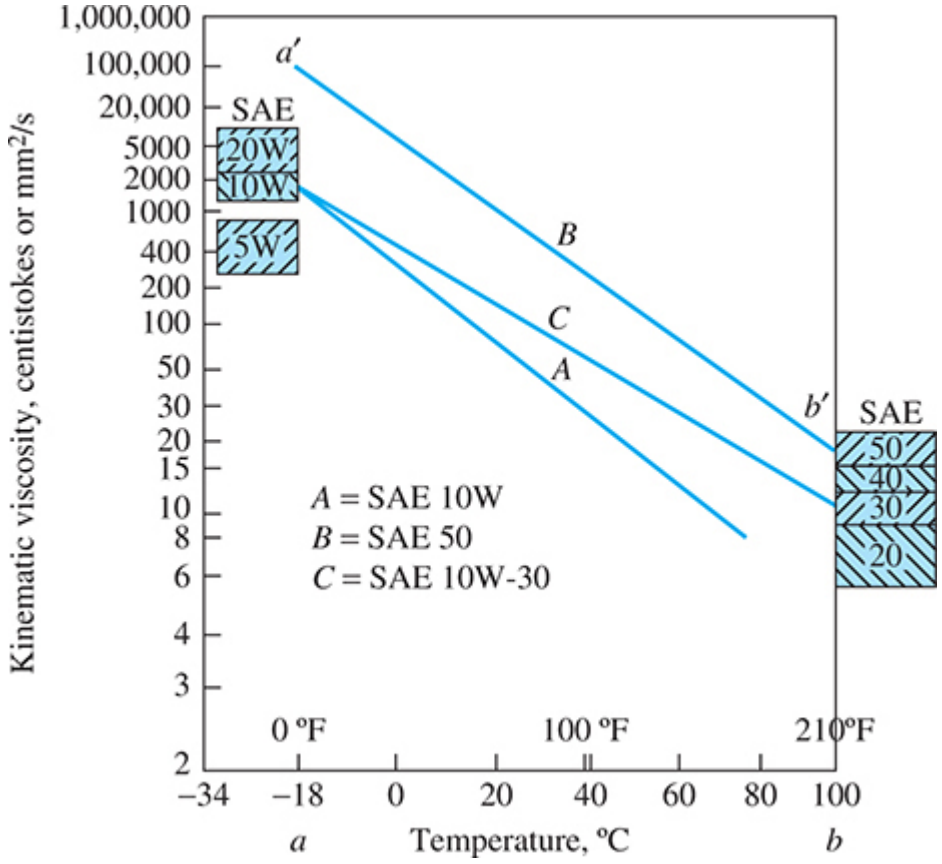


Figure 13.43 Viscosity versus temperature curves illustrating SAE lubricating oil classification.[33]

PROBLEMS

13.1 **(a)** Show how friction mean effective pressure for a four-stroke cycle engine can be obtained from the brake power P_b, engine speed N, displaced volume V_d, and $\int p dV$ over the compression and expansion strokes ($W_{c,ig}$). **(b)** How is pumping mean effective pressure related to $\int p dV$ over the compression and expansion strokes and $\int p dV$ over the full four-stroke cycle? **(c)** Find the brake power, total friction power, total friction imep, and pumping imep for a four-stroke cycle SI engine operating at 1800 rev/min with a measured brake torque of 32 N · m, a gross imep of 933 kPa, and a net imep of 922 kPa. $V_d = 0.496$ liter.

13.2 Three categories of friction are described in Sec. 13.3 : boundary friction, hydrodynamic (or fully-lubricated) friction, and turbulent dissipation. By means of Eq. (13.6) , estimate the relative proportion of total friction work per cycle in each category for a four-cylinder automobile spark-ignition engine operating at 3000 rev/min.

13.3 For four-stroke cycle naturally-aspirated multi-cylinder spark-ignition and diesel automobile engines at full load and one-third full load, at mid speed (2000 rev/min), give approximate estimates of the percentages of total friction mep in these three categories: pumping mep, rubbing friction mep, and accessory friction mep. State explicitly how you develop these estimates and what you include as accessories.

13.4 All of the friction measurement procedures, except the difference between brake and gross indicated power or mep measured directly, assume that motored engine friction and firing engine friction are closely comparable. This is not an accurate assumption for the pumping component. Summarize the differences between the gas exchange processes under motoring and firing conditions for a spark-ignition engine at a fixed part-load throttle setting that will result in the pumping work being significantly different under these two conditions.

13.5 On separate accurately proportioned sketches of the piston, cylinder, connecting rod, and crank mechanism (similar to Fig. 2.1), during the intake stroke (at 120° ATC), compression stroke (at 60° BTC), expansion stroke (at 60° ATC), and exhaust stroke (at 120° BTC), draw an arrow for each of the forces acting on the piston (pressure forces, force from connecting rod,

friction force, inertia force). Mark clearly the *positive* direction of each force. Express each force in terms of cylinder pressure p_c, crankcase pressure p_{cc}, friction force F_f, piston area A_p, effective piston (and part of connecting rod) mass m_p, piston acceleration a, connecting rod force F_{cr}.

13.6 **(a)** For the DI diesel engine friction force data shown in Fig. P13.6, estimate the maximum pressure force on the piston (under full-load conditions) and the approximate magnitude of the inertia force [mass of piston plus part of the connecting rod (7 kg) × $\overline{S}_p$ × (N/4)]. Compare these forces with the piston friction force at time of peak pressure.

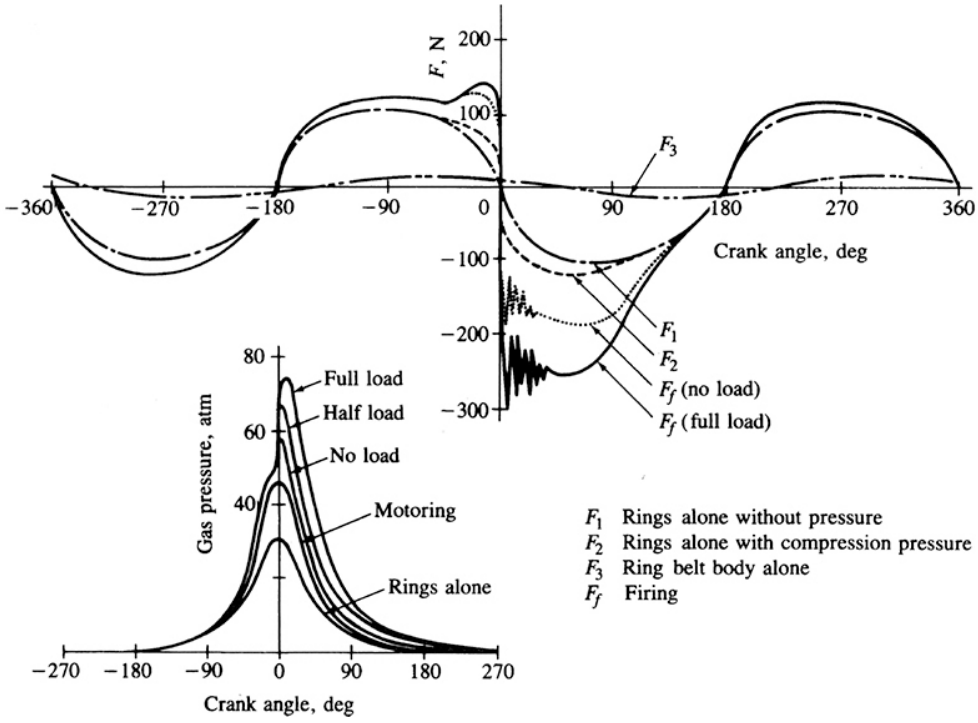


Figure P13.6 Measured frictional force on cylinder liner of 137 mm bore and 135 mm stroke single-cylinder DI diesel engine. 1200 rev/min, coolant temperature 80°C, cylinder liner inside temperature 90°C. [15]

(b) The figure shows the variation in friction force acting on the piston of a DI diesel engine under no-load and full-load firing conditions. Carefully sketch the shape (indicating direction and rough magnitude) of the cylinder pressure force on the piston, the piston velocity, and the piston acceleration, as functions of crank angle on the same graph as these friction forces. Use

these graphs to explain the variation of piston friction force throughout the four strokes of the cycle.

13.7 Show by dimensional analysis of the variables that govern the friction in a journal bearing (friction force F_f, oil viscosity μ, bearing diameter D_b, length L_b, mean clearance $\bar{h}$, shaft rotational speed N) that

$$\frac{F_f}{\mu D_b^2 N} = f\left(\frac{L_b}{D_b}, \frac{\bar{h}}{D_b}\right)$$

What additional physical assumptions are then required to obtain an equation of the form of Eq. (13.11) ?

13.8 Explain whether each of the following components of engine friction would be expected to depend on (1) crankshaft rotational speed N, (2) mean piston speed S_p, (3) or both of these variables. Crankshaft journal bearings, connecting rod bearings, valve train, piston rings, piston skirt, water pump, fan, valve flow loss (resistance to flow through the inlet and exhaust valves).

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[a] This definition gives $W_p > 0$ for naturally-aspirated engines. This is then added to the other friction components (and thus subtracted from the work transferred to the piston). For supercharged and turbocharged engines at high loads, where p_i is greater than p_e, this definition gives $W_p < 0$; thus it

adds to the work transferred to the piston. (Different sign conventions for pumping work are sometimes used in order to maintain Wp as a positive quantity.)

[b] This neglects the second-order effect of the efficiency benefit that results from reduced pumping work.

[c] The fuel pump in a gasoline engine is electrically driven and its power requirements are small; the diesel fuel pump, which must deliver fuel at pressures of some 800 bar, needs to be included.[27]

[d] These oil flows are discussed in an overview sense. The details of these flows are only partly understood.

CHAPTER 14

Modeling Real Engine Flow and Combustion Processes

14.1 PURPOSE AND CLASSIFICATION OF MODELS

In engineering, modeling a process means developing and using the appropriate combination of assumptions and equations that permit critical features of the process to be analyzed. The modeling of engine processes continues to evolve as our basic understanding of the physics and chemistry of the phenomena of interest steadily expands and as the capability of computers to solve complex equations continues to increase. Modeling activities can make major contributions to engine engineering at different levels of generality or detail by:

- Developing a more complete understanding of the process under study from the discipline involved in formulating the model;
- Identifying key controlling variables that provide guidelines for better structured and therefore less costly experimental development efforts;
- Predicting engine behavior over a wide range of design and operating variables to screen concepts prior to major hardware programs, to determine trends and tradeoffs, and, if the model is sufficiently accurate, to optimize design and control;
- Providing a rational basis for design innovation.

Each of these contributions is valuable. Whether a model is ready to pass from one stage to the next depends on the accuracy with which it represents the actual process, the extent to which it has been tested and validated, and the time and effort required to use the model for extensive sets of calculations and interpret the results.

This chapter reviews the types of models and their primary components that are used to describe engine performance, fuel consumption, and emissions characteristics. These models describe the thermodynamic, fluid-flow, heat-transfer, combustion, and pollutant-formation phenomena that govern these aspects of engine performance. Many of the building blocks for these models have been described in previous chapters. The purpose here is to show how fluid dynamics, heat-transfer, thermodynamics, and kinetics fundamentals can be combined at various levels of sophistication and complexity to predict, with varying degrees of completeness, internal combustion engine combustion and emissions processes, and engine operating characteristics.

For the processes that govern engine performance and emissions, two basic types of models have been developed. These can be categorized as *thermodynamic* or *fluid dynamic* in nature, depending on whether the equations which give the model its predominant structure are based on energy conservation or on a full analysis of the fluid motion. Other labels given to thermodynamic energy-conservation-based models are: zero-dimensional (since in the absence of any flow modeling, geometric features of the fluid motion cannot be predicted), phenomenological (since additional detail beyond the energy conservation equations is added for each phenomenon in turn), and quasi-dimensional (where specific geometric features, e.g., the spark-ignition engine flame or the diesel fuel spray shapes, are added to the basic thermodynamic approach). Fluid-dynamic-based models (referred to as computational fluid dynamics, CFD, models) are often called multi-dimensional, due to their inherent ability to provide detailed geometric information on the flow behavior, based on solution of the governing flow equations.

Some general observations about models of engine processes provide a context for the details that follow. The physical and chemical processes involved are extremely complex. While much is known about these processes, they are not fully understood at a fundamental level. At present, it is not possible to construct models that predict engine operation from the

basic governing equations alone. Thus the objectives of any modeling effort should be clearly defined, and the structure and detailed content of the model should be appropriate to these objectives. It is not yet feasible to construct models that attempt to describe *all* important aspects of engine operation: more limited objectives are usually appropriate.

Due to this complexity of engine processes and our inadequate understanding at a fundamental level, most engine models are incomplete. Empirical relations and *ad hoc* approximations are often needed to bridge gaps in our understanding of critical phenomena. Since models will continue to develop greater completeness and generality, the emphasis in this chapter is on the basic relationships used in engine process models rather than the current status of these models.

Finally, an important issue in any overall engine model is balance in complexity and detail amongst the process submodels. A model is no more accurate than its weakest link. Thus it makes sense to describe the key phenomena involved at comparable levels of sophistication.

14.2 GOVERNING EQUATIONS FOR AN OPEN THERMODYNAMIC SYSTEM

It is often required to model a region of the engine as an open thermodynamic system. Examples are the cylinder volume and the intake and exhaust manifolds (or portions of these volumes). Such an approach is appropriate when the gas inside the system boundary can be assumed uniform in composition and state at each point in time, and when that state and composition vary with time due to heat transfer, work transfer and mass flow across the boundary, and due to boundary displacement. Such an open system is illustrated in Fig. 14.1. The important equations are mass and energy conservation. These equations for an open system, with time or crank angle as the independent variable, are the building blocks for thermodynamic-based engine models.

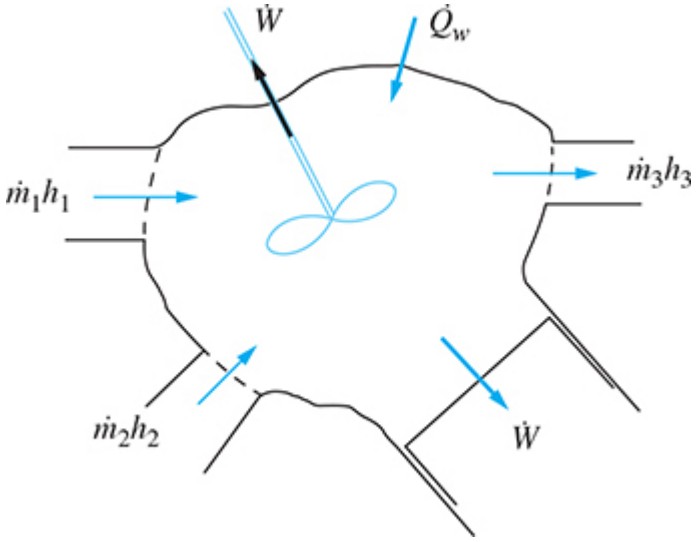


Figure 14.1 Schematic of an open thermodynamic system.

14.2.1 Conservation of Mass

The rate of change of the total mass m of an open system is equal to the sum of the mass flows into and out of the system, $\dot{m}_j$:

$$\dot{m} = \sum_j \dot{m}_j \tag{14.1}$$

Mass flows into the system are taken as positive, mass flows out as negative. For conservation of the fuel chemical elements, it is convenient to use the fuel fraction f, which is defined as mf / m, where m_f denotes the mass of fuel (or fuel elements in the combustion products) in the open system:

$$\dot{m}_f = \frac{d}{dt}(mf) = \sum_j \dot{m}_{f,j} = \sum_j \dot{m}_j f_j \tag{14.2}$$

Differentiation of Eq. (14.2) leads to an equation for the rate of change of fuel fraction:

$$\dot{f} = \sum_j \left(\frac{\dot{m}_j}{m}\right)(f_j - f) \tag{14.3}$$

The fuel/air equivalence ratio φ is related to f via $\varphi = f/[(F/A)_s(1-f)]$. Hence the rate of change of equivalence ratio of the material in the open system is

$$\dot{\phi} = \frac{1}{(F/A)_s}\frac{\dot{f}}{(1-f)^2} \tag{14.4}$$

14.2.2 Conservation of Energy

The first law of thermodynamics for the open system in Fig. 14.1 can be written:

$$\dot{E} = \dot{Q}_w - \dot{W} + \sum_j \dot{m}_j h_j \tag{14.5}$$

$\dot{Q}_w$ is the total heat-transfer rate into the system, across the system boundary, and equals the sum of the heat-transfer rates across each part of the boundary, $\Sigma_i \dot{Q}_{w,i}$. $\dot{W}$ is the worktransfer rate out of the system across the boundary: where the piston is displaced, the worktransfer rate equals $p\dot{V}$. Because all energies and enthalpies are expressed relative to the same datum (see Sec. 4.5.3), it is not necessary to explicitly identify the heat released by combustion in Eq. (14.5) : this is already accounted for in the energy and enthalpy terms.

The goal here is to define the rate of change of state of the open system in terms of $\dot{T}$ and $\dot{p}$. Two approaches are commonly used, depending on whether the thermodynamic property routines provide values for internal energy u or enthalpy h. Thus $\dot{E}$ in Eq. (14.5) can be expressed as

$$\dot{E} = \frac{d}{dt}(mu) \quad \text{or} \quad \dot{E} = \frac{d}{dt}(mh) - \frac{d}{dt}(pV) \tag{14.6a,b}$$

It is assumed that the system can be characterized by T, p, and φ; thus

$$u = u(T,p,\phi) \quad h = h(T,p,\phi) \quad \rho = \rho(T,p,\phi) \tag{14.7}$$

and the rate of change of u, h, and ρ can be written in the form

$$\dot{\alpha} = \left(\frac{\partial \alpha}{\partial T}\right)\dot{T} + \left(\frac{\partial \alpha}{\partial p}\right)\dot{p} + \left(\frac{\partial \alpha}{\partial \phi}\right)\dot{\phi} \tag{14.8}$$

where α is u, h, or ρ. Using the ideal gas law in its two forms, $p = \rho RT$ and $pV = mRT$, and Eq. (14.8) for $\dot{\alpha}$, an equation for $\dot{p}$ can be derived:

$$\dot{p} = \frac{\rho}{\partial \rho / \partial p}\left(-\frac{\dot{V}}{V} - \frac{1}{\rho}\frac{\partial \rho}{\partial T}\dot{T} - \frac{1}{\rho}\frac{\partial \rho}{\partial \phi}\dot{\phi} + \frac{\dot{m}}{m}\right) \tag{14.9}$$

Returning now to the energy conservation equation, Eq. (14.5) , expressing $\dot{E}$ in terms of $\dot{u}$ or $\dot{h}$, and $\dot{u}$ or $\dot{h}$ in terms of partial derivatives with respect to T, p, and φ, and substituting for $\dot{p}$ with Eq. (14.9) , one can obtain equations for $\dot{T}$:

$$\dot{T} = \left[B - \frac{p}{D}\frac{\partial u}{\partial p}\left(\frac{\dot{m}}{m} - \frac{\dot{V}}{V} + \frac{\partial R}{\partial \phi}\frac{\dot{\phi}}{R}\right) - \frac{\partial u}{\partial \phi}\dot{\phi}\right] \Bigg/ \left(\frac{\partial u}{\partial T} + \frac{C}{D}\frac{p}{T}\right) \tag{14.10}$$

where

$$B = -RT\frac{\dot{V}}{V} + \frac{1}{m}\left(\dot{Q}_w + \sum_j \dot{m}_j h_j - \dot{m}u\right)$$

$$C = 1 + \frac{T}{R}\frac{\partial R}{\partial T}, \quad \text{and} \quad D = 1 - \frac{p}{R}\frac{\partial R}{\partial p}$$

(see Ref. 1, for example). From Ref. 2,

$$\dot{T} = \frac{B'}{A'}\left[\frac{\dot{m}}{m}\left(1 - \frac{h}{B'}\right) - \frac{\dot{V}}{V} - \frac{C'}{B'}\dot{\phi} + \frac{1}{B'm}\left(\sum_j \dot{m}_j h_j - \dot{Q}_w\right)\right] \tag{14.11}$$

where

$$A' = \frac{\partial h}{\partial T} + \frac{\partial \rho / \partial T}{\partial \rho / \partial p}\left(\frac{1}{\rho} - \frac{\partial h}{\partial p}\right),$$

$$B' = \frac{1 - \rho(\partial h / \partial p)}{\partial \rho / \partial p}, \quad \text{and} \quad C' = \frac{\partial h}{\partial \phi} + \frac{\partial \rho / \partial \phi}{\partial \rho / \partial p}\left(\frac{1}{\rho} - \frac{\partial h}{\partial p}\right)$$

Equations (14.1) , (14.3), (14.4), (14.9), and (14.10) or (14.11) can now be used to obtain the state (defined by p and T) of the gases within the cylinder as a function of time. $\dot{V}$ is obtained from Eq. (2.6) , and the thermodynamic properties and their derivatives from the models described in Chap. 4 .

Often, for specific applications, the above equations can be simplified substantially. For the intake and exhaust systems (or sections of these systems such as the manifold or plenum, etc.), $\dot{V}$ is zero and effects of dissociation (the terms $\partial u/\partial p$, $\partial h/\partial p$, and $\partial R/\partial p$) can usually be neglected. For the cylinder during compression, dissociation can usually be neglected, also. Application of these equations during combustion must be related to the combustion model used. For the single-zone model often used in diesel engine simulations (see Sec. 10.4), the whole of the combustion chamber is treated as one system. For the two-zone model used for spark-ignition engine simulations, the unburned mixture zone and the burned mixture zone are each treated as separate open systems, with volumes V_u and V_b, respectively, where $V_u + V_b = V$. If a thermal boundary-layer region is included (see Sec. 12.6.5), an additional open system must be defined.

This open-system energy-balance approach is embedded in engine "cycle-simulation" codes such as GT-Power [3] and WAVE. [4]

14.3 INTAKE AND EXHAUST FLOW MODELS

14.3.1 Background

The behavior of the intake and exhaust systems is important because these systems govern the air flow into the engine's cylinders, and the outflow of burned gases. Inducting the maximum airflow at full load at any given speed and retaining that mass within the engine's cylinders is a primary design goal.

The higher the airflow, the larger the amount of fuel that can be burned and the greater the power produced. The exhaust and intake behavior control the amount of hot residual burned gas left inside the cylinder. The important airflow parameters are volumetric efficiency (for four-stroke cycle engines) or scavenging and trapping efficiencies (for two-stroke cycle engines), along with equal air flows to each engine cylinder (see Secs. 6.2 , 6.6, and 7.6.2).

The objectives for any manifold model have an important bearing on its complexity and structure. If the goal is to provide the input or boundary conditions to a detailed model of in-cylinder processes, then sophisticated intake and exhaust system models are not necessarily required. If the manifold flows are the primary focus, then models that adequately describe the unsteady gas-flow phenomena which occur are normally needed. Then simple models for the in-cylinder phenomena often suffice to connect the intake and exhaust processes. The valves and ports, which together provide the major restriction to the intake and exhaust flow, largely decouple the manifolds from the cylinders.

Three types of models for calculating details of intake and exhaust flows have been developed and used:

1. Quasi-steady models for flows through the restrictions which the valve and port (and other components) provide
2. Filling and emptying models which account for the finite volume of critical manifold components
3. Gas dynamic models which describe the spatial variations in flow and pressure throughout the manifolds

Each of these types of models can be useful for analyzing engine behavior. The appropriate choice depends on objectives, and the time and effort available. Each will now be reviewed.

14.3.2 Quasi-Steady Flow Models

Here the manifolds are considered as a series of interconnected components, which each constitute a significant flow restriction: for example, air cleaner, throttle, port, and valve for the intake system. The flow restriction each of these components represents is defined by their geometry and discharge coefficient, usually determined empirically under steady-state conditions.

The gas flow rate through each component is computed using steady one-dimensional flow equations [see App. C, Eqs. (C.8) and (C.9)]: the actual flow is assumed to be quasi steady. These components are connected by the gas flow passing through them and the pressure ratios across them: mass accumulation between components is neglected.

In quasi-steady models, the assumption is made that any time-varying aspects of the processes involved are slow enough (have long-enough time constants) that the equilibrating process (with flows, usually pressure waves) is fast enough to maintain close to a quasi-equilibrium situation. Such models are often used to calculate the flows into and out of the cylinder through the inlet and exhaust valves (see Secs. 6.3 and 6.5 and Fig. 6.20). If the pressure variation with time upstream of the valve is known or is small, as usually occurs with large plenums and short manifold pipe lengths, such methods are accurate enough to be useful. This approach has been used extensively with engine cycle simulations, which predict engine performance characteristics from a thermodynamics-based analysis, to calculate the mass flow rates into and out of the cylinder (see Sec. 14.4). However, such methods are not able to accurately predict the variation of volumetric efficiency with engine speed because some of the phenomena which govern this variation are omitted from this modeling approach (see Sec. 6.2 and Fig. 6.12).

14.3.3 Filling and Emptying Methods

In "filling and emptying" models, the manifolds (or sections of manifolds) are represented by finite volumes where the mass of gas can increase or decrease with time. Such models can range from treating the whole intake or exhaust system as a single volume to dividing these systems into many sections, with flow restrictions such as the air cleaner, throttle valve, or inlet valve at the beginning, in between volumes, or at the end. Each volume is then treated as a control volume (an open system of fixed volume) which contains gas at a uniform state. The mass and energy conservation equations developed in Sec. 14.2 [Eqs. (14.1) , (14.3), (14.9), and (14.10) or (14.11)], coupled with information on the mass flow rates into and out of each volume [e.g., determined by the equations for flow through a restriction, Eqs. (C.8) and (C.9)], are used to define the gas state in each control volume. For intake and exhaust flows, these equations can be simplified since the volumes are fixed ($\dot{V} = 0$) and gas composition can be assumed frozen ($\partial u/\partial p$, $\partial h/\partial p$,

and $\partial R/\partial p$ are then zero). Unless backflow occurs or recycled exhaust is used for emission control, changes in fuel fraction are not significant, and for intake systems it may be acceptable to omit heat transfer to the walls ($\dot{Q}_w$). Such methods characterize the contents of the manifold (or a region thereof) with a single gas temperature, pressure, and composition. These vary periodically with time as each cylinder in turn draws on the intake system and discharges to the exhaust system. Also, under transient conditions when engine load and/or speed change with time, manifold conditions will vary until the new engine steady-state conditions are established. Watson and Janota [5] discuss the application of filling and emptying models to manifolds in more detail. Such models can characterize these time-varying phenomena, spatially averaged over each manifold region corresponding to each volume analyzed: however, they cannot describe the spatial variation of pressure (and other gas properties) due to unsteady gas dynamics in the manifolds.

An application of a filling and emptying model to the intake manifold of a spark-ignition engine was described in Sec. 7.6.2 . The manifold was analyzed as a single control volume with the throttle plate controlling mass flow into the manifold and the engine cylinders controlling mass flow out. An equation for the rate of change of manifold pressure [Eq. (7.9)] was derived and used to explain how the air flow past the throttle varied as the throttle open angle was increased, as would occur following the start of a vehicle acceleration at part-throttle conditions (see Fig. 7.24).

A second example will illustrate the conditions under which filling and emptying models give sufficiently accurate predictions to be useful. It concerns a single-cylinder two-stroke cycle loop-scavenged direct-injection diesel engine. The engine was modeled as three open systems (the intake system, the cylinder, and the exhaust system) connected by flow restrictions. Various exhaust manifold volumes and shapes were examined, using nozzles at the manifold exit to simulate the exhaust-driven turbine. The in-cylinder models were calibrated to match the measured engine performance. Figure 14.2 shows the predicted and measured pressure variation at the exhaust system exit for two exhaust system volumes (V_e). With the compact manifold, the measured and predicted pressures were in good agreement. With the larger exhaust system shown in the bottom of the figure ($V_e/V_d = 5.2$) and the compact manifold, good agreement is again obtained. Only with the larger volume and long pipe exhaust system, is there evidence in the measured

pressure variation of substantial unsteady gas-dynamic effects. When used appropriately, filling and emptying models can be a useful predictive tool.

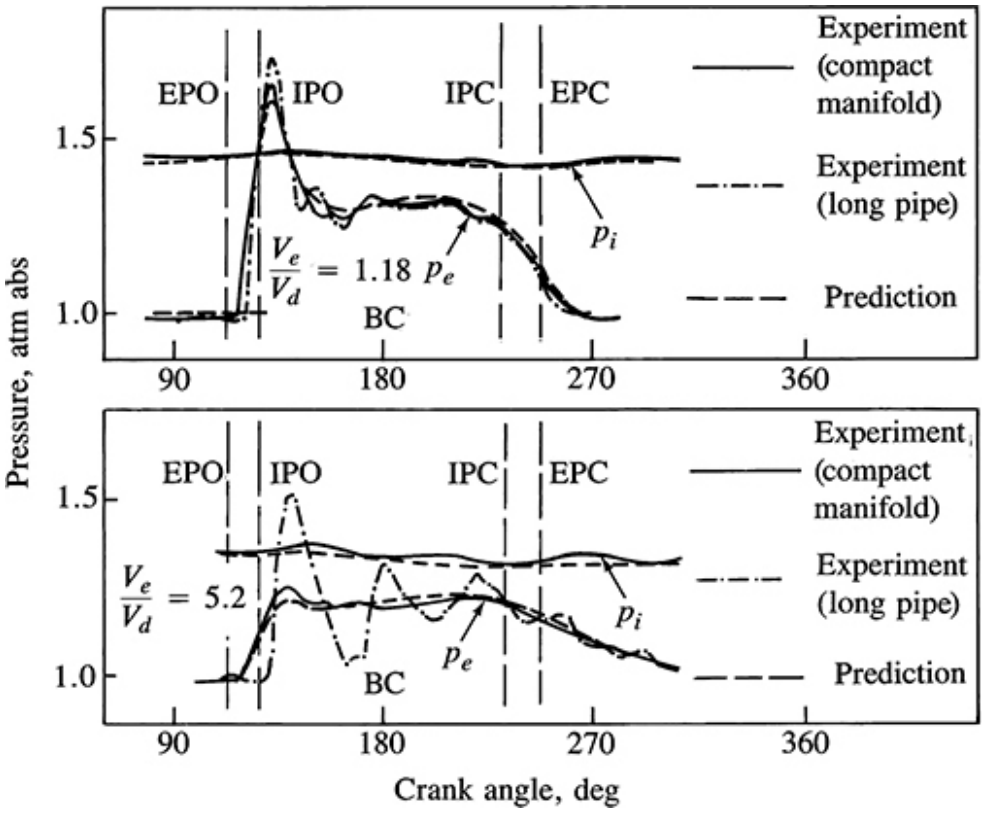


Figure 14.2 Comparison of intake and exhaust manifold pressures, p_i and p_e predicted by filling and emptying model, with experimental data. Single-cylinder two-stroke loop-scavenged direct-injection diesel engine. Top and bottom: Different ratios of exhaust system volume V_e to displaced volume V_d, and exhaust manifold shapes. [6] EPO, exhaust port opens; IPO, inlet port opens; IPC, inlet port closes; EPC, exhaust port closes.

14.3.4 Gas Dynamic Models

Many intake and exhaust system design variables determine their performance. These variables include the length and cross-sectional area of both primary and secondary runners, the volume and location of the plenums or junctions which join the various runners, the entrance or exit angles of the runners at a junction, the number of engine cylinders and their dimensions, intake and exhaust port and valve design, and valve lift and timing (see Secs.

6.2 , 6.3, 6.7, and 7.6). Most of this geometric detail is beyond the level that can be incorporated into the models discussed above. Coupled with the pulsating nature of the flow into and out of each cylinder, these details create significant gas dynamic effects on intake and exhaust flows which require a more complete modeling approach.

Gas dynamic models have been used for many years to study engine gas exchange processes. These models utilize the mass, momentum, and energy conservation equations for the unsteady compressible flow in the intake and exhaust. Normally, the *one-dimensional* unsteady flow equations are used. [a] These models often use a thermodynamic analysis of the in-cylinder processes to link the intake and exhaust flows. In the past, the method of characteristics was used to solve the gas dynamic equations. Finite difference techniques are used in more recent intake and exhaust flow models. The basic equations and assumptions of these models will now be reviewed. [7]

Unsteady Flow Equations

Consider the flow through the control volume within a straight duct shown in Fig. 14.3. It is assumed that the area change over the length d_x of the control volume is small so the flow is essentially one dimensional. Mass conservation requires that the rate of change of mass within the control volume equals the net flow into the control volume: that is,

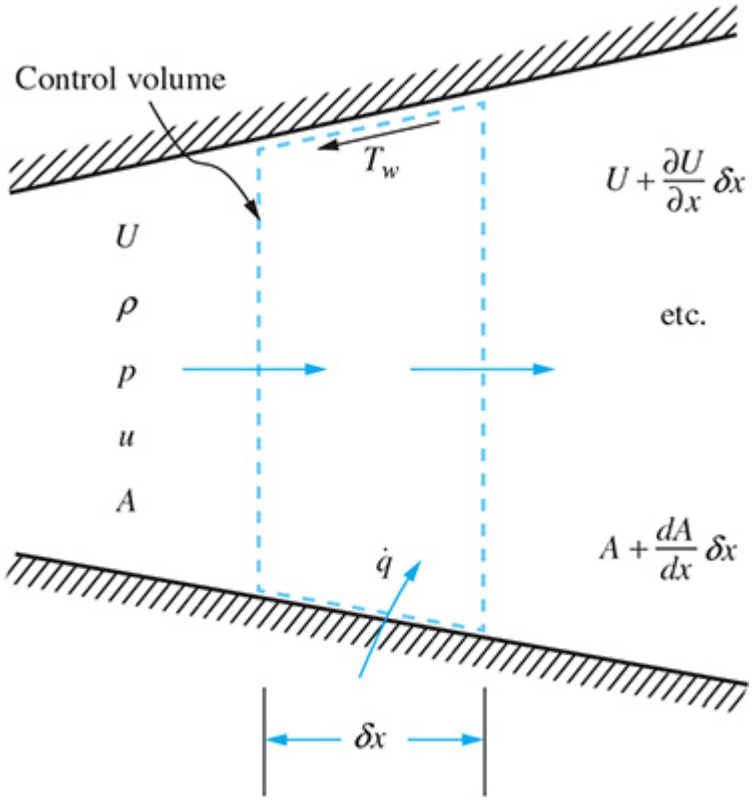


Figure 14.3 Control volume for unsteady one-dimensional flow analysis.

$$\frac{\partial}{\partial t}(\rho A \delta x) = \rho A U - \left[\rho A U + \frac{\partial}{\partial x}(\rho A U)\delta x\right] \tag{14.12}$$

Retaining only first-order quantities, this equation simplifies to

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho U) + \frac{\rho U}{A}\frac{dA}{dx} = 0 \tag{14.13}$$

The momentum conservation equation states that the net pressure forces plus the wall shear force acting on the control volume surface equal the rate of change of momentum within the control volume plus the net flow of momentum out of the control volume. The net forces and momentum changes are given by:

Pressure forces:

$$pA-\left(p+\frac{\partial p}{\partial x}\delta x\right)\left(A+\frac{dA}{dx}\delta x\right)+p\frac{dA}{dx}\delta x=-A\frac{\partial p}{\partial x}\delta x$$

Shear forces:

$$-\tau_w \pi D\delta x=-\xi\frac{\rho U^2}{2}\pi D\delta x$$

where D is the equivalent diameter $(4\,A/\,\pi)^{1/2}$, and ξ is the friction coefficient given by $\tau\,w/(\,\rho\,U^2/2)$.

The rate of change of momentum within the control volume is

$$\frac{\partial}{\partial t}(U\rho A\delta x)$$

and the net efflux of momentum across the control volume surface is

$$\left(\rho+\frac{\partial \rho}{\partial x}\delta x\right)\left(U+\frac{\partial U}{\partial x}\delta x\right)^2\left(A+\frac{dA}{dx}\delta x\right)-\rho U^2 A=\frac{\partial}{\partial x}(\rho U^2 A)\delta x$$

Combining these terms into the momentum conservation equation yields

$$-A\frac{\partial p}{\partial x}\delta x-\xi\frac{\rho U^2}{2}\pi D\delta x=\frac{\partial}{\partial t}\left(\rho UA\,\delta x\right)+\frac{\partial}{\partial x}(\rho U^2 A)\,\delta x \qquad (14.14)$$

This can be rearranged and combined with the mass conservation equation (14.13) to give

$$\frac{\partial U}{\partial t}+U\frac{\partial U}{\partial x}+\frac{1}{\rho}\frac{\partial p}{\partial x}+2\xi\frac{U^2}{D}=0 \qquad (14.15)$$

Energy Conservation

The first law of thermodynamics for a control volume states that the energy within the control volume changes due to heat and shear work transfers across the control volume surface and due to a net efflux of stagnation enthalpy resulting from flow across the control volume surface. The stagnation enthalpy h_0 is

$$h_0 = h + \frac{U^2}{2} = u + \frac{p}{\rho} + \frac{U^2}{2}$$

where u is the specific internal energy of the fluid (often approximated by $c_v T$). The shear work transfer across the control volume surface is zero. The heat-transfer rate $\dot{Q}_w$ is given by

$$\delta \dot{Q}_w = \dot{q} \rho A \, \delta x$$

where $\dot{q}$ is the heat transfer per unit mass of fluid per unit time into the control volume.

The rate of change of energy within the control volume is

$$\frac{\partial E}{\partial t} = \frac{\partial}{\partial t}\left[(\rho A \, \delta x)\left(u + \frac{U^2}{2}\right)\right]$$

The net efflux of stagnation enthalpy is

$$\frac{\partial}{\partial x}\left[(\rho U A)\left(u + \frac{p}{\rho} + \frac{U^2}{2}\right)\right]\delta x$$

Hence, the equation for energy conservation becomes

$$\frac{\partial}{\partial t}\left[(\rho A \, \delta x)\left(u + \frac{p}{\rho} + \frac{U^2}{2}\right)\right] + \frac{\partial}{\partial x}\left[(\rho U A)\left(u + \frac{p}{\rho} + \frac{U^2}{2}\right)\right]\delta x - \dot{q} \rho A \, \delta x = 0 \qquad (14.16)$$

Additional simplifications are possible. Expanding Eq. (14.16) and using the mass and momentum conservation equations yield

$$\frac{\partial u}{\partial t} + U\frac{\partial u}{\partial x} = \dot{q} + 2\xi \frac{U^3}{D} - \frac{p}{\rho A}\frac{\partial (UA)}{\partial x} \qquad (14.17)$$

If u can be represented by $c_v T$, and $R/c_v = \gamma - 1$ is constant, Eq. (14.17) can be rearranged and simplified to give

$$\frac{\partial p}{\partial t}+U\frac{\partial p}{\partial x}-a^2\left(\frac{\partial \rho}{\partial t}+U\frac{\partial \rho}{\partial x}\right)-(\gamma-1)\rho\left(\dot{q}+2\xi\frac{U^3}{D}\right)=0 \tag{14.18}$$

where the sound speed a for an ideal gas is given by

$$a^2=\left(\frac{\partial p}{\partial \rho}\right)_s=\gamma\frac{p}{\rho} \tag{14.19}$$

If friction and heat-transfer effects are small enough to be neglected, Eqs. (14.15) and (14.18) can be considerably simplified. In the absence of these effects the flow is *isentropic;* it has uniform entropy which is constant with time and is often called *homentropic* flow. If the duct area can be neglected then the continuity equation, (14.13), can be simplified also.

These one-dimensional unsteady flow equations have been used for a number of years to study the flow in the intake and exhaust systems of spark-ignition and diesel engines, both naturally-aspirated and turbocharged. Two types of methods have been used to solve these equations: (1) the method of characteristics and (2) finite difference procedures. The characteristic methods have a numerical accuracy that is first order in space and time, and require a large number of computational points if resolution of short-wavelength variations is important. Finite difference techniques are of higher order and more efficient: [8] this approach is now preferred. Methods for treating the boundary conditions will be described below.

Finite Difference Methods

Finite difference methods for solving the one-dimensional unsteady flow equations in intake and exhaust manifolds are more efficient and flexible than the method of characteristics. The conservation equations, (14.13), (14.14), and (14.16), can be rearranged and written in matrix form as

$$\frac{\partial}{\partial t}\begin{pmatrix}\rho\\ \rho U\\ \rho u\end{pmatrix}+\frac{\partial}{\partial x}\begin{pmatrix}\rho U\\ \rho U^2+p\\ p+\rho U u\end{pmatrix}=\begin{pmatrix}-\rho U\dfrac{dA}{dx}\\ -\rho\dfrac{U^2}{A}\dfrac{dA}{dx}-\rho\dfrac{2\xi U|U|}{D}\\ -\dfrac{4h_c(T-T_w)}{D\rho}-\dfrac{1}{A}\dfrac{dA}{dx}\left(\dfrac{1}{2}\rho U^3+\dfrac{\gamma}{\gamma-1}Up\right)\end{pmatrix} \tag{14.20}$$

The fluid viscous shear is small relative to friction at the wall in the momentum equation, and heat conduction and viscous dissipation prove negligible relative to convective heat transfer at the wall in the energy conservation equation. These equations have the vector form:

$$\frac{\partial F}{\partial t}+\frac{\partial G}{\partial x}=H \tag{14.21}$$

where G and H are functions of F only. Several finite difference methods have been used to solve Eq. (14.21) (e.g., see Refs. 7, 8, and 9). The one-step Lax-Wendroff method will be illustrated. [9] Equation (14.21) can be developed into a Taylor series with respect to time, and the time and space derivatives approximated by central differences around the mesh point, shown in Fig. 14.4, as

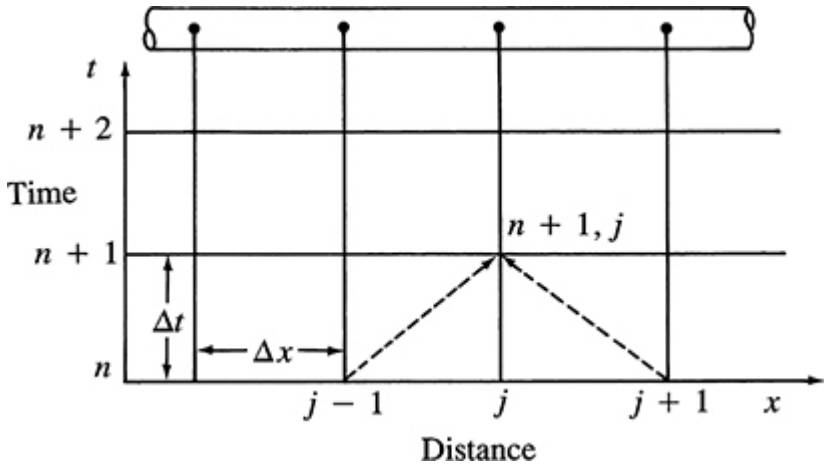


Figure 14.4 Mesh in time-distance plane for application of one-step Lax-Wendroff method to intake or exhaust pipe.

$$\begin{aligned}F_j^{n+1}&=F_j^n-\frac{1}{2}\frac{\Delta t}{\Delta x}(G_{j+1}^n-G_{j-1}^n)+\Delta t H_j^n\\&+\frac{1}{4}\left(\frac{\Delta t}{\Delta x}\right)^2[(G_{j+1}'^n+G_j'^n)(G_{j+1}^n-G_j^n)-(G_j'^n+G_j'^n)(G_j^n-G_{j-1}^n)]\end{aligned} \tag{14.22}$$

where $G' = \partial G/\partial F$. This equation is first-order accurate, unless H is small. For stability in the integration process, the time step and mesh size must satisfy the requirement that

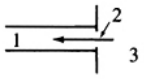
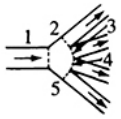
$$C = (|U| + a)\frac{\Delta t}{\Delta x} < 1 \tag{14.23}$$

where C is the Courant number.

These finite difference solution methods usually require the introduction of some dissipation or damping to prevent instabilities and large non-physical oscillations from occurring with nonlinear problems with steep gradients. Amplification of the physical viscosity and the addition of artificial viscosity, damping, and smoothing terms to Eq. (14.22) are frequently used techniques.

The boundary conditions at pipe ends and junctions are obtained from the appropriate conservation equations and pressure relations, as illustrated in Table 14.1. Out-flows and in-flows obviously conserve mass and energy. For the flow out through a restriction, there is no pressure recovery downstream: for flow in through a restriction, the flow upstream of the restriction is isentropic. For pipe junctions, the conservation equations are applied to the control volume (V) contained within the dashed lines in the sketch in the table. The pressure boundary conditions are most easily estimated by modifying the simple constant-pressure assumption with pressure losses at each pipe exit or entry, calculated from experimentally determined loss coefficients (see Fig. 6.7).

TABLE 14.1 Boundary conditions for unsteady one-dimensional finite element analysis [10]

Pipe ends			
Out-flow	Mass	$\rho_1 U_1 A_1 = \rho_2 U_2 A_2$	
	Energy	$c_p T_1 + \frac{U_1^2}{2} = c_p T_2 + \frac{U_2^2}{2}$	
	Isobaric	$p_2 = p_3$	
In-flow	Mass	$\rho_1 U_1 A_1 = \rho_2 U_2 A_2$	
	Energy	$c_p T_3 = c_p T_2 + \frac{U_2^2}{2} = c_p T_1 + \frac{U_1^2}{2}$	
	Isentropic	$p_2 / \rho_2^\gamma = p_3 / \rho_3^\gamma$	
Pipe junctions			
	Mass	$V \frac{\partial \rho}{\partial t} = \sum_i \rho_i U_i A_i$	
	Energy	$\rho V \frac{\partial u}{\partial t} = \sum_i (\rho_i U_i A_i) \left(c_p T_i + \frac{U_i^2}{2} \right)$	
	Pressure	$p_1 + \Delta p_1 = p_2 + \Delta p_2 = p_3 + \Delta p_3 = \ldots$ $\Delta p_i / p_i = C_i (U_i / a_i)^2$	

Calculations of intake and exhaust flows using these techniques predict the variations in intake and exhaust manifold pressure with crank angle (as shown, for example, in Fig. 6.10), in single and multi-cylinder engines, with acceptable accuracy. [8,10] Measured volumetric efficiency variations with engine speed, manifold design, and valve dimensions and timing are adequately predicted also. Figure 14.5 *a* shows the instantaneous exhaust and intake mass flow rates for cylinder number 1 of a four-cylinder spark-ignition engine at wide-open throttle at 1500 rev/min. Note how gas dynamic effects distort the exhaust flow. Note also the "reverse" flows into the cylinder past the exhaust valve and out of the cylinder past the intake valve at the end of the exhaust process, and the larger reverse flow at the end of the intake process at this low engine speed. Figure 14. *5b* shows the volumetric efficiency for this engine based on these predicted mass flow rates, as a function of speed. Experimental values and values predicted with quasi-steady flow equations and infinite plenums for manifolds are also shown. These results clearly demonstrate the important role that intake and exhaust system gas dynamics play in determining both the engine speed at which peak breathing efficiency occurs and the air charging characteristics over the full engine speed range. [8]

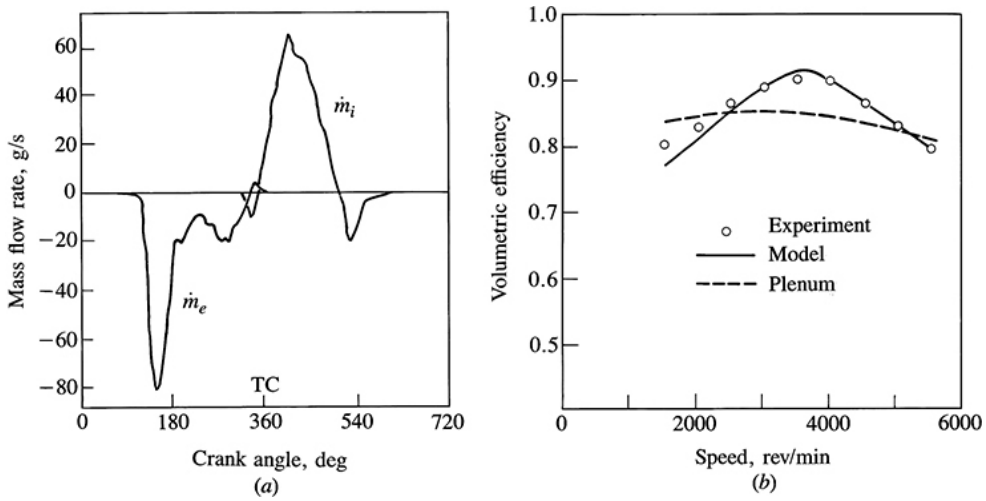


Figure 14.5 (*a*) Predicted mass flow rate through the exhaust valve $\dot{m}_j$ and through the intake valve $\dot{m}_i$ in cylinder 1; four-cylinder four-stroke cycle spark-ignition engine at wide-open throttle and 1500 rev/min. Flows into cylinder are positive; flows out are negative. (*b*) Predicted and measured volumetric efficiency at wide-open throttle for four-cylinder spark-ignition engine. Solid line: one-dimensional unsteady flow model. Dashed line: quasi-steady flow calculation based on infinite plenums for manifolds. [8]

14.4 THERMODYNAMIC-BASED IN-CYLINDER MODELS

14.4.1 Background and Overall Model Structure

If the mass transfer into and out of the cylinder during intake and exhaust, the heat transfer between the in-cylinder gases and the cylinder head, piston, and cylinder liner, and the rate of charge burning (or rate of energy release from the fuel) are all known, the energy and mass conservation equations permit the cylinder pressure and the work transfer to the piston to be calculated. Engine models of this type have been developed and used extensively to predict engine operating characteristics (indicated power, mean effective pressure, specific fuel consumption, etc.) and to define the gas state for emission calculations. These models effectively follow the changing thermodynamic and chemical state of the working fluid through the engine's

intake, compression, combustion, expansion, and exhaust processes. Then, once the indicated performance of the engine has been computed, friction work or losses are deducted. The pumping component is included in the intake and exhaust process analysis: so the needed phenomenological model here is of mechanical (rubbing) and accessary friction. The total model (as outlined in Fig. 14.6) is often called an engine *cycle simulation.*

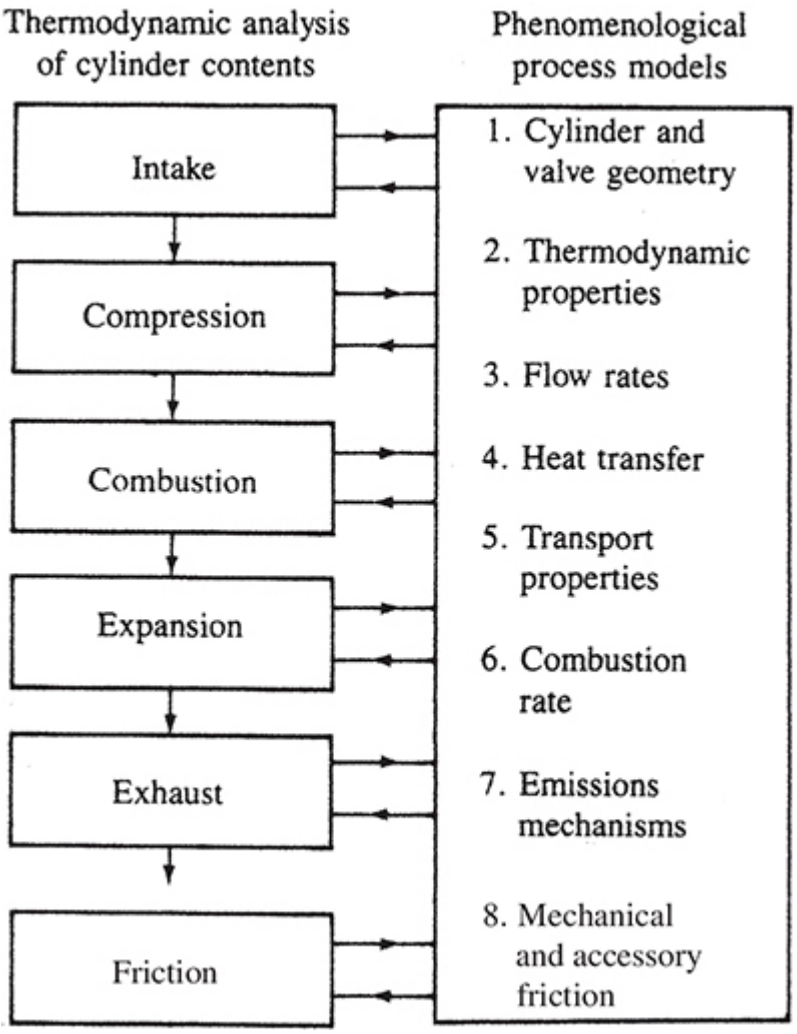


Figure 14.6 Logic structure of thermodynamic-based simulation of internal combustion engine operating cycle.

The starting point for these cycle simulations is the first law of thermodynamics for an open system, developed in Sec. 14.2 . This is applied to the cylinder volume for the intake, compression, combustion, expansion, and exhaust processes that in sequence make up the engine's operating cycle.

The structure of this type of engine simulation is indicated in Fig. 14.6. Then, during each process, submodels are used to describe geometric features of the cylinder and valves or ports, the thermodynamic properties of the unburned and burned gases, the mass and energy transfers across the system boundary, and the combustion process.

During intake and compression, the cylinder volume is modeled as a single open system. Application of the conservation equations in the form of Eqs. (14.1) , (14.3), and (14.10) or (14.11) for the intake and then the compression process gives[2]

Intake:

$$\dot{m} = \dot{m}_i - \dot{m}_e \tag{14.24}$$

$$\dot{f} = \frac{\dot{m}_i}{m}(f_i - f) - \frac{\dot{m}_e}{m}(f_e - f) \tag{14.25}$$

$$\dot{T} = \frac{B}{A}\left[\frac{\dot{m}}{m}\left(1 - \frac{h}{B}\right) - \frac{\dot{V}}{V} - \frac{C}{B}\dot{\phi} + \frac{1}{Bm}(\dot{m}_i h_i - \dot{m}_e h_e - \dot{Q}_w)\right] \tag{14.26}$$

where m is the mass of gas in the cylinder, $\dot{m}_i$ and $\dot{m}_e$ are the mass flow rates through the intake valve and the exhaust valve, and f is the fuel fraction m_f / m. The subscripts i and e denote properties of the flow through the intake and exhaust valves, respectively. The thermodynamic properties for these flows are the values upstream of the valves and therefore depend on whether the flow is into or out of the cylinder.

Compression:

$$\dot{m} = 0 \quad \dot{f} = 0 \tag{14.27a,b}$$

$$\dot{T} = \frac{B}{A}\left(-\frac{\dot{V}}{V} - \frac{\dot{Q}_w}{Bm}\right) \tag{14.28}$$

The pressure is then determined from Eq. (14.9) .

During intake and compression, the working fluid composition is frozen. The composition and thermodynamic properties can be determined using the models described in Secs. 4.2 and 4.7. Mass flows through open valves are

usually calculated using one-dimensional compressible flow equations for flow through a restriction (see App. C and Secs. 6.3.2 and 14.3.2) or filling and emptying models (Sec. 14.3.3). The more accurate unsteady gas-dynamic intake (and exhaust) flow models described in Sec. 14.3.4 can be used to calculate the mass flow into the engine cylinder in complete engine cycle simulations when the variation in engine flow rate with speed is especially important: the disadvantage is increased computing time. Heat transfer during intake and compression is calculated using one of the Nusselt-Reynolds number relations for turbulent convective heat transfer described in Sec. 12.4.5 . The transport properties, viscosity, and thermal conductivity used in these correlations can be obtained from relations such as Eqs. (4.52) to (4.55).

During combustion which starts with the spark discharge in spark-ignition engines and with spontaneous ignition of the developing fuel-air jets in compression-ignition engines, the actual processes to be modeled become more complex. Many approaches to predicting the burning or chemical energy release rate have been used successfully to meet different simulation objectives. The simplest approach has been to use a one-zone model where a single thermodynamic system represents the entire combustion chamber contents and the energy release rate is defined by empirically based functions specified as part of the simulation input. Or, in spark-ignition engines, quasi-geometric models of turbulent premixed flames are used with a two-zone analysis of the combustion chamber contents—an unburned and a burned gas region. In compression-ignition engines, multiple-zone models of the developing fuel-air jets have been used to provide more detailed predictions of the combustion process and the nonuniform cylinder composition and state. These combustion models will be reviewed in the following sections (14.4.2 and 14.4.3) and the appropriate conservation equations for the *combustion process* will be developed there. In diesels, radiation heat transfer becomes important during the combustion process (see Sec. 12.6.3).

The *expansion process* is either treated as a continuation of the combustion process or, once combustion is over, can use the form of the mass, fuel, and energy conservation equations which hold during compression [Eqs. (14.27) and (14.28)]. The exhaust process conservation equations for a one-zone open-system model of the cylinder contents are [2]

Exhaust:

$$\dot{m} = -\dot{m}_e \quad \dot{f} = -\frac{\dot{m}_e}{m}(f_e - f) \tag{14.29a,b}$$

$$\dot{T} = \frac{B}{A}\left[-\frac{\dot{m}_e}{m}\left(1-\frac{h}{B}\right) - \frac{\dot{V}}{V} - \frac{C}{B}\dot{\phi} + \frac{1}{Bm}(-\dot{m}_e h_e - \dot{Q}_w)\right] \tag{14.30}$$

where h_e, the enthalpy of the flow through the exhaust valve, is the cylinder average enthalpy at that point in time for flow *out* of the cylinder, or the exhaust system gas enthalpy if reverse flow back into the cylinder occurs.

The engine operating cycle should end with the working fluid at the same state that it started out in. For the first calculations of the sequence of processes in Fig. 14.6, property values defining the initial state of the fluid in the cylinder are assumed. If the values of these properties at the end of the first cycle differ from the assumed values, the cycle calculation is repeated with the appropriate new initial values until the discrepancy is sufficiently small. Convergence with these cycle simulations occurs within a few iterations.

The working fluid state is now defined throughout the operating cycle. The work transfer to the piston per cycle

$$W_c = \oint p dV \tag{14.31}$$

can now be obtained. From W_c, the masses of fuel and air inducted, m_f and m_a, and engine speed N, all the engine *indicated* performance parameters can be calculated: power, mean effective pressure, specific fuel consumption, fuel-conversion efficiency; as well as volumetric efficiency, residual gas fraction, total heat transfer, etc. With a friction model, the indicated quantities can be converted to brake quantities (including torque).

The more sophisticated of these thermodynamic-based engine cycle simulations define the working fluid state throughout the cycle in sufficient detail for predictions of engine emissions to be made. The discussion in Chap. 11 of emission-formation mechanisms indicates that our understanding of how some of these pollutants form (e.g., NO_x, CO) is reasonably complete, and thus can be modeled. The formation processes of the other pollutants (unburned hydrocarbons and particulates) are not yet adequately understood, though modeling activities continue to contribute to that

understanding. The key features of models for predicting engine emissions are discussed in Chap. 11 .

Cycle simulations and combustion models which have been developed for spark-ignition engines, where the fuel, air, residual gas mixture is essentially uniform, are discussed in Sec. 14.4.2 . Compression-ignition engine simulations and their combustion models are then discussed in Sec. 14.4.3 . The special features required for prechamber engine models are reviewed in Sec. 14.4.4 . Finally, thermodynamic-based models for more complex engine systems—multi-cylinder, turbocharged, and turbocompounded engines—are discussed in Sec. 14.4.5 .

14.4.2 Spark-Ignition Engine Models

These models have usually followed the conceptual structure indicated in Fig. 14.6. We focus here on the combustion submodels that have been developed and used successfully. Features of the spark-ignition engine combustion process that permit major simplifying assumptions for thermodynamic modeling are: (1) the fuel, air, residual-gas charge is essentially uniformly premixed; (2) the volume occupied by the reaction zone where the fuel-air oxidation process actually occurs is small compared with the clearance volume—the flame is a thin reaction sheet even though it becomes highly wrinkled and convoluted by the turbulent flow as it develops (see Sec. 9.3); and (3) for thermodynamic analysis, the contents of the combustion chamber during combustion can be analyzed as two zones—an unburned and a burned zone.

Useful combustion chamber design information can be generated with simple geometric models of the flame. In the absence of strong swirl, the surface which defines the flame location can be approximated by a portion of the surface of a sphere. The mean burned gas front can similarly be approximated. Thus, for a given combustion chamber shape and assumed flame center location (e.g., the spark plug electrodes), the spherical burning area A_b [see Eq. (9.43)], the burned gas volume V_b [see Eq. (9.42)], and the combustion chamber surface "wetted" by the burned gases can be calculated for a given flame radius r_b and piston position (defined by crank angle) from purely geometric considerations.[b] The practical importance of such "model" calculations is that (1) the mass burning rate for a given burning speed S_b

(which depends on local turbulence and mixture composition) is proportional to the spherical burning area A_b as given by Eq. (9.47) ; (2) the heat transfer occurs largely between the burned gases and the walls and is proportional to the chamber surface area wetted by the burned gases $A_{b,w}$ (see Fig. 12.21). Using the fact that the density ratio across the flame ρ_u/ρ_b is approximately constant and equals about 4, the unburned and burned gas volumes V_u and V_b can be related to the unburned and burned mass fractions, $(1 - x_b)$ and x_b, respectively.

Examples of the results of such flame geometry calculations are shown in Figs. 14.7 and 14.8. [11] Figure 14.7*a* shows spherical flame areas A_b as a function of flame radius r_b for two different chambers and two plug locations and the TC piston position. The much larger flame area and shorter flame travel length of the central plug location are obvious. Such area data can be plotted as a function of burned gas volume V_b, as shown in Fig. 14.7 *b*, so that comparisons of $A_b(\ r_b)$ for different chambers at the same mass fraction burned can be made. The advantage of a more compact chamber with higher central clearance height is apparent. Figure 14.8 shows burned-gas-wetted wall area on the cylinder head, cylinder wall, and piston as a function of flame radius and crank angle for an open chamber with central ignition. The cylinder head and piston are the dominant areas early in the expansion stroke when the burned gas temperatures and heat fluxes are highest.

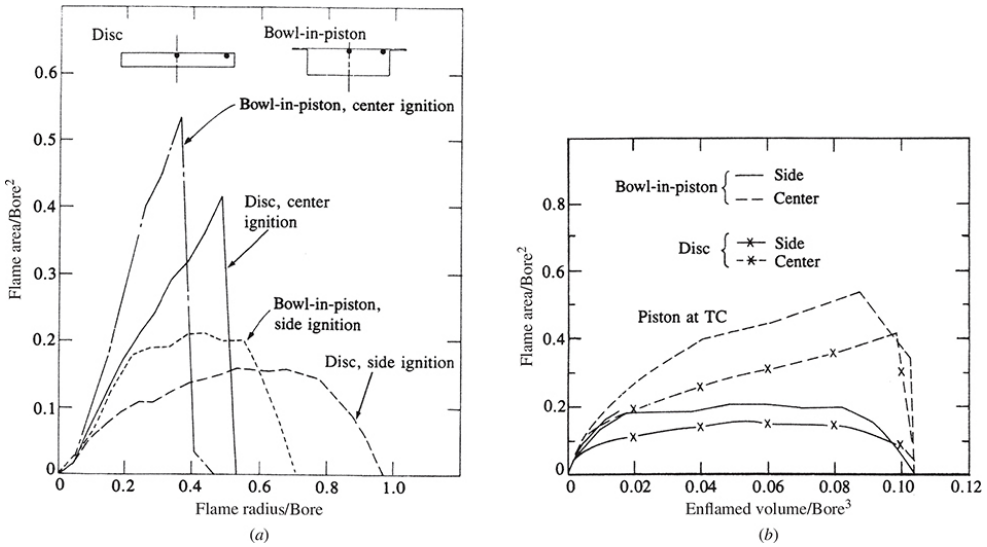


Figure 14.7 Calculated spark-ignition engine spherical flame surface area, scaled by cylinder bore squared: (*a*) as a function of flame radius for different combustion chamber shapes and spark plug locations; (*b*) as a function of enflamed volume, scaled by bore cubed. Piston in top center position. [11]

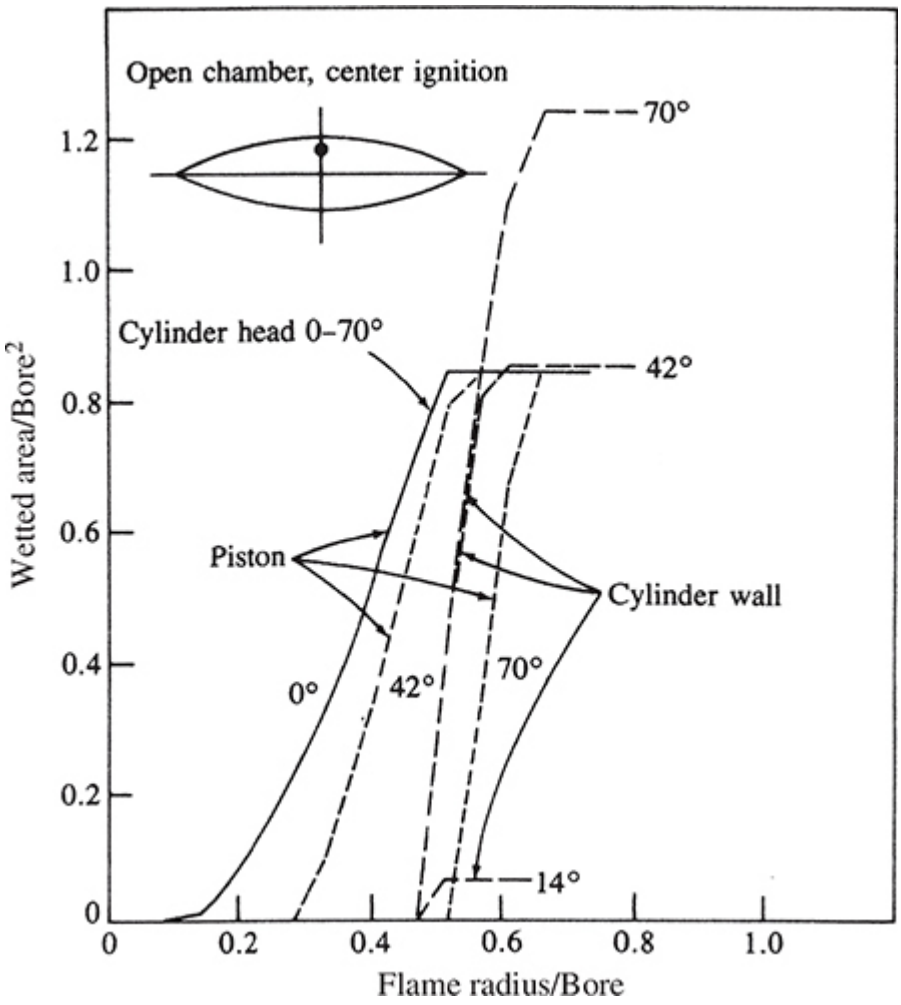


Figure 14.8 Calculated burned-gas-wetted wall area as a function of radius based on spherical flame model of an open-chamber SI engine with center plug location normalized by bore squared: for piston locations of 0°, 42°, and 7°. [11]

Mass fraction burned versus crank-angle profiles determined from a first-law analysis of cylinder pressure data, as shown in Figs. 9.4, 9.8, and 9.10 have an essentially universal dimensionless shape, as indicated in Fig. 9.15.

Much useful analysis has been done with engine simulations where this universal combustion profile has been used as a calculation input. The S-shaped mass fraction burned profile is often represented by the Wiebe function:

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{m+1}\right] \tag{14.32}$$

where θ is the crank angle, θ_0 is the start of combustion, $\Delta\theta$ is the total combustion duration ($x_b = 0$ to $x_b \approx 1$), and a and m are adjustable parameters which fix the shape of the curve. Actual mass fraction burned curves have been fitted with $a = 5$ and $m = 2$. [12]

The conservation equations for an open system [Eqs. (14.1) and (14.10) or (14.11)] are now applied to the unburned gas zone ahead of the flame and to the burned zone behind the flame, in turn (see Fig. 9.4). For premixed engines, $\dot{f}$ and $\dot{\phi}$ are zero. During combustion, $\dot{m}$ and $\dot{m}_j$ in Eq. (14.10) or Eq. (14.11) are the mass flow rate across the flame sheet. This is $-\dot{m}_b$ for the unburned zone system and $+\dot{m}_b$ for the burned zone system; $\dot{m}_b$ is given by $m\,\dot{x}_b$, with $\dot{x}_b$ obtained by differentiating Eq. (14.32) .

To calculate the effect of heat transfer on the burned gas state more accurately, the burned gas zone in Fig. 9.6 can be modeled in two parts: an adiabatic core and a boundary-layer region. The intent here is to account for the fact that heat loss to the walls primarily cools the burned gas adjacent to the wall, and only indirectly affects the core gas through the change in pressure that results from heat loss. The open-system conservation equations, (14.1) and (14.10) or (14.11), are now applied to the core and boundary-layer region separately. The boundary-layer region covers that portion of the combustion chamber wall wetted by the burned gases, as shown in Fig. 12.21, and is of thickness δT, which increases with time. The temperature of the boundary-layer zone (assumed uniform) is usually taken to be the mean of the wall temperature and burned gas core temperature. Equation (14.10) or Eq. (14.11) is used to relate the enthalpy flux due to the mass flow across the edge of the boundary layer (which has an enthalpy equal to the core-gas enthalpy), the heat transfer to the wall, the changing energy within the

boundary-layer system due to its increasing mass and changing state, and the work transfer due to its changing volume.

An example of predictions of cylinder pressure, unburned and burned gas temperatures, heat-transfer rate, and boundary-layer thickness, based on an assumed 50° total burn duration for a 5.7-liter eight-cylinder engine at wide-open throttle and 2500 rev/min, is shown in Fig. 14.9. Appropriately based predictions of overall engine performance parameters made with this type of thermodynamic model agree well with engine data. Figure 14.10 shows predictions of indicated specific fuel consumption and exhaust gas temperature as a function of the fuel/air equivalence ratio at fixed load and speed. The isfc predictions and data agree well (except for very lean mixtures with retarded timing where cycle-by-cycle combustion variations are sufficiently large so predictions based on the average cycle lose accuracy); the predicted curves for exhaust temperature show the same trends as the experimental data. However, they are higher due to underestimation of the heat losses during the exhaust process. [13]

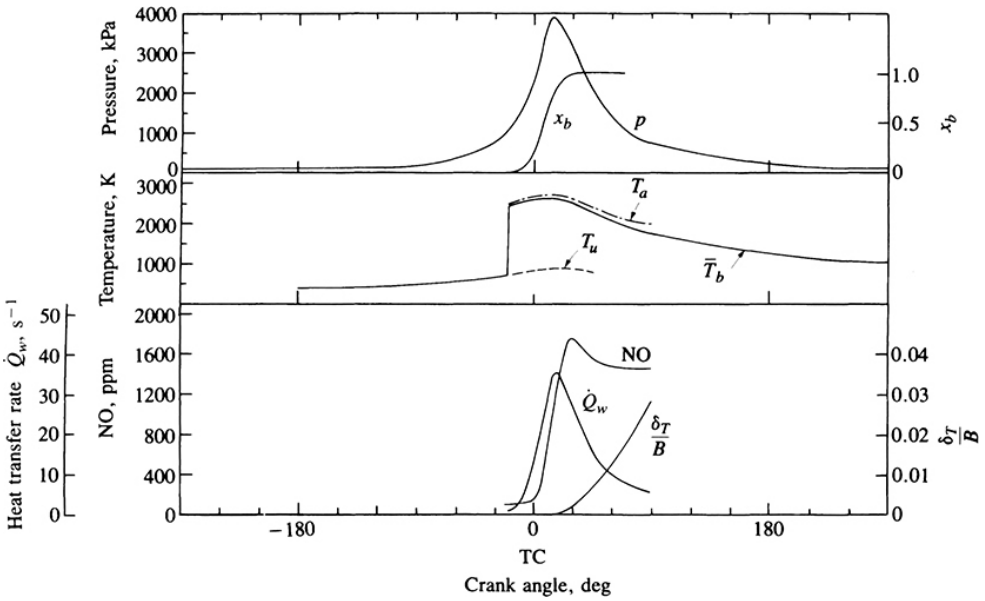


Figure 14.9 Cylinder pressure p, mass fraction burned x_b, unburned and burned gas temperatures (T_u = unburned, T_a = adiabatic burned core, $\bar{T}_b$ = mean burned gas temperatures), heat-transfer rate $\dot{Q}_w$ (normalized by fuel flow rate × heating value), thermal boundary-layer thickness d_T, and mean nitric oxide concentration in the burned gases, through the four-stroke SI engine operating cycle, predicted by a thermodynamic-based cycle

simulation. 5.7-liter displacement eight-cylinder engine operating at wide-open throttle, 2500 rev/min, with fuel/air equivalence ratio = 1.1. Gross indicated mean effective pressure is 918 kPa and specific fuel consumption is 254 g/kW · h. [14]

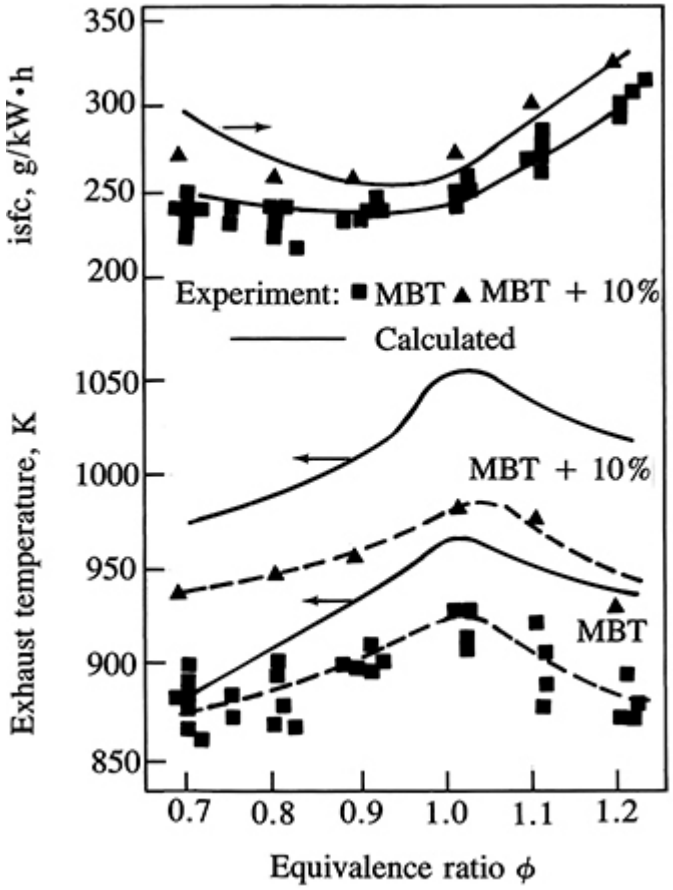


Figure 14.10 Predicted and measured indicated specific fuel consumption and exhaust temperature as a function of the fuel/air equivalence ratio for a spark-ignition engine operated at 1250 rev/min and imep of 379 kPa. MBT, maximum brake torque timing. MBT + 10%: combustion timing was retarded to give a 10% fuel consumption penalty relative to MBT timing. [13]

The output from such thermodynamic-based cycle simulations has replaced the fuel-air cycle as a predictor of effects of major variables on engine performance and efficiency. An instructive example of the value of such predictions is shown in Fig. 14.11, where fuel consumption at constant fuel/air equivalence ratio, load, and speed has been computed as a function

of total burn duration and heat loss to the chamber walls: increasing burn duration and heat loss both worsen fuel consumption. [15] Such data can be used to assess the efficiency improvements that should result from reduced heat transfer (e.g., reduced chamber surface area) and increased burn rate. With 0–100% burned crank angle duration of about 60° (corresponding to $\Delta\theta_{0\text{-}90\%}$ of about 50°), the impact of heat losses on bsfc is about one-third: a 10% reduction (relative) in heat loss per cycle results in a 3 to 3.5% reduction in bsfc at about one-quarter the maximum load. Obviously the dependence of burn rate on engine design and operating parameters has not been modeled; the burn rate profile was a calculation input. Such models are most useful either: (1) when the burn rate profile is *not critical* to the problem under study; or (2) when predictions for a *range* of assumed burn rate profiles provide the required information.

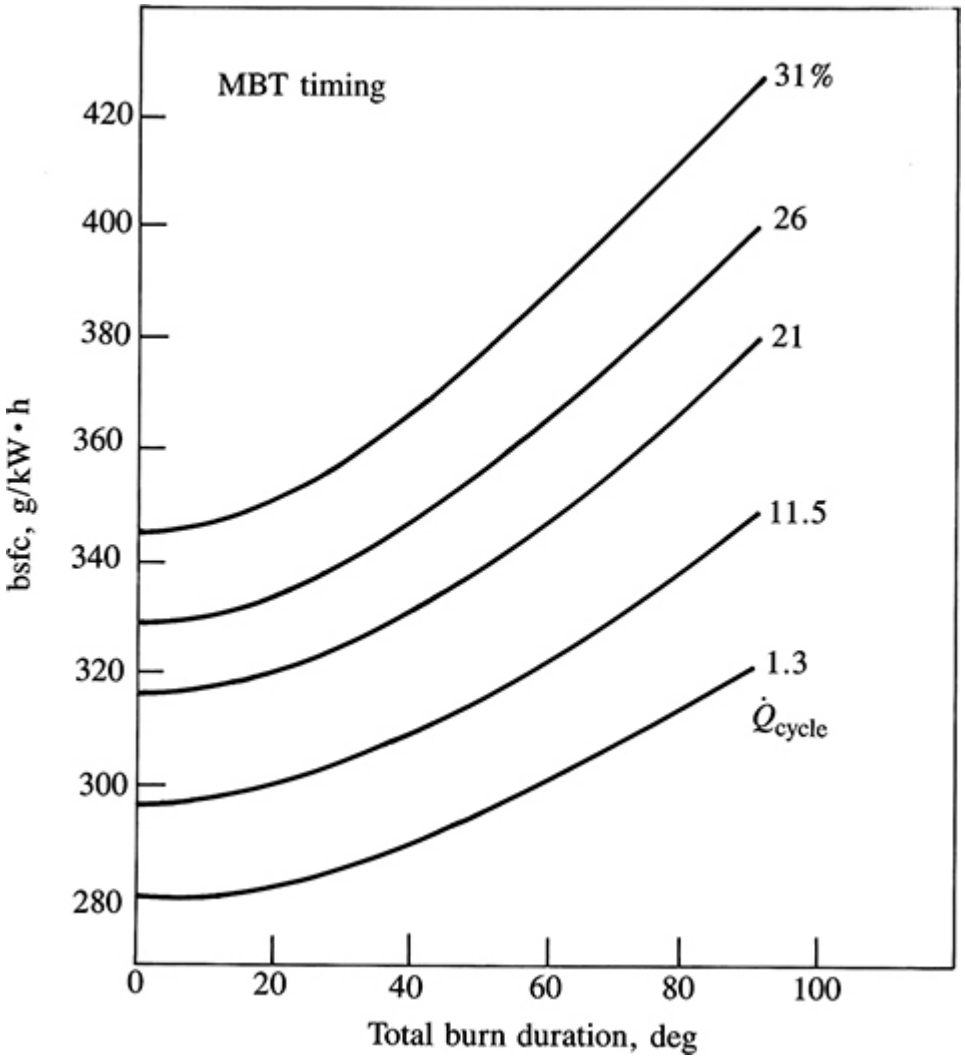


Figure 14.11 Predicted brake specific fuel consumption as a function of heat transfer per cycle to the combustion chamber walls (as a percent of the fuel's heating value) and total burn duration [$\Delta\theta$ in Eq. (14.32)]. 1250 rev/min, 262 kPa bmep, relative air/fuel ratio = 1.1, maximum brake torque spark timing. [15]

So far we have discussed engine cycle simulations where details of the combustion process have been specified as input. The same thermodynamic-based simulation structure can be used in conjunction with a combustion model which *predicts* the rate of fuel burning. Various combustion models have been proposed and used for this purpose. Some of these are empirically based. The more useful are based on the wrinkled thin reaction-sheet flame model described in Sec. 9.3 . These models assume that the overall flame shape approximates a portion of a sphere centered at or near the spark plug. It is important to describe the three phases of the combustion process—flame development, rapid burning, and termination—from the basic flow and chemistry principles. The critical flow feature is the level of in-cylinder turbulence (characterized by the local rms velocity fluctuation u' during combustion). The flame chemistry dependence on fuel-air mixture composition can be incorporated appropriately through the mixture's laminar flame speed.

Spark-ignition engine combustion models with such a fundamental framework have been developed and used extensively. Based on coupled analysis of flame front location and cylinder pressure data, Keck and coworkers [16–18] have derived the following entrainment-based burning law:

Rate of entrainment (laminar diffusion plus turbulent entrainment into the enflamed region):

$$\frac{dm_e}{dt} = \rho_u A_f S_L + \rho_u A_f u_T (1 - e^{-t/\tau_b}) \tag{14.33}$$

Rate of mixture burning (laminar flame propagation plus burn up of entrained yet-to-burn mass):

$$\frac{dm_b}{dt} = \rho_u A_f S_L + \frac{m_e - m_b}{\tau_b} \tag{14.34}$$

where A_f is the flame envelope area (portion of a sphere), S_L is the local laminar flame speed, u_T is the turbulent entrainment speed ($u_T \approx u'$), and $\tau_b = l_T/SL$ is a characteristic burning time, where l_T is the characteristic length scale of the turbulent fluid motion. See Sec. 9.3.4 .

Figure 14.12 illustrates the physical basis for this model. The first term in Eq. (14.33) , the entrainment equation, represents the laminar diffusion of mixture across the approximately spherical front of the "thick" turbulent flame; the second term represents the entrainment of mixture into the flame due to the turbulent motion. In Eq. (14.34) , which describes the rate of burning of the *unburned* mixture, the first term represents the burning due to the propagating laminar flame, and the second term the burn-up of unburned mixture within the turbulent flame "brush." This unburned mixture is the difference between the mass already entrained and the amount already burned, contained within the "wrinkles" and "islands" which the distorting and stretching of the thin reaction sheet by the turbulent flow produce. This is often called an "entrainment" or "eddy-burning "model for the above reasons. The exponential term in brackets in Eq. (14.33) allows for the fact that the flame sheet initially is spherical and "laminar-like": it requires a time of about τ_b to develop into a turbulent flame.[c]

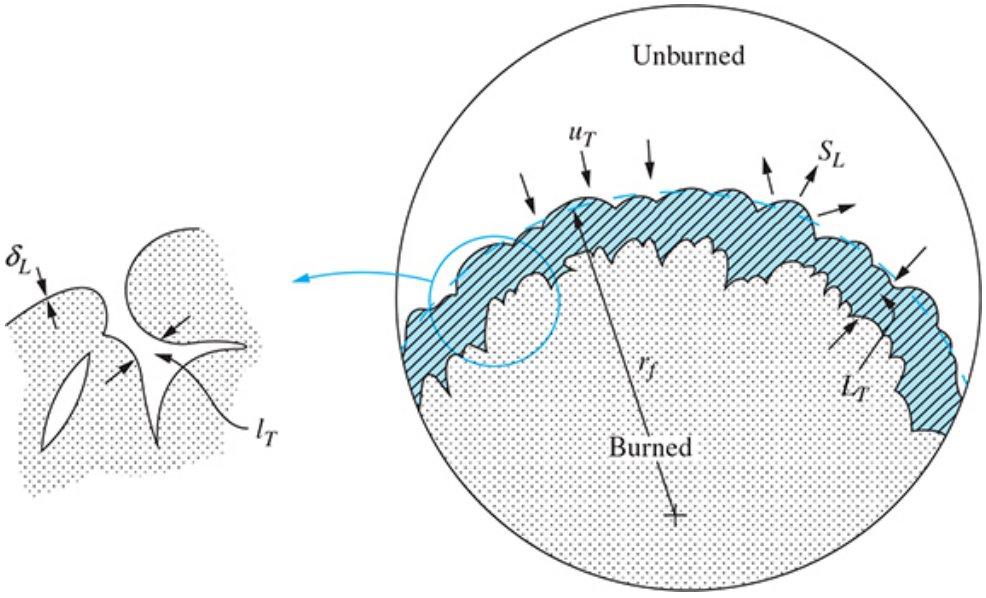


Figure 14.12 Schematic of turbulent premixed spark-ignition engine flame illustrating the physical basis for the burning law of Eqs. (14.33) and (14.34). The approximately spherical front of the "thick" turbulent flame

(dashed line) diffuses outward at the laminar flame speed S_L, and fresh mixture crosses this flame front at a characteristic velocity u_T due to turbulent convection. Schematic on left shows detailed flame structure: δL is the reaction-sheet thickness and l_T is the characteristic scale of wrinkles in the flame sheet.

The behavior of Eqs. (14.33) and (14.34) in four important limits is as follows:

1. For a quiescent mixture, $u_T \to 0$ or $l_T \to \infty$

$$S_b \to S_L \tag{14.35a}$$

2. Initially, as $t \to 0$,

$$S_b \to S_L \tag{14.35b}$$

3. Quasi-steady state, *dmb*/ *dt* equals *dme*/ *dt*,

$$S_b \approx u_T + S_L \tag{14.35c}$$

4. Final burning stage after the flame front reaches the wall, $t \geq t_w$ (when $\boldsymbol{A} f \to 0$) ,

$$\dot{m}_b / \dot{m}_b(t_w) = e^{-(t-t_w)/\tau_b} \tag{14.35d}$$

To apply Eqs. (14.33) and (14.34), the quantities u_T and τ_b (or $l_T = \tau_{bS} L$) must be evaluated. Two approaches have been taken: (1) use of empirical correlations for these variables, derived from engine flame data (such as that described in Sec. 9.3.4); (2) use of more fundamental models to predict these quantities.

Keck has derived the following correlations for u_T and l_T, based on the application of Eqs. (14.33) and (14.34) to several sets of engine combustion data:

$$u_T = 0.08\bar{u}_i \left(\frac{\rho_u}{\rho_i}\right)^{1/2} \tag{14.36}$$

$$l_T = 0.8L_{iv} \left(\frac{\rho_i}{\rho_u}\right)^{3/4} \tag{14.37}$$

u_T was found to be proportional to $\sqrt{\rho_u}$ (at time of spark) and to correlate well with mean inlet gas speed $\bar{u}_i = \eta_v (A_p / A_{iv}) \bar{S}_p$, where η_v is the volumetric efficiency, A_p is piston area, A_{iv} is the maximum open area of the inlet valve, $\bar{S}_p$ is mean piston speed. l_T appears to scale with valve lift, L_{iv}; it decreases with increasing density at a rate proportional to $\rho u^{-3/4}$. While u_T and l_T are not constant during the combustion process, their variation is modest. [16]

A quantitative comparison of predicted and measured flame radius as a function of time is shown in Fig. 14.13 for propane and hydrogen fuel-air mixtures, which exhibit widely different behavior: [16] the figure indicates both the behavior and validity of the model. Predicted burned gas expansion speeds u_b [see Eq. (9.46)] are shown in Fig. 14.13*a* as a function of burned gas radius; the parameters u_T and l_T were chosen to fit the propane data. Figure 14.13 *b* shows that the measured flame front radii, r_f, are in good agreement with the predicted flame and burned gas radii, r_f and r_b, for these two fuels. The initial expansion speed of hydrogen is about 10 times that of propane. Since $r_f \approx r_b$ for early times, and $S_b \approx S_L$, this ratio is expected. As r_b becomes large, ($r_f - rb$) $\rightarrow uT\,\tau b$, which is several times smaller for hydrogen-air mixtures than for propane-air mixtures.

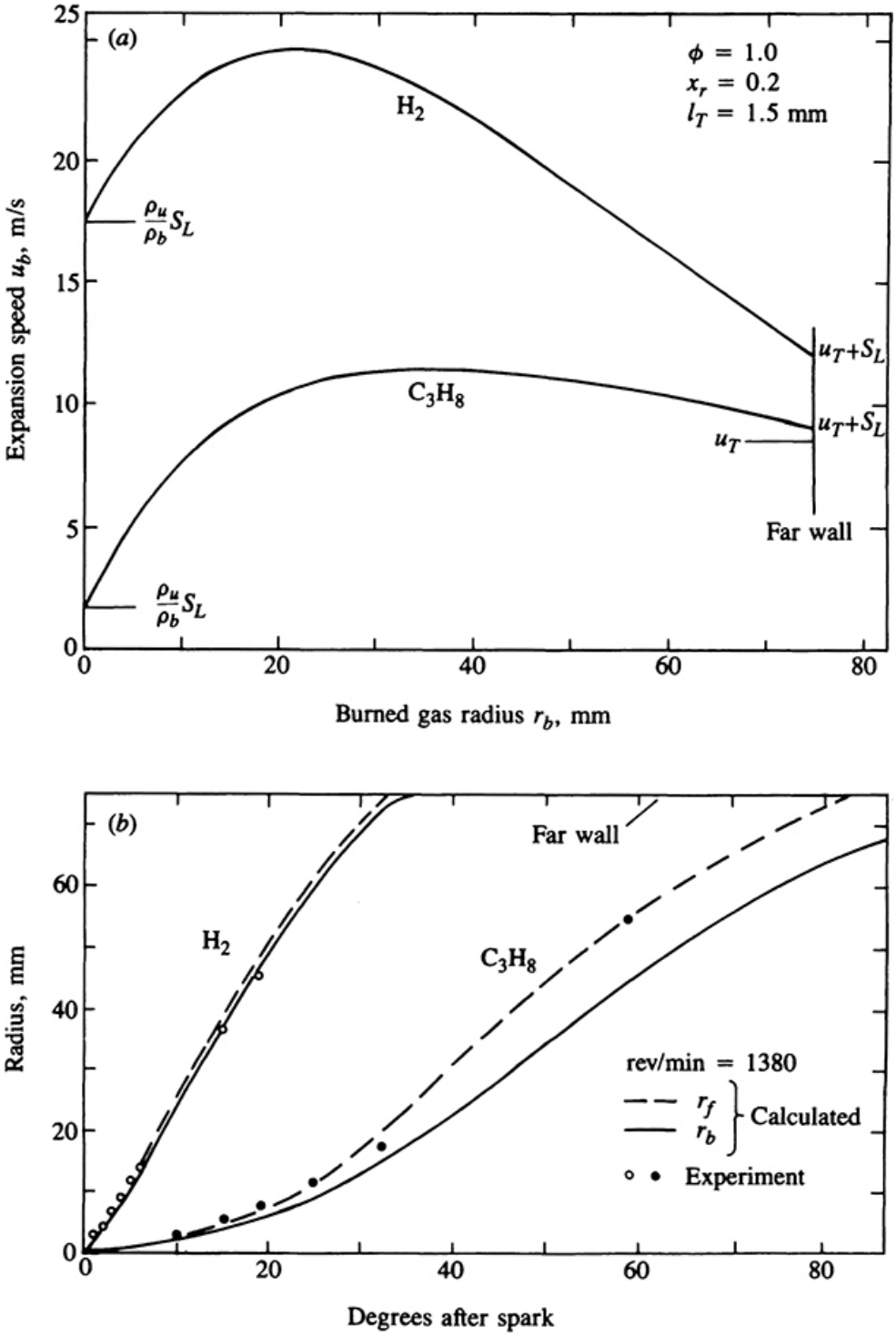


Figure 14.13 (a) Calculated burned gas expansion speed u_b for stoichiometric hydrogen-air and propane-air mixtures as a function of burned gas radius r_b. (b) Comparison of experimentally measured (points) and

calculated (dashed curve) flame radii r_f for these mixtures as a function of crank angle. Also shown (solid curve) is the burned gas radius r_b. [16]

An adaption of this approach developed by Tabaczynski and coworkers [20, 21] is based on the following model of turbulent flame propagation. The vorticity in the turbulent flow field is concentrated in vortex sheets which are of a size comparable to the Kolmogorov scale l_K [see Eq. (8.11)]. These vortex sheets are assumed to have a characteristic spacing which is of the order of the Taylor microscale l_M, which is a function of the integral length scale l_I and the turbulent Reynolds number as indicated by Eq. (8.15) . From these turbulence assumptions, it is argued that ignition sites propagate along the vortex sheets with a velocity $u' + S_L$, where u' is the local turbulence intensity. The propagation of the reaction front between the vortex sheets is assumed to be a laminar process. Thus, in Eqs. (14.33) and (14.34), u_T and τb are given by

$$u_T \approx u' \quad \text{and} \quad \tau_b = \frac{l_T}{S_L} \approx \frac{l_M}{S_L} \tag{14.38}$$

where l_M, the microscale, is determined from the integral scale and the turbulent Reynolds number via Eq. (8.15) , assuming that the turbulence is homogeneous and isentropic. The task therefore becomes one of evaluating u' and l_I.

One approach used is to relate the turbulence intensity at the start of the combustion process to the mean intake flow velocity through the valve: for example, [22]

$$u_0' = \frac{C\bar{S}_p B^2}{L_{\text{iv}} D_{\text{iv}}} \tag{14.39}$$

where $\bar{S}_p$ is the mean piston speed, B the bore, and L_{iv} and D_{iv} the lift and diameter of the intake valve. It is assumed that the integral length scale at the start of combustion, $l_{I,0}$, is proportional to a characteristic flow dimension, usually the clearance height h. Then, during combustion, the unburned portion

of the charge is assumed to undergo isentropic compression sufficiently rapidly that the angular momentum of the "eddies" is conserved and the length scale follows the eddy size: that is, a simple rapid-distortion process occurs:

$$u' = u'_0\left(\frac{\rho}{\rho_0}\right)^{1/3} \quad l_I = l_{I,0}\left(\frac{\rho_0}{\rho}\right)^{1/3} \tag{14.40}$$

This model predicts an increase in turbulence intensity and decrease in length scale with compression, which is only partly confirmed by experiment.

A more sophisticated approach is to describe the dynamic behavior of the turbulence with one or more rate equations for the key turbulence parameters: k the turbulent kinetic energy and e the dissipation rate of k. Turbulence is generated, diffused, and dissipated by the flow field, so the rate of change of turbulent kinetic energy k can be written:

$$\frac{dk}{dt} = P_k + D_k - \rho\varepsilon \tag{14.41}$$

where the term P_k represents the volumetric production of turbulence and the diffusion term D_k can be modeled as a gradient diffusion with an effective turbulent viscosity which dominates the laminar diffusion process. In this application, Eq. (14.41) is integrated over the combustion chamber (or a region of the chamber) to provide spatially averaged turbulence predictions. Then the diffusion terms become boundary fluxes: for example, the transport of kinetic energy across the combustion chamber boundary due to flow through the inlet or exhaust valve. The dissipation rate e is related to the integral length scale via

$$\varepsilon = C_D k^{3/2}/l_I, C_D = 0.09 \tag{14.42}$$

l_I can be taken as proportional to the clearance height ($l_I \approx 0.22\ h$), or an additional rate equation for the second turbulence parameter, the dissipation rate ε, can be used. In the more complete of these k — ε turbulence models, [23] the ε equation is similar to the k equation with production, diffusion, and

dissipation terms. These k — ε turbulence models are discussed more fully in Sec. 14.5.2 .

This turbulence model has been incorporated into spark-ignition engine combustion models: these references give the details. [24–27] Considerable success with predicting trends in mass burning rate has been achieved with this type of model. Design variables examined include: swirl, squish, valve lift, bore/stroke ratio. The advantage of such models is that they are straightforward computationally so that extensive parametric sets of calculations are feasible. Their disadvantage is the *ad hoc* nature of the turbulence and flame models which involve plausible but somewhat arbitrary assumptions. Sample predictions are shown in Fig. 14.14 which shows the variation in turbulence intensity u' in an engine with a disc-shaped combustion chamber, throughout the operating cycle. A normal valve-lift profile and reduced maximum valve-lift profile (one-quarter normal) are shown. The high levels of turbulence generated during the first half of the intake process decay substantially before the latter stages of the compression stroke produce some amplification. Reduced valve lift produces higher levels of turbulence intensity at combustion, as is well known. Predicted turbulence behavior during combustion indicates a moderate (some 10%) rise and fall. Swirl is shown to increase the turbulence intensity: low to high swirl increases u' by about 25%. Comparison of predicted and measured mass fraction burned profiles versus crank angle for different swirl levels and plug locations are shown in Fig. 14.15. The large flame area effects (shown here in the two limiting plug locations: side wall and center) and significant though lesser effect of swirl are correctly modeled. Such models are useful for relating changes in spark-ignition engine design and operating variables to changes in engine performance, via predictions of changes in flame development and propagation.

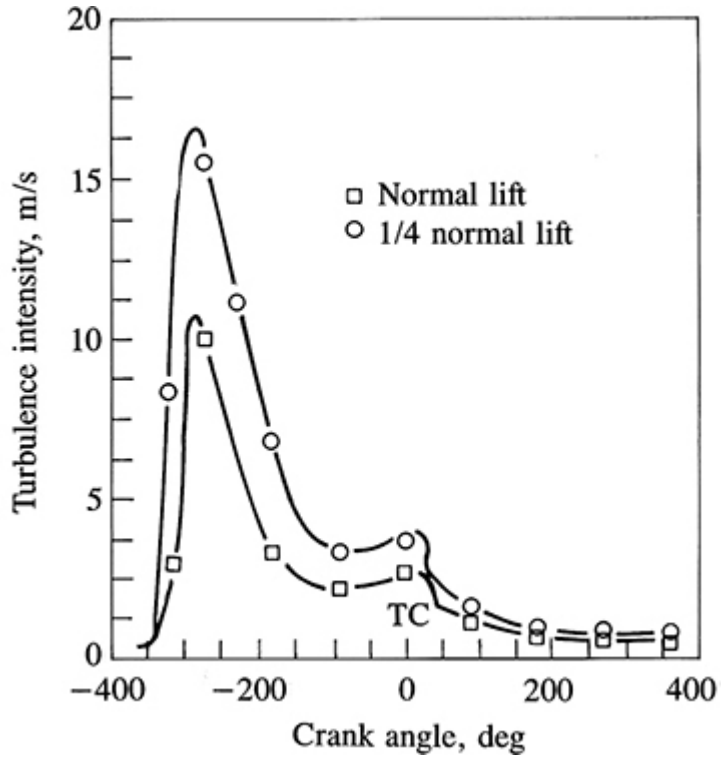


Figure 14.14 Predicted turbulence intensity u' as a function of crank angle and valve lift in a gasoline engine operating at 1500 rev/min, 414 kPa imep, with a compression ratio of 10.[26]

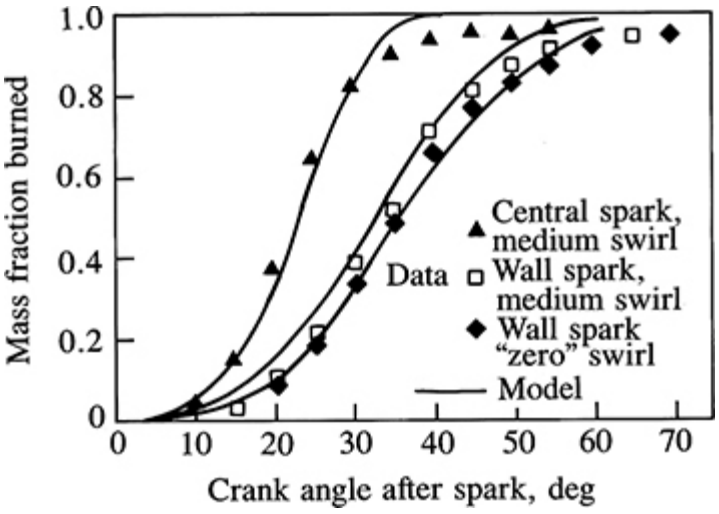


Figure 14.15 Comparisons of predicted and measured mass fraction burned versus crank-angle profiles for various swirl levels and plug locations.[27]

The above type of combustion model has been used to obtain explicit relations for the flame development and rapid burning angles as functions of engine design and operating variables. [28] The equations for the mass burning rate, (14.33 and 14.34), were effectively integrated over the relevant portion of the total combustion process; the turbulent characteristic velocity was assumed proportional to $\overline{S}_p$, the mean piston speed. The flame development angle was found to vary as

$$\Delta\theta_d = C(\overline{S}_p \nu)^{1/3}\left(\frac{h}{S_L}\right)^{2/3} \tag{14.43}$$

where ν is the kinematic viscosity ($\nu = \mu/\rho$) and h is the clearance height at ignition. C is a constant which depends on engine geometry and is determined by matching Eq. (14.43) with engine data. The rapid burn angle (here taken as the crank angle between $x_b = 0.01$ and 1.0) is given by

$$\Delta\theta_b = C'\left(\frac{B}{h^*}\right)\left(\frac{\rho_i}{\rho_u{}^*}\right)^{10/9}(\overline{S}_p \nu^*)^{1/3}\left(\frac{h_i}{S_L{}^*}\right)^{2/3} \tag{14.44}$$

where C' is a constant which depends on engine geometry, B is the bore, the subscript i denotes the value at ignition, and the superscript * denotes the value at cylinder conditions where $x_b = 0.5$. These expressions show reasonable agreement with observed trends in $\Delta\theta_d$ and $\Delta\theta_b$.

14.4.3 Direct-Injection Engine Models

In direct-injection diesel engines, the liquid fuel is injected into the cylinder as one or several jets just prior to ignition. In larger direct-injection compression-ignition engines, the airflow in the cylinder is essentially quiescent. However, in medium and smaller size DI engines, the air in the piston bowl during combustion is usually swirling about the cylinder axis at up to 10 times the crankshaft rotational speed; this swirling air-flow pattern increases the rate of entrainment of air into the fuel jet, increasing the fuel-air mixing rate. Thus modeling of the ignition and combustion processes (and thus performance) of direct-injection diesel engines is much more complex

than for premixed-charge spark-ignition engines. The unsteady liquid-fuel jet phenomena—atomization, liquid jet and droplet motion, fuel vaporization, air entrainment, fuel-air mixing, and the ignition chemistry—all play a role in the heat-release process (see Chap. 10). It is not yet possible to model all these phenomena from a fundamental basis, even with the most sophisticated fluid-dynamic-based codes now available (see Sec. 14.5), since many of these processes are not yet adequately understood. However, models at various levels of detail and empiricism have been developed and have proven useful in direct-injection diesel engine analysis.

The modeling of direct-injection gasoline SI engines also introduces additional complexity. As discussed in Sec. 9.3.5 , DI gasoline engines fall into two distinct categories depending on the nature of the in-cylinder fuel distribution during combustion: homogeneous charge, and stratified. While fuel injection into the cylinder adds some additional complexity, with early injection (during the intake stroke) the mixture becomes relatively homogeneous by start of combustion (though fuel injection into the intake port, vaporization off the intake valve and port surfaces, and mixing with air during flow past the valve head and seat, does produce a more uniform in-cylinder mixture). With stratified DI operation, where the fuel is injected into the cylinder during the middle or later portion of the compression stroke, spray behavior, fuel vaporization, and spray/air motion interactions all become important. But because of the different injection timing (to that in DI diesels), these processes are significantly different. We review stratified DI gasoline SI engine combustion modeling at the end of this section. Homogeneous DI gasoline SI engines are usually modeled, as are port-injected gasoline engines, as operating with close to a premixed essentially uniform fuel-vapor, air, and burned gas mixture. We approach this discussion of phenomenological models of DI engine combustion from a historical perspective since that helps prioritize the more important features of the overall combustion process. Additional details can be found in the references provided.

Diesels

This section reviews the important features of single-zone heat-release models and phenomenological spray-based combustion models for diesels. The relative simplicity and modest computer time requirements of these

models make them especially useful for diesel cycle simulation and broader engine system studies.

Single-zone models assume that the cylinder contents can be adequately described by property values representing the average state, and use one or more algebraic formulas to define the heat-release rate. The functional forms of these formulas are chosen to match experimentally observed heat-release profiles (see Sec. 10.3.3). Coefficients in these formulas, which may vary with engine design details and operating conditions, are determined empirically by fitting with data. Earlier models used the phenomenological description of diesel combustion developed by Lyn (see Sec. 10.3.2) which comprised three primary phases: the ignition delay period, the premixed fuel-burning phase, and the mixing-controlled fuel-burning phase. Ignition delay correlations are reviewed in Sec. 10.5.6 . Here models for the second and third phases, when the major heat release occurs, are summarized (see Ref. 29 for a more extensive review). The attraction of the one-zone heat-release approach is its simplicity; however, since it is a major simplification of the complex phenomena which comprise the combustion process, substantial empirical input must be used. One-zone heat-release models use simple equations to describe the rate of release of the fuel's chemical energy, sometimes modeled on the presumed controlling physical or chemical process and always calibrated by comparison with data.

One extensively used model of this type developed by Watson et al. [30] is especially appropriate for use in overall diesel engine system simulations where the combustion process details are not the primary focus. It is based on Lyn's description of compression-ignition combustion—a rapid premixed burning phase followed by a slower mixing-controlled burning phase. The fraction of the injected fuel that burns in each of these phases is empirically linked to the duration of the ignition delay. One algebraic function is used to describe the premixed heat-release phase and a second function to describe the mixing-controlled heat-release phase. These two functions are weighted with a phase proportionality factor, β, which is largely a function of the ignition delay. Thus:

$$\frac{m_{f,b}(t')}{m_{f,0}} = \beta f_1 + (1-\beta) f_2 \tag{14.45}$$

where $m_{f,b}$ is the mass of fuel burned, $m_{f,0}$ is the total fuel mass injected per cycle per cylinder, and t' is time from ignition non-dimensionalized by total time allowed for combustion [$= (t - t_{ign})/\Delta t_{comb}$]. [d] The premixed-burning function f_1 is

$$f_1 = 1 - \left(1 - t'^{K_1}\right)^{K_2} \tag{14.46}$$

and the mixing-controlled function f_2 is

$$f_2 = 1 - \exp\left(-K_3 t'^{K_4}\right) \tag{14.47}$$

where K_1, K_2, K_3, and K_4 are empirical coefficients. The proportionality factor b was given by

$$\beta = 1 - \frac{a\phi^b}{\tau_{id}^c} \tag{14.48}$$

where φ is the overall fuel/air equivalence ratio and a, b, and c are empirical constants. Correlation with data from a typical turbocharged truck engine gave expressions for K_1 to K_4. [e]

Such single-zone heat-release models can be useful because of their simplicity. They obviously cannot relate engine design and operating variables explicitly to the details of the combustion process. Experience indicates that one-zone combustion models with only one functional equation are not usually able to fit experimentally determined heat-release profiles with sufficient accuracy. Single-zone heat-release models should be checked against experimentally derived heat-release profiles and recalibrated, if necessary, before being used for predictions.

The overall structure of complete thermodynamic-based phenomenological models for predicting diesel engine performance and emissions requires much more than a combustion model. All the important processes (intake, compression, combustion, expansion, heat transfer, NO x and particulate emissions formation mechanisms) must be included and connected. Figure 14.16 illustrates the major components and layout of such *engine cycle simulations*. As is clear, many different complex sub-models

have to come together. An example of a diesel spray development and combustion model is shown in Fig. 14.17. It is based on a spray model with many small packets, each with the same fuel mass, from injection through combustion. The intent is to follow the spray development in a swirling airflow and the spray interaction with the combustion chamber walls. The liquid fuel which enters the chamber through the injector nozzle is divided into many small equal mass "elements." The spray motion is defined by an experimentally based correlation. Air entrainment is calculated from momentum conservation and the spray velocity decrease predicted by this correlation. The processes which occur within each element are also illustrated in Fig. 14.17. The fuel drops evaporate and fuel vapor mixes with entrained air. When ignition occurs, combustible mixture prepared before ignition burns rapidly. The continuation of the burning process may be limited by either the rate of production of fuel vapor by evaporation or the availability of air by the rate of entrainment (paths *A* and *B* in Fig. 14.17). The growth of the spray is determined from the air entrainment into each element and the combustion-produced expansion of each element. When impingement on the wall occurs, the spray is assumed to spread along the wall. When the periphery of the spray reaches that of a neighboring spray, the sideways growth of the spray is then prevented and the thickness of the elements along the wall increases. Swirl effects are calculated from tangential momentum considerations.

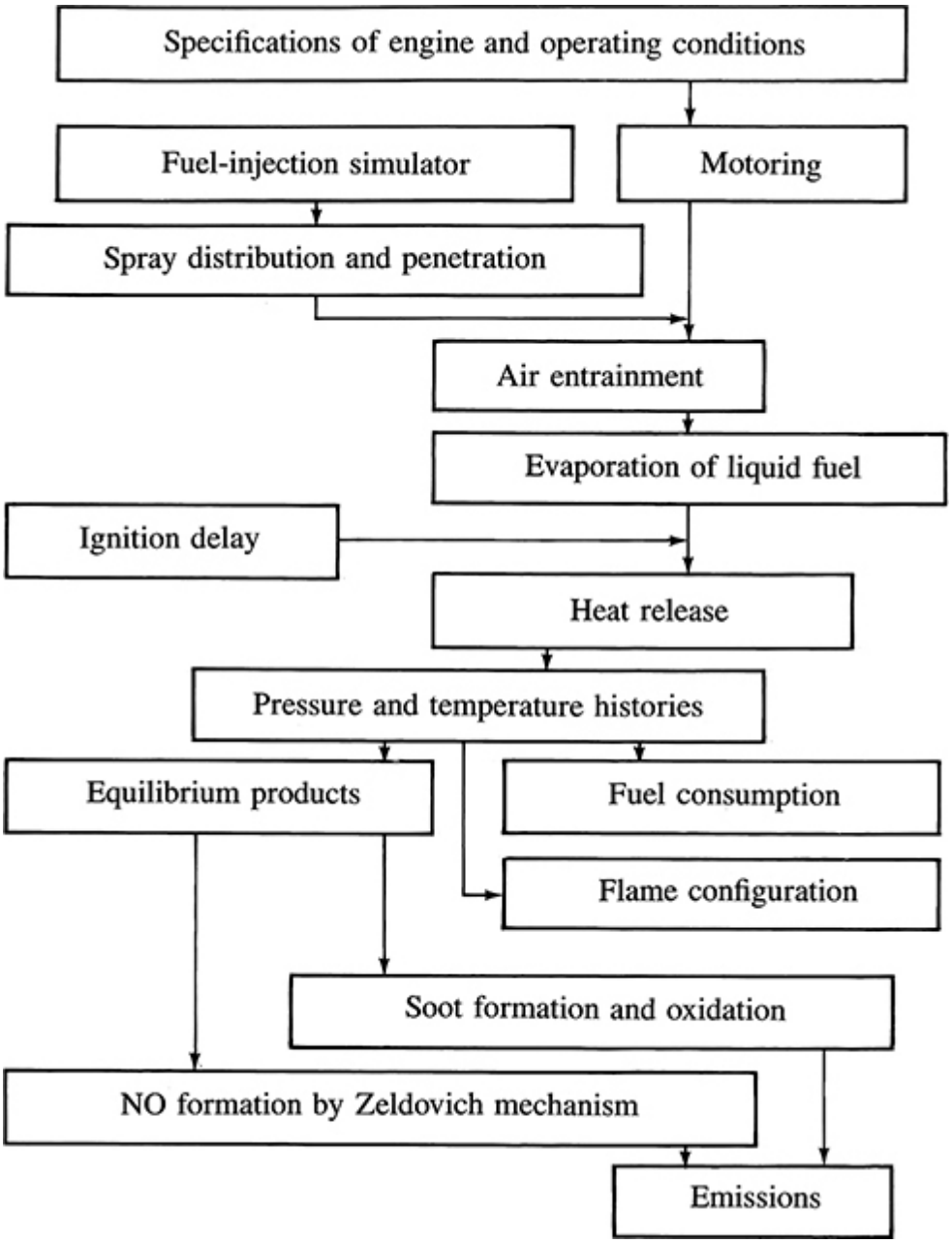


Figure 14.16 Structure of thermodynamic-based DI diesel simulation for predicting engine performance and emissions.[31]

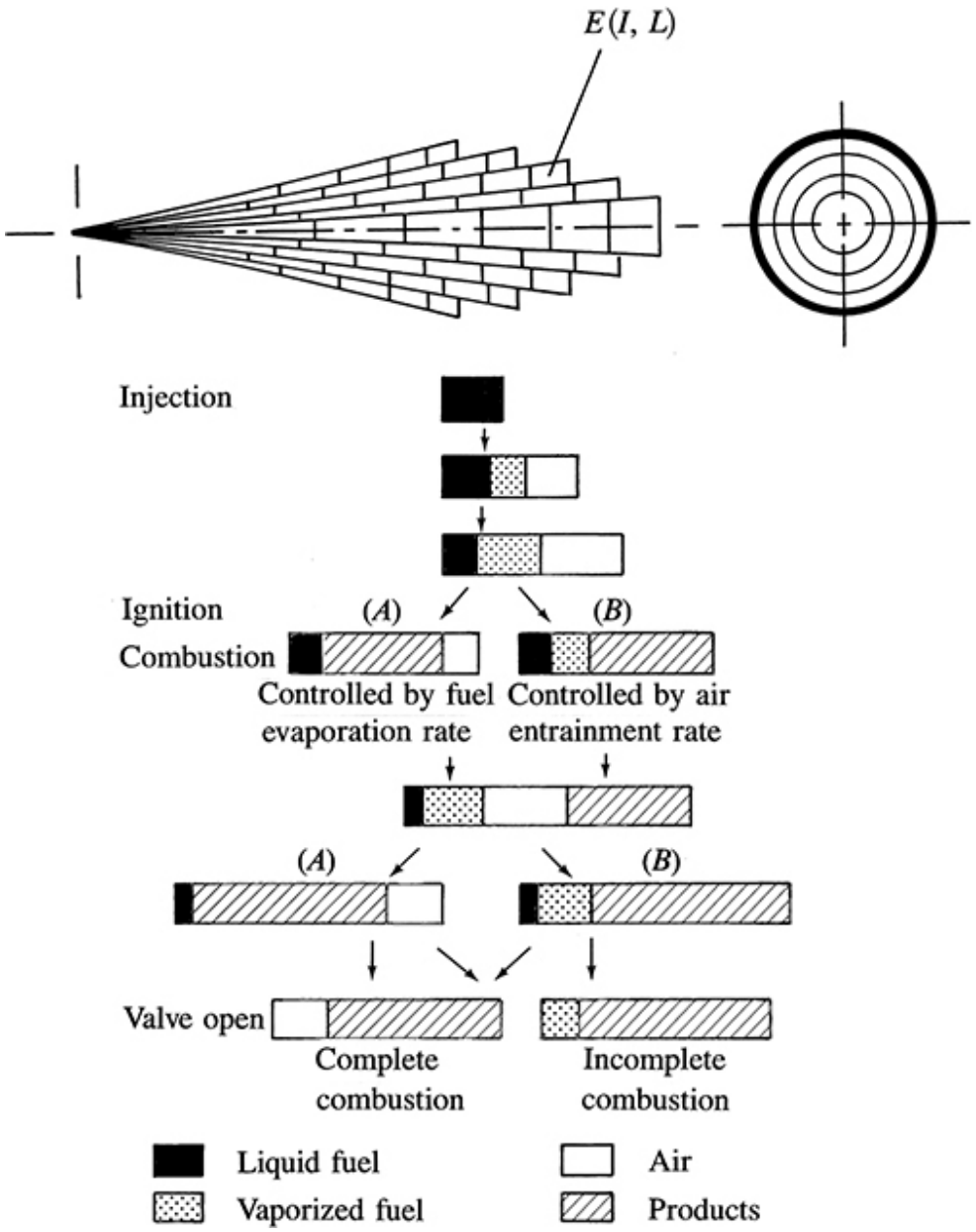


Figure 14.17 Schematic of spray model with many small packages, each with the same fuel mass, and of the processes that occur within each package, developed and used by Hiroyasu et al.[32]

The heat release rate in the combustion chamber is obtained by summing up the heat release in each element. Nitric oxide and soot formation calculations are based on the time histories of temperature, vaporized fuel, air and combustion products in each element. Figure 14.18 shows an example of the output from the above model. The injection rate diagram, the assumed

Sauter mean drop size of the spray, and the air swirl determine the spray development, which leads to the heat-release rate predictions. This determines the cylinder pressure profile. Predicted engine performance results show reasonable agreement with data. A review of these types of jet models is given by Hiroyasu. [33]

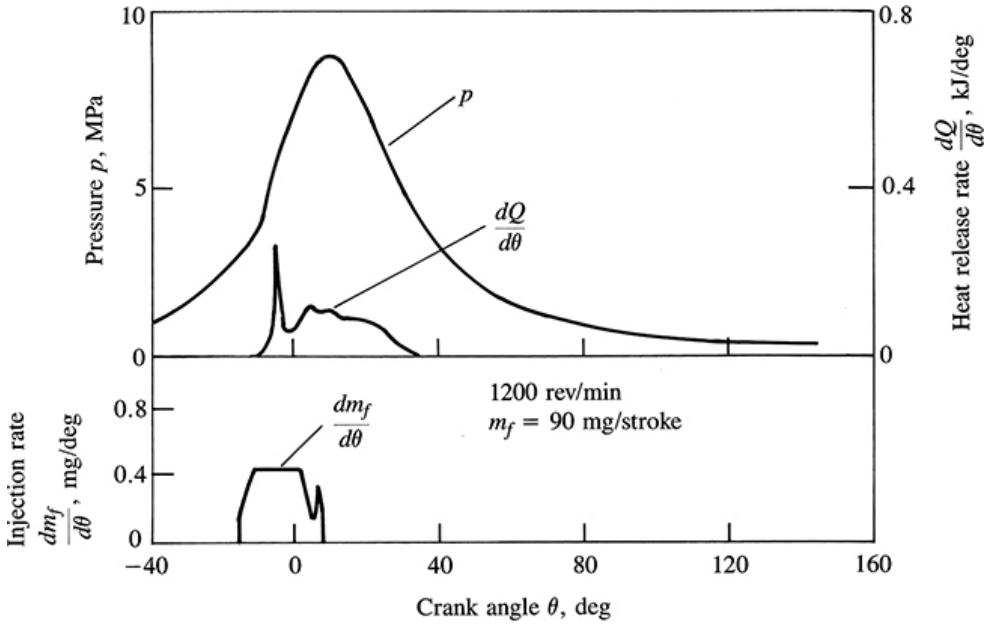


Figure 14.18 Fuel injection rate, heat release rate profile, and cylinder pressure predicted with phenomenological DI diesel cycle simulation.[31]

Our understanding of diesel fuel spray behavior has advanced significantly from what these historical models assumed and could be included. Extensive correlations that describe the evolution of diesel fuel sprays are now available (see Secs. 10.4.2 to 10.4.6). The air entrainment process into each spray along the lift-off portion of the spray has been quantified. We now understand that the "rapid premixed" burning (second) phase occurs within each spray under fuel rich conditions, and the main burning phase occurs in the diffusion flame that rapidly envelops each developing fuel spray (see Sec. 10.3.4).

DI Gasoline Engines

Direct-injection gasoline-fueled spark-ignition engines can be operated in *homogeneous* or *stratified* mode. In homogeneous mode, the in-cylinder charge during combustion is usually well-enough mixed to be modeled as

"uniform." However, direct injection into the in-cylinder charge does result in significant cooling of that charge as the fuel spray evaporates. With gasoline fuel, the maximum reduction in charge temperature is about 20° degrees C. Ethanol has a much higher heat of vaporization than gasoline: with 100% ethanol the maximum (potential) reduction is some 80 degrees C. Blends of ethanol with gasoline have evaporative cooling impacts that are linearly in between. Also, in homogeneous mode, especially when the engine has not fully warmed-up, the penetration and trajectory of the DI spray and consequent wall-wetting with liquid fuel can be important. Both these effects need to be modeled.

In stratified mode, usually at low- to mid-loads, a uniform in-cylinder mixture is not the objective. The injected fuel needs to mix with enough air yet stay confined within an evolving spray (or set of sprays) to form a "cloud" of close to stoichiometric mixture. Thus modeling the fuel spray trajectory and dispersion is an important task.

Figure 14.19 illustrates both the spray modeling approach, and an example of model results. The injected spray is analyzed as a quasi-steady one-dimensional turbulent gaseous jet. [34.] The intention is to describe the position of the jet as a function of time, as illustrated in Fig. 14.19 *a*. Entrainment of air into the jet is assumed to take place at each point along the jet surface at a rate proportional to the velocity difference between the jet and surrounding air at that point. Two empirical entrainment coeffcients [35] are used for the proportionality constants for the relative motion in the jet axial and transverse directions. Conservation equations for fuel mass and total mass, and momentum (in two or three orthogonal directions) are used to determine the jet trajectory and size. The jet slows down due to air entrainment. Deflection of the jet results forms the entrainment of air with a momentum component normal to the jet axis, and forms drag forces due to the normal component of air flow past the jet. This approach does not define the velocity and concentration profiles across the jet: it only calculates the mean values at any jet axial position. Experimentally determined radial profiles for an axisymmetric turbulent jet [36] are often assumed. Although the fuel spray is initially mainly liquid, the fuel drops soon become a small fraction of the jet volume due to vaporization and air entrainment. Downstream of the initial liquid breakup region, the velocity of the drops relative to that of the vaporized fuel and air is small, so the spray acts as a gaseous jet.

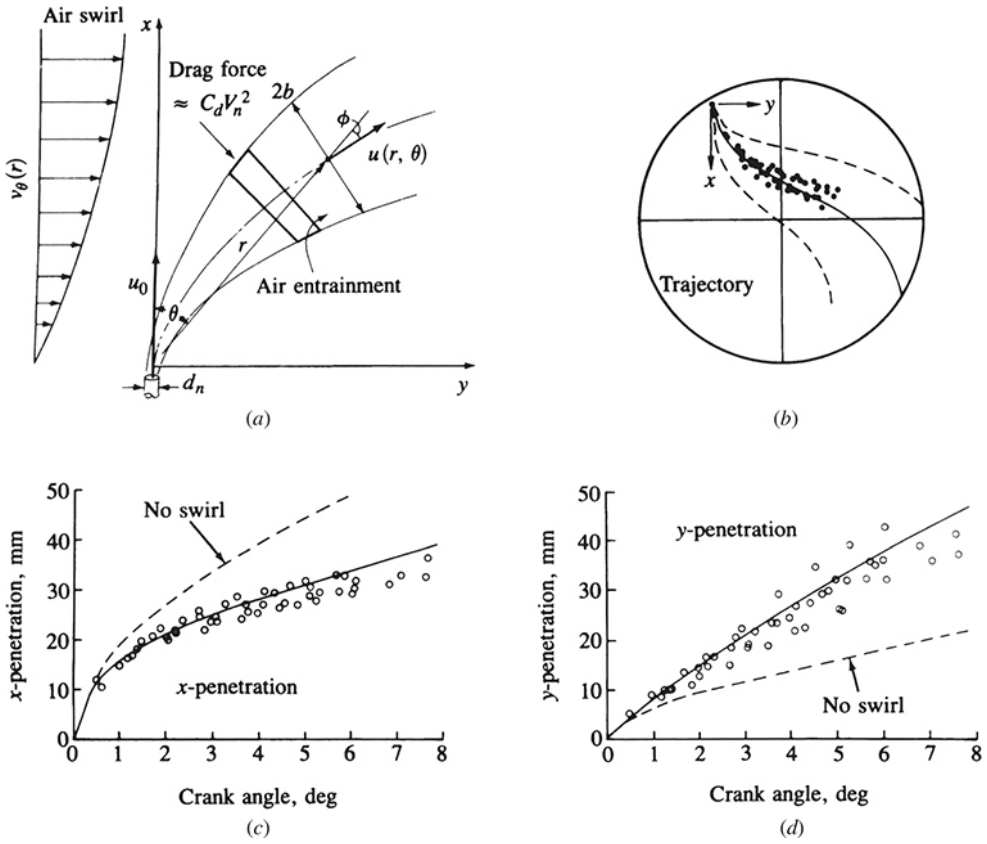


Figure 14.19 (*a*) Schematic of one-dimensional quasi-steady fuel spray model used to define spray centerline trajectory and width as the outward-moving spray interacts with the swirling airflow. (*b*) Spray trajectory and width calculated using a one-dimensional quasi-steady spray model of the type illustrated in (*a*), compared with experimental data taken in special visualization direct-injection stratified-charge engine. (*c*) and (*d*) Jet penetration as a function of time with and without swirl. [37]

In this comparison between a gaseous jet model and an experimental engine spray, shown in Fig. 14.19 *b*, *c*, and *d*, a single fuel jet was injected into a disc-shaped chamber in the location shown, and Schlieren photography used to observe the spray trajectory. Good agreement was obtained for the spray center-line: note the significant effect of swirl. Reasonable agreement was also obtained between predicted and measured spray boundaries.

Combustion models for stratified operation of DI gasoline engines are more complex. Combustion of the stratified in-cylinder mixture usually occurs in two stages (see Sec. 9.3.5). The spray region or fuel vapor/air cloud where much of the fuel-air mixture will readily burn (since it is close

to stoichiometric and has a high laminar flame speed) is ignited by the spark discharge, and a turbulent thin wrinkled-laminar flame propagates through this partially premixed region. This is followed by the still-rapid burnout of locally rich regions behind the flame front as oxygen from leaner regions and some of the surrounding air-mix into the hot post flame gases. Then a slower mixing-controlled combustion occurs of the remaining fuel-rich pockets, such as that resulting from the last-injected fuel, which enters the combustion chamber at lower velocity with poorer atomization and thus slower fuel-air mixing.

Models that analyze such a complex combustion process embedded in the fuel spray evolution and geometric detail (location of the spark plug and piston bowl in the context of the spray location and composition) are built into professional engine simulation software such as GT-Power. [3] (See Sec. 14.4.5 .)

14.4.4 Prechamber Engine Models

Historically, small high-speed compression-ignition engines used an auxiliary combustion chamber, or prechamber, to achieve high fuel-air mixing rates. The prechamber was connected to the main combustion chamber above the piston via a nozzle, passageway, or one or more orifices (see Secs. 1.9 , 8.7, and 10.2.2). Auxiliary chambers have sometimes been used in spark-ignition engines, also. The plasma and flame-jet ignition systems described in Sec. 9.5.3 enclose the spark plug in a cavity or small prechamber which connects to the main chamber via one or more orifices. The function of the prechamber is to increase the initial growth rate of the flame. Combustion in the main chamber is initiated by one or more flame jets emanating from the prechamber created by the ignition process and subsequent energy release within the prechamber. If the mixture within the prechamber is richer than that in the main chamber (due to fuel injection or a separate prechamber intake valve—see Sec. 1.9), these are called stratified-charge engines.

The additional phenomena which these prechambers introduce beyond those already present in conventional chamber engines are: (1) gas flows through the nozzle or orifice between the main chamber and prechamber due to piston motion; (2) gas flows between these chambers due to the combustion-generated pressure rise; and (3) heat is transferred to the nozzle

or passageway walls due to these flows. The first of these phenomena results in nonuniform composition and temperature distributions between the main and prechamber due to gas displacement primarily during compression, and determines the nature of the flow field within the prechamber toward the end of compression just prior to combustion. The second phenomenon controls the rate of energy release in the main chamber. The heat losses in the passageway and to the additional chamber surface area of the prechamber designs relative to conventional open chambers result in decreased engine performance and efficiency. Thus the prechamber concept adds additional complexity to the engine processes that must be modeled to predict engine behavior.

The following variables are important to prechamber engine performance and emissions characteristics, in addition to the design and operating variables which govern single-chamber engine behavior: prechamber geometry—size, shape, flow area, and shape of connecting passageway(s); prechamber location in relation to main chamber geometry; geometry and timing of any auxiliary prechamber valve; fuel metering strategy in prechamber compression-ignition or stratified-charge engines. Thermodynamic-based models have been developed and used to examine the overall impact of these variables (see Ref. 38). CFD models (see Sec. 14.5) have also been used to examine specific prechamber engine flow and combustion processes.

Useful predictions of fuel, air, and residual gas distributions and the corresponding temperature within the prechamber and main chamber can be obtained with simple gas displacement models. Only during combustion is the pressure difference across the nozzle or orifice sufficiently large in magnitude for its modeling to be essential; the assumption of uniform pressure during compression, the critical process for determining conditions just prior to combustion, introduces little error into calculations of the flows between the chambers. Section 8.5 develops the appropriate equations for these piston-motion driven gas displacements. Use of the conservation equations for an open system, for total mass, fuel mass, residual gas, and energy given in Sec. 14.2 , for the main chamber and the prechamber, then give the mean composition and temperature variation in each chamber as a function of time due to this flow.

During combustion, the pressure difference across the connecting passageway or orifice is the driving force for the flow between chambers.

Since combustion starts in the prechamber, the initial flow is into the main chamber; later, as the heat release in the main chamber becomes dominant, the flow may reverse direction and be into the prechamber. In thermodynamic-based models, the equations for one-dimensional quasi-steady ideal gas flow through a restriction given in App. C are used to relate these flows to the pressure difference between the two chambers. Open-system conservation equations are again used to calculate mean properties in each chamber.

Combustion models used are either empirically based [e.g., using specified heat-release or mass burning rates such as Eq. (14.32) [39]] or are developed from direct-injection compression-ignition engine models with spray evaporation, fuel-air mixing, and ignition delay processes explicitly included.[40] Because of the complexity of these processes in the prechamber engine geometry, substantial simplifying assumptions and empiricism must be used.

Heat transfer to the passageway and chamber walls is affected by the flows between the chambers: high velocities within the passageway result in high heat-transfer rates to the passageway walls, and the vigorous flows set by the passageway exit flow entering the prechamber or the main chamber increase heat-transfer rates to the walls of these chambers. The standard engine heat-transfer correlations which relate the heat-transfer coefficient to mean flow field variables via Nusselt-Reynolds number relationships (see Sec. 12.4) are normally used to describe these heat-transfer processes. The length scales are chosen to match the prechamber or main chamber or passageway dimensions. The characteristic velocities in these relationships are equated with velocities which are representative of the flow in each of these regions at the relevant time in the engine operating cycle. [41,42]

The utility of the more sophisticated of these prechamber engine performance and emissions models is illustrated by the results shown in Fig. 14.20. This simulation of the indirect-injection compression-ignition engine's flow and combustion processes describes, through the use of stochastic mixing models, the development of the fuel/air ratio distribution and fuel-energy release distribution, and hence the development of the gas pressure and gas temperature distribution, within the prechamber and main chambers of the engine. With the (nonuniform) gas composition and state defined, the models for NO formation described in Sec. 11.2.1 was used to predict NO*x*

emissions. The approaches used to describe the evolution of the prechamber, main chamber, and passageway contents are summarized in Fig. 14.20 *a*.

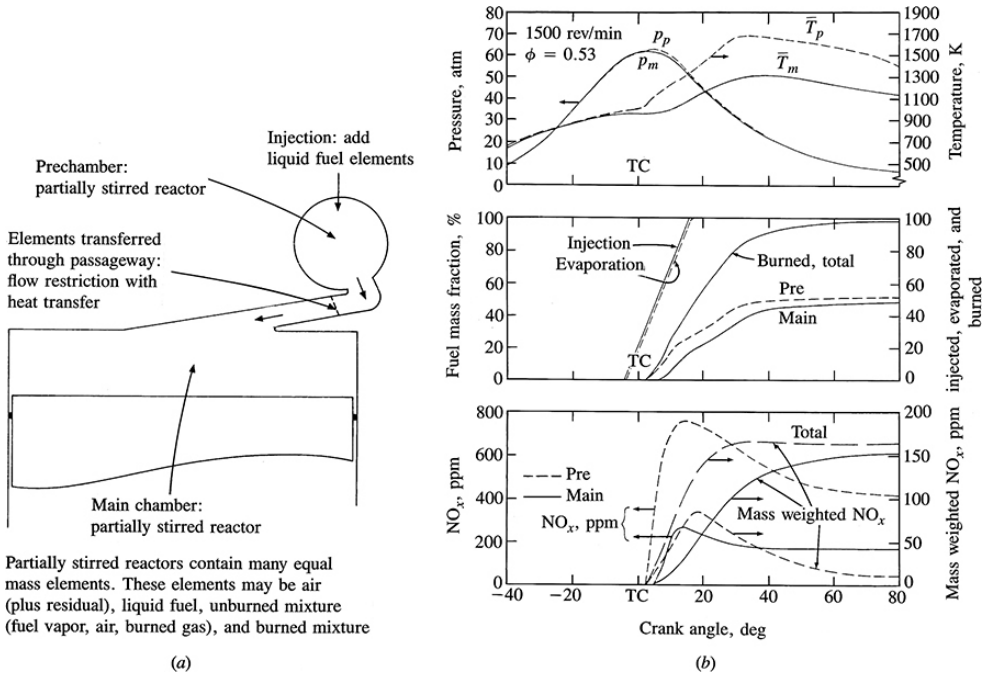


Figure 14.20 (*a*) Schematic of IDI diesel engine illustrating how stochastic mixing models are applied to the prechamber and main chamber, with a connecting passageway, to simulate turbulent mixing processes and pressure-driven flows; (*b*) example of simulation predictions through the engine's operating cycle. Shown are prechamber and main chamber pressures, prechamber and main chamber average gas temperatures; fuel mass injected, evaporated, and burned in prechamber and main chamber; average NO *x* concentration in each chamber (and total) in ppm (by volume) and mass weighted ppm (mass in chamber × NO *x* concentration in chamber/total mass in cylinder). [43]

The cylinder contents were divided up into a large number of elements. Pairs of elements are selected at random to undergo "turbulent mixing" interactions at a frequency related to the turbulence in each region. Rate processes—evaporation, ignition, NO formation, etc.—proceed within each element between these mixing interactions. Figure 14.20 *b* shows sample results. At about TC, after some of the injected fuel has evaporated and the ignition delay is over, combustion starts in the prechamber and the

prechamber pressure p_p rises above the main chamber pressure p_m. This forces air, fuel, and burned gases to flow from the prechamber into the main chamber; fuel and rich products can now mix with air and burn in the main chamber. NO starts to form in each mass element once it burns, at a rate dependent on each element's composition and state. Most of the NO forms within the prechamber and then flows into the main chamber as the expansion process proceeds. The attractive feature of this type of emission calculation is that the kinetically controlled NO formation calculations are based directly on local gas composition and temperature in a manner that approximates the mean and turbulent nonuniformities in these variables. Predictions of engine operation and emissions showed good agreement with data. [43]

14.4.5 Multi-Cylinder and Complex Engine System Models

The models discussed in the previous parts of Sec. 14.4 focus on the processes occurring within *each* cylinder of an internal combustion engine. Most engines are multi-cylinder engines and the individual cylinders interact via the intake and exhaust manifolds. Also, many engine systems are more complex: internal combustion engines can be supercharged, turbocharged, or turbocompounded, and the manifolds then connect to the atmosphere via compressors or turbines (see Fig. 6.45 and Sec. 6.8). Thermodynamic-based simulations of the relevant engine processes, constructed from the types of model components already described, prove extremely useful for examining the behavior of these more complex engine systems. By describing the mass and energy flows between individual components and cylinders of such systems throughout the engine's operating cycle, the total system performance can be predicted. Such models have been used to examine steady-state engine operation at constant load and speed (e.g., where time-varying conditions in the manifolds due to individual cylinder filling and emptying events affect multi-cylinder engine behavior), and how the total system responds to changes in load and speed during engine transients.

The block diagram of a turbocharged and turbocompounded diesel engine system in Fig. 14.21 illustrates the interactions between the system components. By describing the mass and energy flows between components and the heat and work transfers within each component, total system behavior

can be studied. In such engine simulations, the reciprocator cylinders, the intake manifold, and the various sections of the exhaust system are treated as connected open systems. The flows into and out of these volumes are usually analyzed using the quasi-steady emptying and filling approach described in Sec. 14.3.3 , using the open-system conservation equations of Sec. 14.2 . The reciprocator cycle is treated as a sequence of processes within each cylinder: intake, compression, combustion expansion, and exhaust. These are modeled using the approaches described previously in Secs. 14.4.1 to 14.4.4. Heat transfer has, of course, an important effect on the in-cylinder processes. It also is important in the exhaust system since the performance of the turbocharger turbine and of any compounded turbine depends on the gas state at turbine inlet. The performance of the turbomachinery components is normally defined by maps that interrelate efficiency, pressure ratio, mass flow rate, and shaft speed for each component (see Secs. 6.8.2 to 6.8.4).

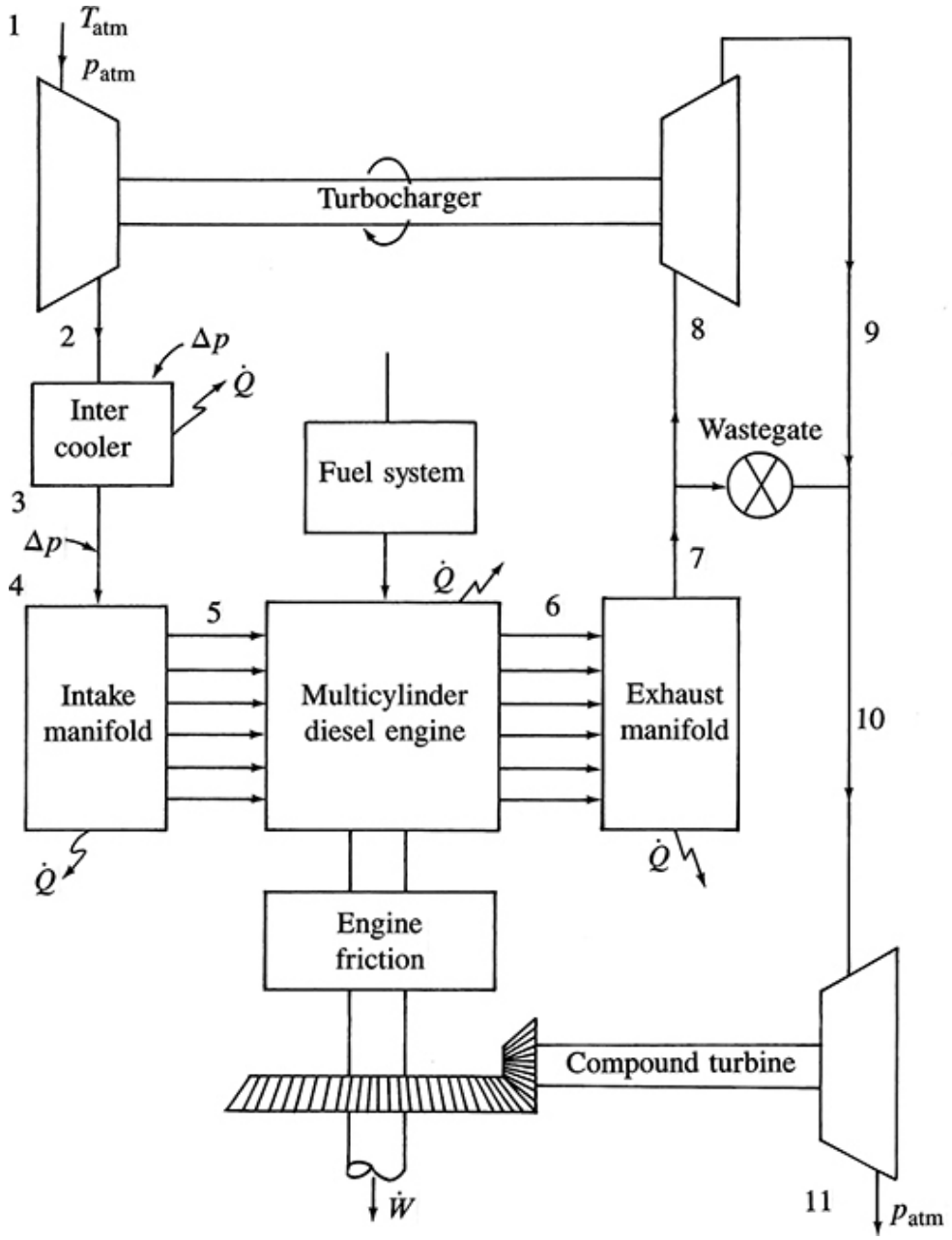


Figure 14.21 Block diagram of turbocharged turbocompounded diesel engine system.[2]

When a reciprocating engine is coupled with turbomachinery, its manifolds no longer connect directly with the atmosphere: matching procedures are required to ensure that the pressure levels and mass flow rates of the compressor and turbines match with those of the engine. The following matching process is typical of those used for the simpler type of

turbocharged engine (one compressor and one turbine). At a given time, the values of the variables describing the state of the various system components are known (from integration of the system governing equations over the previous time step). These include the intake and exhaust manifold pressures and the turbocharger rotor speed. The compressor inlet pressure is atmospheric pressure less the intake air-filter pressure drop. The turbine exit pressure is atmospheric plus the aftertreatment system, the exhaust system, and muffler pressure drops. By relating the compressor discharge pressure to the intake manifold pressure, and the turbine inlet pressure to the exhaust manifold pressure (incorporating suitable pressure drops), the pressure ratio across each machine is determined. Hence, the compressor and turbine maps can be entered using the calculated pressure ratios and the rotor speed as inputs (the latter the same for both the compressor and turbine). The output from the map interpolation routines determines the mass flow rate and efficiency of each component for the next time step. From these, the power required to drive the compressor ($-\dot{W}_C$), and to drive the turbine ($\dot{W}_T$) are determined from Eqs. (6.42) and (6.48), respectively. Any excess power (or power deficiency) will result in a change of rotor speed according to the turbocharger dynamics equation

$$\dot{W}_C + \dot{W}_T = I_{\mathrm{TC}}\omega\frac{d\omega}{dt} + B\omega^2 \tag{14.49}$$

where I_{TC} is the rotational inertia of the turbocharger, ω is its angular velocity, and B is the rotational damping. The values of the other state variables for the next time step are determined from the solution of the mass and energy conservation equations for each open system, with the compressor and turbine mass flows taken from the output of the turbomachinery map interpolation routines.

This approach can be used to establish the steady-state engine operating characteristics from an assumed initial set of state variables. (Of course, due to the pulsating nature of the flows into and out of the cylinders, these state variables will vary in a periodic fashion throughout the engine cycle at a fixed engine load and speed.) This approach can also be used to follow transient engine behavior as load or speed is varied from such a steady-state condition. The additional inputs required are the fuel pump delivery characteristics as a function of fuel pump rack position and speed, with the

latter evaluated from an appropriate model for dynamic behavior of the fuel system. From the brake torque of the engine T_B (determined by subtracting friction torque from the indicated torque), the torque required by the load T_L, the inertia of the engine and the load I_E and I_L, the dynamic response of the engine and load to changing fuel rate or engine speed can be obtained from

$$T_B - T_L = (I_E + I_L)\frac{d\omega}{dt} \tag{14.50}$$

An example of the output from this type of engine model is plotted in Fig. 14.22 where the response of a turbocharged DI diesel engine to an increase in load from 0 to 95% of full load is shown. The predictions come from a model of the type shown in Fig. 14.21, and engine details correspond to the experimental configuration. [44] The simulation follows the data through the engine transient with reasonable accuracy. Note that with the assumed fuel injection system response, during this transient the equivalence ratio of the trapped mixture rises to close to stoichiometric because the increase in air flow lags the increase in fuel flow. This would result in excessive smoke emissions. Such models prove extremely useful for exploring the effect of changes in engine system design on transient response. [45]

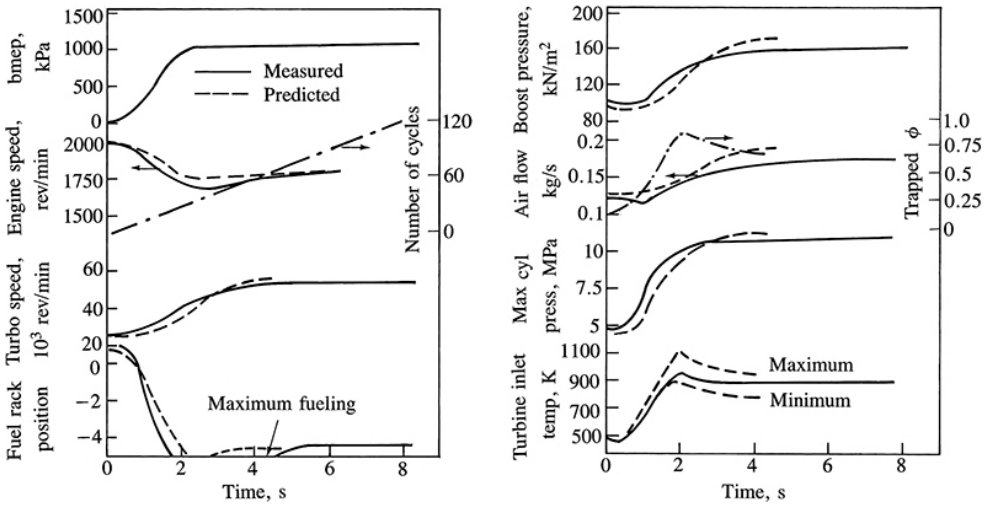


Figure 14.22 Predicted (– – –) and measured (—) response of a turbocharged direct-injection diesel engine to an increase in load.[44]

For two-staged turbocharged or turbocompounded systems the engine turbocharger matching process is more complicated. The division of the pressure ratio between the exhaust manifold and atmosphere between the two turbines in Fig. 14.21 is not known *a priori*. Nor, with two compressors, is the intake pressure ratio distribution known. Iterative procedures based on an assumed mass flow rate are used to determine the pressure level between the two turbines such that mass flow and pressure continuity through the exhaust system are satisfied (e.g., Ref. 2).

Commercial engine simulation software for analyzing the performance and emissions of internal combustion engines of many types and sizes are now available at high levels of sophistication and completeness. One such widely-used and comprehensive model is GT-Power. [3] It can be used to analyze naturally-aspirated, turbocharged, supercharged, turbocompounded, spark-ignition, diesel, homogeneous-charge compression-ignition (HCCI) engines, operating on a range of fuels. It can predict steady-state or transient engine behavior. It incorporates variable valve timing and lift. It calculates, from the basic principles reviewed in this section, airflow and volumetric efficiency, fuel consumption and fuel conversion efficiency, torque and power curves, and emissions. It does acoustic analysis of intake and exhaust systems, analysis of emissions reductions in exhaust aftertreatment systems (catalysts and filters), and finite-element thermal analysis of manifold and cylinder components.

14.4.6 Second-Law Analysis of Engine Processes

The first-law-based methods for evaluating power plant performances do not explicitly identify those processes within the engine system that cause unrecoverable degradation of the thermodynamic state of the working fluid. However, second-law-based analysis methods do provide the capability to identify and quantify this unrecoverable energy degradation. The cause and effect relationships that relate these losses to individual engine processes can, therefore, be determined. The first-law analysis approaches summarized in this section (14.4) are based on the fact that energy is conserved in every engine component and process. Thus, they take account of the conversion of energy from one form to another: for example, chemical, thermal, and mechanical. Although energy is conserved, second-law analysis indicates

that various forms of energy have differing levels of ability to do useful work. This ability to perform useful work is defined as *availability*.

The availability of a system at a given state is defined as the amount of useful work that could be obtained from the combination of the system and its surrounding atmosphere, as the system goes through reversible processes to equilibrate with the atmosphere. It is a property of the system and the environment with which the system interacts, and its value depends on both the state of the system and the properties of the atmosphere. Availability is not a conserved property; availability is destroyed by irreversibilities in any process the system undergoes. When availability destruction occurs, the potential for the system to do useful work is permanently decreased. Thus to make a proper evaluation of the processes occurring within an engine system, both energy and availability must be considered concurrently.

The basis for an availability analysis of realistic models of internal combustion engine processes has already been developed in Sec. 5.7 . The change in availability of any system undergoing any process where work, heat, and mass transfers across the system boundary occur (see Fig. 5.13) can be written as

$$\Delta A = A_{\text{in}} - A_{\text{out}} - A_{\text{destroyed}} \tag{14.51}$$

where A_{in} and A_{out} represent the availability transfers into and out of the system across the system boundary. Since availability is not a conserved quantity, this equation can only be used to solve for the availability destruction term, $A_{\text{destroyed}}$. Table 14.2 summarizes the equations for the availability change of the system and the availability transfers associated with work, heat and mass transfer across the system boundary, developed in Sec. 5.7 .

TABLE 14.2 Available energy equations for various processes

Mechanism	Equation
Work transfer	$dA_w = dW$
Heat transfer	$dA_Q = dQ(1 - T_0/T)$
Gas transfer	$dA_g = dm_g[(h - h_0) - T_0(S - S_0)]$
Liquid fuel transfer*	$dA_f = dm_f(1.0338Q_{LHV})$
Control volume storage	$dA_{cv} = d\{m_{cv}[(u - u_0) - T_0(s - s_0) + p_0(v - v_0)]\}$

*The availability of the fuel is 1.0338 times its lower heating value; see Sec. 5.7.
Note: Subscript zero denotes the property value at standard atmospheric conditions.

This availability balance is applied to the internal combustion engine operating cycle as follows. A first-law-based cycle analysis of the type described above in this section (14.4) is used to define the variation in working fluid thermodynamic state, and the work, heat, and mass transfers that occur in each of the processes that make up the total engine cycle. Integration of the availability balance over the duration of each process then defines the magnitude of the availability destruction that occurs during that process.

To illustrate this procedure, consider the operating cycle of a 10-liter six-cylinder turbocharged and aftercooled direct-injection four-stroke cycle diesel engine, operating at its rated power and speed of 224 kW and 2100 rev/min. The variations in temperature, energy, and entropy are determined with a first-law-based analysis. Figure 14.23 shows the *T-s* diagram for the working fluid as it goes through the sequence of processes from air inlet from the atmosphere (state 1) to exhaust gas exit to the atmosphere (state 10).[46] The incoming air is compressed (with some irreversibility) in the turbocompressor to state 2 and cooled with an aftercooler to state 3. The air at state 3 is drawn into the cylinder and mixed (irreversibly) with residual gases until, at the end of the intake process, the cylinder gases are represented by state 4. That mixture is subsequently compressed (with modest heat loss) to state 5. Fuel addition commences close to state 5; subsequent burning increases the combustion chamber pressure and temperature along the line 5-6. At 6 the heat release, heat transfer, and volume change rate are such that the maximum cylinder pressure is reached (a few degrees after TC). From 6 to 7, combustion continues to completion, the burned gases continue to expand doing work on the piston and losing heat to the walls. At state 7 the exhaust valve opens initiating a rapid pressure

equilibration with the exhaust manifold to a pressure corresponding to point 8. Gases are expelled from the cylinder into the exhaust manifold. After the intake valve opens, in-cylinder residual gases mix with incoming air at state 3 to yield gases at state 4 to complete the cycle. The exhaust gases that have been expelled from the cylinder experience additional thermodynamic losses and can be represented by state 9. These gases then pass through the turbocharger turbine to state 10 to provide the work needed to drive the compressor.

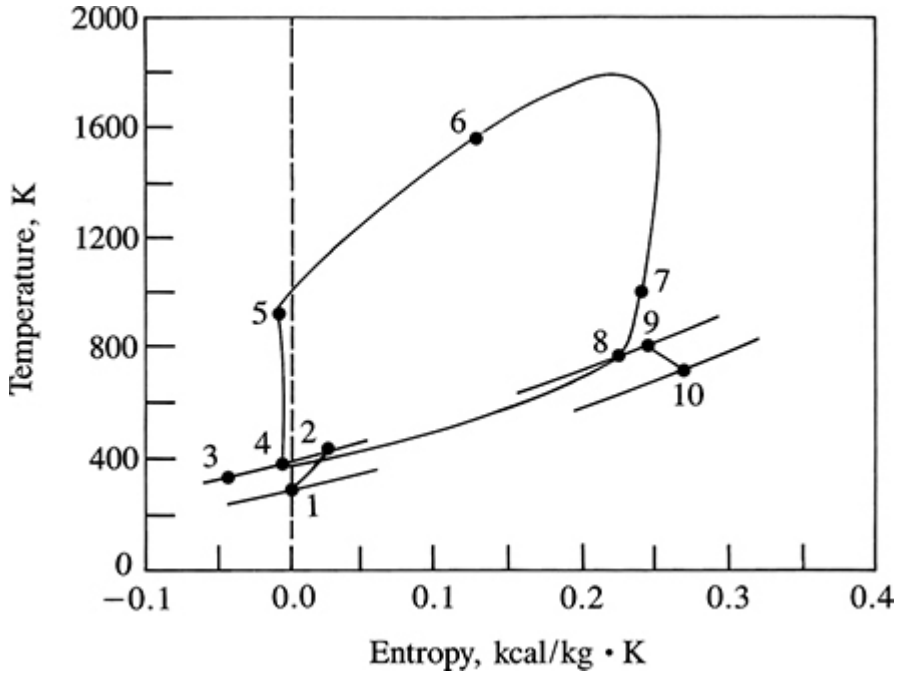


Figure 14.23 *T-s* diagram for the working fluid as it goes through the sequence of processes from intake to exhaust in a turbocharged aftercooled DI diesel engine. The 10-liter six-cylinder engine is operated at its rated power (224 kW) and speed (2100 rev/min). The text relates the processes occurring to the numbered end states. [46]

A first-law and second-law analysis of a naturally-aspirated diesel engine are compared in Table 14.3. Also shown is a second-law analysis of a turbocharged version of this engine. These results illustrate the value of defining the losses in availability that occur in each process.

TABLE 14.3 Comparison of first- and second-law analysis for six-cylinder 14-liter naturally-aspirated and turbocharged diesel engine at 2100 rev/min [47]

	Naturally-aspirated		Turbocharged
	First law, % fuel energy	Second law, % fuel availability	
Indicated work*	40.3	39.1	43.9
Combustion loss	—	15.9	19.2
Cylinder heat transfer	25.1	21.4	17.6
Intake valve throttling	—	0.7	0.7
Exhaust valve throttling	—	2.5	2.3
Loss in compressor	—	—	1.4
Loss in turbine	—	—	0.8
Exhaust to ambient	34.6	20.4	14.1
Total	100.0	100.0	100.0
Brake power, kW	185	185	220

*Note that the indicated work for the second-law balance is a lower percentage than for the first-law balance. This occurs because the availability of the fuel is 1.0317 times the fuel's heating value.

Consider the first- and second-law analysis results for the naturally-aspirated engine. While 25.1% of the fuel energy leaves the combustion chamber in the form of heat transfer, the availability transfer corresponds to 21.4% of the fuel's availability. It is this latter number that indicates the maximum amount of the heat transfer that can be converted to work. The table shows that 34.6% of the fuel energy is carried out of the engine in the exhaust gases. However, the second-law analysis shows that the exhaust contains only 20.4% of the available energy of the fuel. The ratio of these quantities shows that only about 60% of the exhaust energy can be converted to work using ideal thermodynamic devices.[f] The exhaust gas leaves the system in a high-temperature ambient-pressure state, and therefore has high entropy (relative to the p_0, T_0 reference state). This, via the gas-transfer equation in Table 14.2, reduces the available energy of the exhaust gas stream.

The quantity referred to as combustion loss in Table 14.3 is determined from an availability balance for the combustion chamber over the duration of

the combustion period. The "availability destroyed" term in Eq. (14.51) then represents the deviation of the actual combustion process from a completely reversible process. The second-law analysis shows that the availability loss associated with the combustion irreversibilities is 16 or 19% of the fuel's availability. This loss depends on the overall equivalence ratio at which the engine is operating, as indicated in Fig. 5.17. Combustion of leaner air/fuel ratios would give a higher fractional availability loss due to mixing of the fuel-air combustion products with the increased amounts of excess air and the lower bulk temperature, as evidenced by the turbocharged engine value.

Overall, the most important point emerging from this comparison is that the work-producing potential of the heat loss to the combustion chamber walls and the exhaust mass flow out of the engine is not as large as the magnitude of the energy transferred out of the cylinder: some of these energy transfers, even with ideal thermodynamic work-producing devices, must ultimately be rejected to the environment as heat.

A comparison of the second and third columns in Table 14.3, both obtained with a combined first- and second-law analysis, illustrates how turbocharging improves the performance of a naturally-aspirated engine. The brake fuel conversion efficiency of the turbocharged engine is considerably improved—from 33.9 to 39.2%. The table indicates that through turbocharging, the availability transfers associated with the heat loss and exhaust gas flow are reduced from 41.8 to 31.7% (a difference of 10 percentage points), while the combustion and added turbomachinery availability losses increase from 15.9 to 21.4% (a difference of about 5 percentage points). By turbocharging, advantage has been taken of the following changes. While the leaner air/fuel ratio operation of the turbocharged engine increases the combustion availability losses due to the use of a greater portion of the chemical energy of the fuel to mix with and heat excess air, the lower burned gas temperature this produces results in reduced heat losses and lower cylinder exhaust temperature. In addition, the turbocharger transfers available energy from the cylinder exhaust to the inlet air. The reduced heat loss and lower final exhaust availability level give a substantial performance improvement. [47]

To interpret the second-law analysis results, one must remember that the desired output is brake work and increases in this quantity (for a given fuel flow) represent improved performance and efficiency. All other availability terms represent losses or undesirable transfers from the system; decreasing

these terms constitutes an improvement. These undesirable available energy transfer and destruction terms fall into five categories: (1) heat transfer; (2) combustion; (3) fluid flow; (4) exhaust to ambient; and (5) mechanical friction. The available energy flows identified as heat transfer represent the summation of all availability transfers that occur due to heat transfers. The most significant of these are the in-cylinder and aftercooler heat rejection. The combustion loss represents the amount of available energy destroyed due to irreversibilities occurring in releasing the chemical potential of the fuel as thermal energy and mixing the combustion products with any excess air. The fluid flow losses include the available energy destroyed within the working fluid in the compressor, aftercooler, intake valve, exhaust valve, exhaust manifold, and turbine due to fluid shear and throttling. The availability destroyed due to lubricant shear and mechanical rubbing friction is separate from these working-fluid process availability losses.

Figure 14.24 shows the availability transfers or losses in each of these categories for a turbocharged six-cylinder 10-liter displacement direct-injection diesel engine, expressed as a percentage of the fuel availability, as a function of engine load. The percentage of fuel availability associated with the heat transfers varies little over the load range. The combustion loss increases from 22 to 33% as load is decreased due to the increasingly lean operation of the engine. Fluid friction losses, as a percentage, increase slightly as load increases due to larger mass flow rates. Since total engine friction is approximately constant in absolute magnitude, its relative importance increases drastically as the brake output goes to zero. Exhaust flow available energy decreases from 12 to 8% as load is decreased from 100 to 0%.

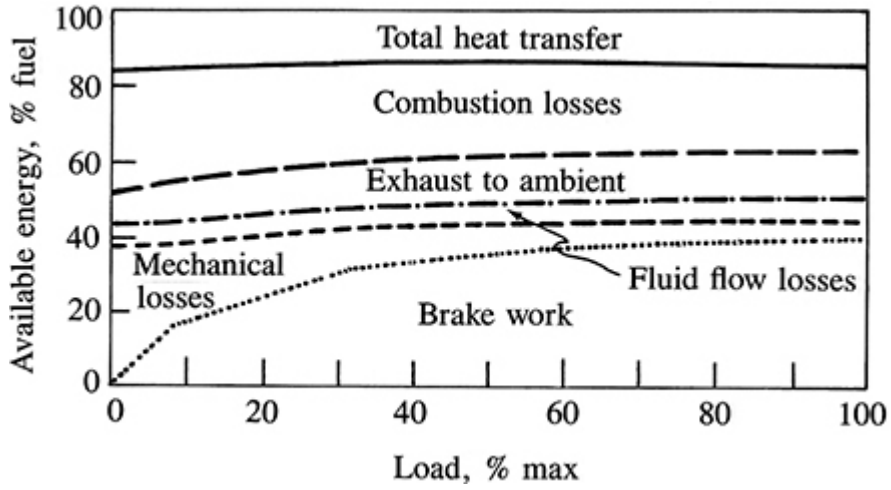


Figure 14.24 Distribution of available energy into major categories for the engine of Fig. 14.23 as a function of engine load.[46]

The effect of varying engine speed (at full load) is shown in Fig. 14.25. The availability associated with heat transfer changes over the speed range from 16 to 21%: more time during each cycle is available for heat transfer at lower speeds. Fluid flow and friction losses decrease slightly with decreasing speed. Other availability losses remain essentially constant as a percentage of the fuel's availability. [46]

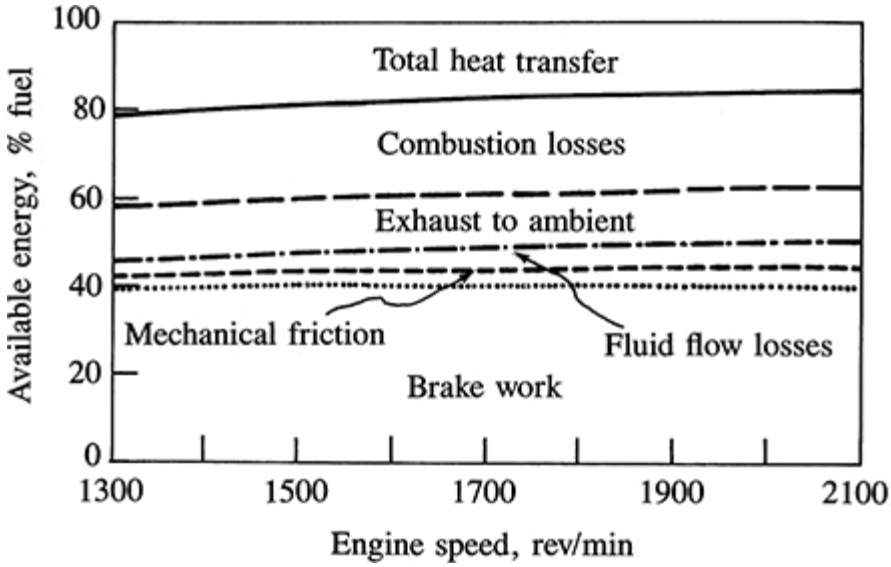


Figure 14.25 Distribution of available energy into major categories for the engine of Fig. 14.23 as a function of engine speed.[46]

14.5 FLUID-MECHANIC-BASED MULTI-DIMENSIONAL MODELS

14.5.1 Basic Approach and Governing Equations

The prediction of the details of the flow field within engines, and the heat-transfer and combustion processes that depend on those flow fields, by numerical solution of the governing conservation equations has become an important engineering goal. Such computational-fluid-dynamic (CFD) based modeling approaches have been developing for several decades. They are now able to predict the complex fluid flow behavior in engine components with realistic geometries. While the overall dynamic behavior of intake and exhaust flows can usefully be studied with one-dimensional unsteady fluid-dynamic computer calculations (see Sec. 14.3.4), flows within the cylinder and in the intake and exhaust ports are usually inherently unsteady and three dimensional. Increases in computing power, and the development of better-grounded turbulence and combustion models now make realistic three-dimensional calculations feasible. Gas-flow patterns can be predicted best; predictions of fuel spray behavior are less complete, and fundamentally based combustion and emissions calculations still present some accuracy challenges.

These CFD-based engine analysis codes solve the partial differential equations for conservation of mass, momentum, energy, and species concentrations. To apply digital computers to the solution of a continuum problem (such as the flow field inside the cylinder), the continuum must be represented by a finite number of discrete elements. The usual method of discretization is to divide the region of interest into a large number of small zones or cells. These cells form a grid or mesh which serves as a framework for constructing finite volume approximations to the governing partial-differential equations. The time variable is similarly discretized into a sequence of small time steps, and the transient solution is "marched out" in time: the solution at time t_{n+1} is calculated from the known solution at time t_n. Three-dimensional formulations of the finite-difference equations are required for most practical engine calculations.

The principal components of these multi-dimensional engine flow models are the following: [48]

- The mathematical equations used to describe the flow processes. Especially important is the turbulence model, which describes the small-scale features of the flow which are not readily accessible to direct calculation.
- The discretization procedures used to transform the differential equations of the mathematical model into algebraic relations between discrete values of velocity, pressure, temperature, etc., located on a computing mesh which conforms to the geometry of the combustion chamber with its moving valves and piston.
- The solution algorithm whose function is to solve the algebraic equations.
- The computer codes which translate the numerical algorithm into computer language and also provide easy interfaces for the input and output of information.

The basic equations for in-cylinder flow calculation methods are the differential equations expressing the conservation laws of mass, momentum (the Navier-Stokes equations—a set of three), energy, and species concentrations. These equations, in the above order, may be written:

$$\frac{D}{Dt}\begin{pmatrix}1\\ u_i\\ e\\ Y_\alpha\end{pmatrix}=\begin{pmatrix}0\\ 0\\ Q\\ S_\alpha\end{pmatrix}-\frac{\partial}{\partial x_j}\begin{pmatrix}0\\ \tau_{ij}\\ q_j\\ J_{\alpha i}\end{pmatrix} \tag{14.52}$$

The first term (on the right of the equals sign) gives the source terms, the second term the diffusive transport. The *D/Dt* operator provides the convective transport terms, and is

$$\frac{Df}{Dt}\equiv\frac{\partial(\rho f)}{\partial t}+\frac{\partial}{\partial x_j}(\rho u_j f) \tag{14.53}$$

Here, ρ is the density, u_i the ith velocity component, e the internal energy per unit mass, and Y_a the concentration of species α per unit mass.

In the IC engine context, the thermal energy source term Q involves a viscous dissipation term and source terms arising from chemical reaction of the fuel-air mixture. Both Q and the species source term, S_a, will depend upon the chemical rate equations, which must be known to close the problem. Note that diffusion of the various species contributes to the diffusive flow of internal energy, q_j, in addition to conductive heat diffusion.

The fact that turbulent flows exhibit important spatial and temporal variations over a range of scales (dictated at the upper end by chamber dimensions and at the lower end by viscous dissipative processes; see Sec. 8.2.1) makes direct numerical solution of these governing equations impractical for complex engine flows. Recourse must therefore be made to some form of averaging or filtering which removes the need for direct calculation of the small-scale motions. Two approaches have been developed for dealing with this turbulence modeling problem: full-field modeling (FFM), sometimes called statistical flux modeling; and large-eddy simulation (LES) or subgrid-scale simulation. In FFM, one works with the partial differential equations describing suitably averaged quantities, using the same equations everywhere in the flow. For periodic engine flows, time averaging must be replaced by ensemble or phase averaging (see Sec. 8.2.1). The variables include the velocity field, thermodynamic state variables, and various mean turbulence parameters such as the turbulent kinetic energy, the turbulent stress tensor, etc. In FFM, models are needed for various averages of the turbulence quantities. These models must include the contributions of all scales of turbulent motion.

Large-eddy simulation (LES) is an approach in which one actually calculates the large-scale three-dimensional time-dependent turbulence structure in a single realization of the flow. Thus, only the small-scale turbulence need be modeled. Since the small-scale turbulence structure is more isotropic than the large-scale structure and responds rapidly to changes in the large-scale flow field, modeling of the statistical fluxes associated with the small-scale motions is a simpler task than that faced in FFM where the large-scale turbulence must be included.

An important difference between FFM and LES is their definition of "turbulence." In FFM the turbulence is the deviation of the flow at any instant from the average over many engine cycles of the flow at the same point in space and oscillation phase [i.e., the fluctuation velocity defined by Eq.

(8.16) or (8.18)]. Thus, FFM "turbulence" contains some contribution from cycle-by-cycle flow variations. LES defines turbulence in terms of variations about a local average; hence in LES turbulence is related to events in the current cycle. [49]

14.5.2 Turbulence Models

In *full-field modeling* (FFM), equations for the averaged variables are formed from Eqs. (14.52) . With periodic engine flows, phase or ensemble averaging must be used (see Secs. 8.2.1 and 8.2.2). Since the flow during the engine cycle is compressed and expanded, mass-weighted averaging (called Favre averaging) can be used to make the averaged compressible-flow equations look almost exactly like the averaged equations for incompressible flows. The combined ensemble-Favre averaging approach works as follows. [49]

We denote the phase-averaging process by $\{\ \}$, that is:

$$\{\rho(\mathbf{x},t)\} = \lim_{N\to\infty} \frac{1}{N} \sum_{n=1}^{N} \rho(\mathbf{x}, t + n\tau) \tag{14.54}$$

where τ is the cycle period. We also write $\{\rho\} = \tilde{\rho}$, and decompose ρ into $\rho = \tilde{\rho} + \rho'$. The mass-weighted phase-averaged quantities (indicated by an overbar) are defined by

$$\rho(\mathbf{x},t)\bar{f}(\mathbf{x},t) = \lim_{N\to\infty} \sum_{n=1}^{N} \rho(\mathbf{x}, t + n\tau) f(\mathbf{x}, t + n\tau) \tag{14.55}$$

where all flow variables (except density and pressure) have been decomposed as $f = \bar{f} + f'$. Note that $\{\rho'\}$ is zero, $\{\bar{f}\} = \bar{f}$, the mass-weighted phase average of f' is zero, but $\{f'\}$ is not zero. With these definitions:

$$
\begin{aligned}
\{\rho f\} &= \tilde{\rho}\bar{f} \\
\{\rho f'\} &= 0 \\
\{\rho fg\} &= \tilde{\rho}\left(\bar{f}\bar{g} + \overline{fg'}\right) \\
\{\rho fgh\} &= \tilde{\rho}\left(\bar{f}\bar{g}\bar{h} + \bar{f}\,\overline{g'h'} + \bar{g}\,\overline{f'h'} + \bar{h}\,\overline{f'g'} + \overline{f'g'h'}\right)
\end{aligned}
\tag{14.56}
$$

Phase-averaging Eq. (14.54) , one obtains[49]

$$
\frac{\bar{D}}{Dt}\begin{pmatrix} 1 \\ \bar{u}_i \\ \bar{e} \\ \bar{Y}_\alpha \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \\ \{Q\} \\ \{S_\alpha\} \end{pmatrix} - \frac{\partial}{\partial x_j}\begin{pmatrix} 0 \\ \{\tau_{ij}\} \\ \{q_j\} \\ J_{\alpha j} \end{pmatrix} - \frac{\partial}{\partial x_j}\begin{pmatrix} 0 \\ \tilde{\rho}\overline{u_i'u_j'} \\ \tilde{\rho}\overline{e'u_j'} \\ \tilde{\rho}\overline{Y_\alpha'u_j'} \end{pmatrix}
\tag{14.57}
$$

where

$$
\frac{\overline{Df}}{Dt} = \frac{\partial}{\partial t}(\tilde{\rho}\bar{f}) + \frac{\partial}{\partial x_j}(\tilde{\rho}\bar{u}_j\bar{f})
\tag{14.58}
$$

The terms on the left-hand side in Eq. (14.57) involve only the solution variables $\tilde{\rho}, \bar{u}_i, \bar{e}$, and $\bar{Y}_\alpha$, and hence require no modeling. However, all of the terms on the right, particularly the last terms that represent turbulent transport, involve turbulent fluctuation quantities and must be modeled in terms of the solution variables. The source terms $\{Q\}$ and $\{S_a\}$ present special difficulties to the engine modeler. Due to the exponential dependence of the heat release Q on temperature, $\{Q\}$ will be strongly influenced by temperature fluctuations. These issues are discussed more fully in Secs. 14.5.5 and 14.5.6.

The momentum equations contain terms, $\tilde{\rho}\overline{u_i'u_j'}$, which represent turbulent stresses (often called the Reynolds stresses). These terms must be modeled with additional equations before the set of equations, (14.57), is "closed" and can be solved. The most widely used *turbulence model* or equation set is the *k-ε model.* [49–52] This assumes a Newtonian relationship between the turbulent stresses and mean strain rates, and computes the (fictitious) turbulent viscosity appearing in this relationship from the local turbulent

kinetic energy $k\ (= u_i u_i\ /2)$ and its dissipation rate ε. An equation governing k can be developed by multiplying the u_i equation in Eq. (14.52) by u_i, subtracting from this the equation formed by multiplying the $\overline{u}_i$ equation in Eq. (14.57) by $\overline{u}_i$, and phase-averaging the result. The equation so obtained is

$$\frac{\overline{D}\overline{k}}{Dt} = \tilde{\rho}(P - \varepsilon) - \frac{\partial}{\partial x_j} J_k \tag{14.59a}$$

where P is the rate of turbulence production per unit mass

$$P = \overline{u_i' u_j'} \frac{\partial \overline{u}_i}{\partial x_j} \tag{14.59b}$$

and J_k represents diffusive transport.

In the most commonly used two-equation k-ε model, all the unknown turbulence quantities are modeled in terms of the turbulent velocity scale $k^{1/2}$ and the turbulence length scale $k^{3/2}/\varepsilon$ obtained from the definition of the energy dissipation rate, via

$$\varepsilon \propto \frac{k^{3/2}}{l} \tag{14.60}$$

The rationale is that the rate of energy dissipation is controlled by the rate at which the large eddies feed energy to the smaller dissipative scales which in turn adjust to handle this energy. [49]

A turbulent viscosity μT is defined:

$$\mu_T = \frac{C_0 \tilde{\rho} k^2}{\varepsilon} \tag{14.61}$$

where C_0 is a model constant. The turbulent stress terms appearing in Eqs. (14.57) and (14.59) are then modeled in a quasi-Newtonian manner:

$$\tilde{\rho}\overline{u_i' u_j'} = \frac{2}{3}\tilde{\rho} k \delta_{ij} + \frac{2}{3}\mu_T \nabla \cdot \overline{\mathbf{u}} \delta_{ij} - 2\mu_T \overline{S}_{ij} \tag{14.62}$$

where $\overline{S}_{ij}$ is the strain rate of the $\overline{u}_i$ field:

$$\overline{S}_{ij} = \frac{1}{2}\left(\frac{\partial \overline{u}_i}{\partial x_j} + \frac{\partial \overline{u}_j}{\partial x_i}\right) \tag{14.63}$$

The viscous-stress terms in the momentum equations are evaluated using a Newtonian constitutive relation. The turbulent-diffusion terms in the various transport equations are modeled using the turbulent diffusivity. The diffusing flux of a quantity φ is given by

$$J_{\phi_i} = \frac{\mu_T}{\sigma_\phi}\frac{\partial \phi}{\partial x_i} \tag{14.64}$$

where $s\varphi$ is the turbulent Prandtl number for φ.

The model is completed with a transport equation for ε. An exact equation can be developed by suitable manipulation of the Navier-Stokes equations. All ε equation models are of the form [49]

$$\frac{\overline{D}\varepsilon}{Dt} = W - \frac{\partial H_i}{\partial x_i} \tag{14.65}$$

where W is the source term and H_i is the diffusive flux of $\tilde{\rho}\varepsilon$ which is modeled similarly to the other diffusion terms. The appropriate form of W is the subject of much debate. For an incompressible flow, W can be modeled adequately by

$$W = \left(C_1\frac{P}{\varepsilon} - C_2\right)\frac{\tilde{\rho}\varepsilon^2}{k} \tag{14.66}$$

C_1 and C_2 are constants: the C_2 term produces the proper behavior of homogeneous isotropic turbulence and the C_1 term modifies this behavior for homogeneous shear. However, for a flow with compression and expansion, an additional term in Eq. (14.66) is needed to account for changes in ε produced by dilation. Several forms for this additional term have been proposed [49, 50, 53] (e.g., $C_3\tilde{\rho}\varepsilon\nabla\cdot\overline{\mathbf{u}}$) and compared. [51] The goal is to construct

a W that predicts the appropriate physical behavior under the relevant engine conditions. While different choices for modeling these terms do affect the results (especially the behavior of the turbulence length scale during the cycle [50]), the predictions of mean flow and turbulence intensity do not differ significantly. [51]

One FFM that has been applied to engines is the *Reynolds stress model* (RSM) which, in its most general form, comprises seven simultaneous partial differential equations for the six stress components and the dissipation rate ε. This obviously imposes a much greater computing burden compared with the two-equation k-ε model. The results available indicate that RSM predictions of the flow field are closer to corresponding measured data than the simpler k-ε model predictions. [52]

The LES approach to turbulence modeling has also been applied to engines. Since here one calculates the large-scale three-dimensional time-dependent flow structure directly, only the turbulence smaller in scale than the grid size need be modeled. Hence these are often referred to as subgrid-scale models. A new dependent variable q, which represents the kinetic energy per unit mass of the turbulent length scales that are too small to be resolved in the mesh, is introduced. This variable satisfies a transport equation which contains terms for production and decay of q and for its convection and diffusion. This equation has the form:

$$\frac{\partial}{\partial t}(\rho q)+\nabla\cdot(\rho q\mathbf{u})=-\frac{2}{3}\rho q\nabla\cdot\mathbf{u}+\boldsymbol{\sigma}:\nabla\mathbf{u}+\nabla\cdot(\mu\nabla q)-C\rho L^{-1}q^{3/2}+\dot{W}_s \qquad (14.67)$$

where $\boldsymbol{\sigma}$ is the turbulent stress tensor, μ the turbulent viscosity, C a constant of order unity, and L a characteristic length on the order of twice the mesh spacing. $\dot{W}_s$ is a source term representing the production of turbulence by the motion of fuel droplets in situations where fuel sprays are important.

The physical meaning of the terms in Eq. (14.67) are as follows. The term $\nabla\cdot(\rho q\mathbf{u})$ is the convection of the turbulence by the resolved (large-scale) velocity field. The term $-\frac{2}{3}\rho q\nabla\cdot\mathbf{u}$ is a compressibility term that is the turbulent analog of pdV work. The term $\boldsymbol{\sigma}:\nabla\mathbf{u}$ represents the production of turbulence by shear in the resolved velocity field; $\nabla\cdot(\mu\nabla q)$ is the self-diffusion of the turbulence with diffusivity μ/ρ. The term $-C\rho L^{-1}q^{3/2}$ represents the decay of turbulent energy into thermal energy. This term also appears, with opposite sign, as a source term in the thermal internal energy

equation in place of $\boldsymbol{\sigma}$: $\nabla \mathbf{u}$, which can be thought of as the rate at which kinetic energy of the resolved motions is dissipated by the turbulence. Before it is dissipated, the kinetic energy of the resolved velocity field is first converted into subgrid-scale turbulent energy q, which is then converted into heat by the decay term $C\rho L^{-1} q^{3/2}$. [53]

Under most circumstances, the velocity and temperature boundary layers in an engine cylinder will be too thin to be resolved explicitly with a computing mesh that is practical. However, these layers are important because they determine the wall shear and heat flux which are essential boundary conditions for the numerical simulation, and are of practical importance (see Secs. 8.3 and 12.6.5). Special submodels for these boundary layers, referred to as wall functions, are used to connect the wall shear stresses, heat fluxes, wall temperatures, etc., to conditions at the outer edge of the boundary layer. This removes the need to place grid points within the layer. Since the boundary layers are usually turbulent, the logarithmic "turbulent law of the wall" is commonly used.

14.5.3 Numerical Methodology

The three important numerical features of multi-dimensional methods are: the computational grid arrangement which defines the number and positions of the locations at which the flow parameters are to be calculated; the discretization practices used to transform the differential equations of the mathematical model into algebraic equations; and the solution algorithms employed to obtain the flow parameters from the discrete equations. [54]

Computing Mesh

The requirements of the computing mesh are:

- It fits the topography of the combustion chamber and/or inlet port, including the moving components.
- It allows control of local resolution to obtain the maximum accuracy with a given number of grid points.
- It has the property that each interior grid point is connected to the same number of neighboring points.

The first requirement obviously follows from the need to simulate the effects of changes in engine geometry. The second requirement stems from the fact

that computing time increases at least linearly with the number of mesh points. Thus it is desirable that the mesh allow concentration of grid points in regions where steep gradients exist such as jets and boundary layers. The third requirement comes from the need for the mesh to be topologically rectangular in some transformed space so that efficient equation solvers for such mesh systems can be utilized.

In the engine context the simplest grids are defined by the coordinate surfaces of a cylindrical-polar frame. Such an approach is only feasible if the combustion chamber walls coincide with the coordinate surfaces. This occurs for a restricted number of chamber geometries (e.g., disc and centered cylindrical bowl-in-piston shapes). Even for these, the inlet and exhaust valve circumferences would in general cut across the grid (see Fig. 14.26 *a*). While procedures have been devised for modifying the difference equations for such grids to allow for noncoincident boundaries, the preferred approach is to employ some form of flexible "body-fitting" coordinate frame/grid whose surfaces can be shaped to the chamber geometry, as illustrated in Fig. 14.26 *b* which shows a diesel engine combustion chamber fitted by a mesh which is orthogonal-curvilinear in the bowl. This enables the bowl shape to be accurately represented and the boundary layers on its surfaces to be resolved in greater detail. The region between the piston-crown and cylinder-head surfaces is fitted with a bipolar system which expands and contracts axially to accommodate the piston motion. The orthogonality constraint of this mesh limits its usefulness: the generation of orthogonal meshes for general three-dimensional geometries is time-consuming, and the resulting mesh often far from optimal. These problems can be largely surmounted by "arbitrary" nonorthogonal Lagrangian-Eulerian meshes like that used in KIVA. [54] This has the additional advantage that the mesh points in the swept volume are not constrained to move axially: their motion can be arbitrarily prescribed. [48] Figure 14.26*c* shows the full CFD mesh for a cylinder head and liner of a four-valve per cylinder gasoline direct-injection spark-ignition engine. The complexity of mesh generation is obvious.

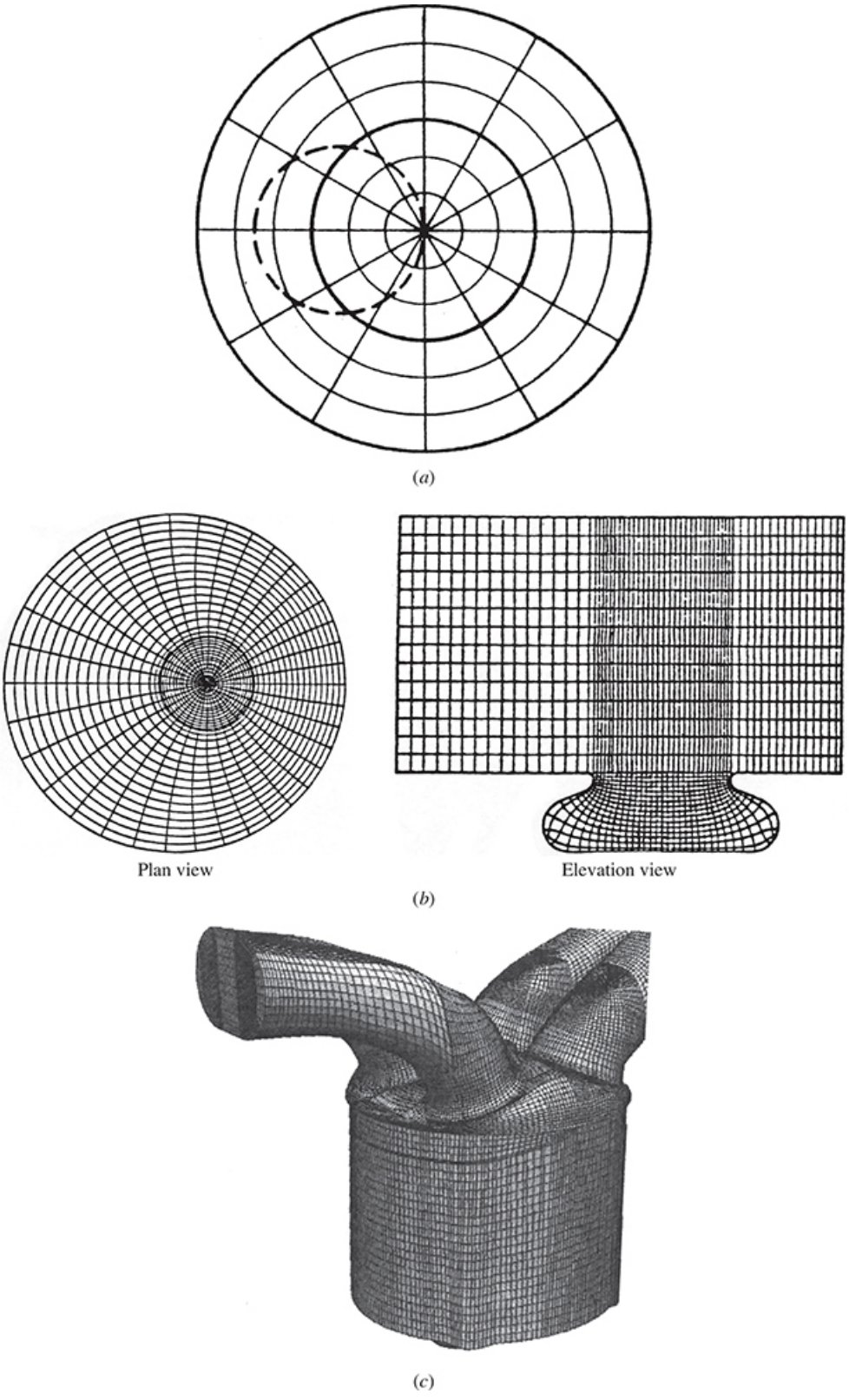


Figure 14.26 Illustrations of the different types of computing mesh arrangements for engine combustion chambers. (*a*) Cylindrical polar mesh;

dashed line shows intake valve head circumference. [55] (*b*) Body-fitted orthogonal curvilinear mesh fitted to DI diesel combustion chamber bowl. [56] (*c*) Cylinder head (with intake and exhaust ports), pent roof combustion chamber, and liner. [57]

Discretization Practices

These multi-dimensional engine flow models are time-marching programs that solve finite-difference approximations to the governing differential equations. The individual cells formed by the mesh or grid serve as the spatial framework for constructing these algebraic finite difference equations. The time variable is similarly discretized into a sequence of small time intervals called time steps: the solution at time t_{n+1} is calculated from the known solution at time t_n. The spatial differencing procedure used is to difference the basic equations in integral form, with the volume of a typical cell used as the control volume and the divergence terms transformed into surface integrals using the divergence theorem. [54]

The discretized equations for any dependent variable φ are of the general form:

$$A_p\phi_p^{i+1} = \sum_n A_n\phi_n^{i+1} + S_{\phi,p}V_p + A_p^i\phi_p^i \tag{14.68}$$

where the *A's* are coefficients expressing the combined influences of convection and diffusion, $S_{\varphi,p}V_p$ is the source integral over the cell volume V_p, the subscript p denotes a typical node point in the mesh, the summation Σ is over its (six) nearest neighbors, and the superscripts $i + 1$ and i denote "new" and "old" values, at times $t + \delta t$ and t, respectively, where δt is the size of the time step. [55]

Historically, all methods involved similar spatial approximations to calculate convective and diffusive transport, using a blend of first-order upwind differencing for the former and second-order central differencing for the latter. Unfortunately, all discretization practices introduce inaccuracies of some kind, and the standard first-order upwind scheme produces spatial diffusion errors which act in the same way as real diffusion to "smooth" the solutions. The magnitude of the numerical diffusion reduces as the mesh

density is increased. A recent development has been the introduction of "higher order" spatial approximations which, in the past, had a tendency to produce spurious extrema. This problem has been overcome by the use of "flux blending" techniques. First-order upwind and higher-order approximations are blended in appropriate proportions to eliminate the overshoots of the latter.

Solution Algorithms

Numerical calculations of compressible flows are inefficient at low Mach numbers because of the wide disparity between the time scales associated with convection and with the propagation of sound waves. While all methods use first-order temporal discretization and are therefore of comparable accuracy, they differ in whether forward or backward differencing is employed in the transport equations leading to implicit or explicit discrete equations, respectively. In explicit schemes, this inefficiency occurs because the time steps needed to satisfy the sound-speed stability condition are much smaller than those needed to satisfy the convective stability condition alone. In implicit schemes, the inefficiency manifests itself in the additional computational labor needed to solve the implicit (simultaneous) system of equations at each time step. This solution is usually performed by iterative techniques.

The computing time requirements of these two approaches scale with the number of equations n and the number of mesh points m, as follows. For explicit methods, computing time scales as nm, but the time step is limited by the stability condition as summarized above. For implicit methods, computing time scales as $n^3 m$ and Δt is only limited by accuracy considerations. The implicit equations that result from forward differencing consist of simultaneous sets for all variables and thus require more elaborate methods of solution. However, they contain no intrinsic stability constraints. Fully iterative solution algorithms for solution of these equation sets are being replaced with more efficient simultaneous linear equation solvers. [55]

14.5.4 Flow Field Predictions

To illustrate the capabilities of multi-dimensional modeling of IC engine flows, examples of such calculations will now be reviewed. A large amount of information on many fluid flow and state variables is generated with each

calculation, and the processing, organization, and presentation of this information are tasks of comparable scope to its generation. Flow field results are usually presented in terms of the gas velocity vectors at each grid point of the mesh in appropriately selected planes. Arrows are usually used to indicate the direction and magnitude (by length) of each vector. Examples of such plots—of the flow pattern in the cylinder during the intake process—are shown in Fig. 14.27. [58] The flow field is shown 60° ATC during the intake stroke. A helical intake port is used to general swirl, and details of the flow through the valve curtain area (see Sec. 6.3.2)—the inlet boundary conditions for the calculation—were determined by measurement. The calculation used a curvilinear, axially expanding and contracting grid with about 16,000 mesh points of the type shown in Fig. 14.26 *b*. It employed a fully iterative solution algorithm with standard upwind differencing and the k-ε turbulence model. Shown in Fig. 14.27 are the velocity components in the plane through the valve and cylinder axis (top left), the perpendicular plane through the valve axis (bottom left), and in a circumferential radial plane halfway between the cylinder head and the piston (right). Large-scale rotating flows are set up as the piston travels downward, created by the conical high velocity flow past the intake valve head.

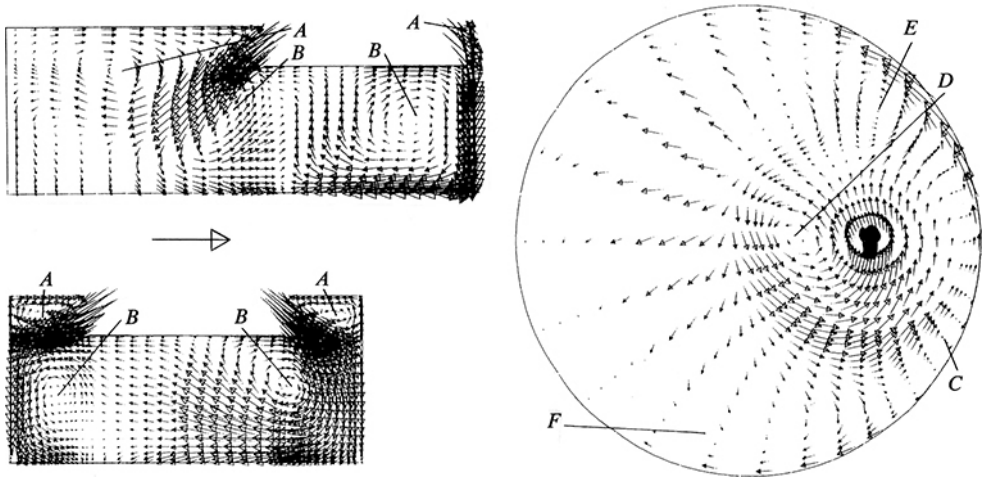


Figure 14.27 Computed velocity field within the cylinder at 60° ATC during the intake stroke. Top left: plane through cylinder and inlet valve axes. Bottom left: orthogonal plane through valve axis. Right: circumferential-radial plane halfway between piston and cylinder head. Reference vector arrow corresponds to velocity of 132 m/s. Letters denote centers of toroidal flow structures. [58]

The major features of the conical jet flow through the inlet valve into the cylinder are apparent (see Sec. 8.1). However, the off-cylinder-axis valve and the swirl generated by the helical port produce substantial additional complexity. The letters on the figures show regions of local recirculation. Regions *A* and *B* correspond to the rotating flow structures observed in simpler geometries (see Fig. 8.3); however, regions *C* through *F* indicate that the swirling motion is far from a solid-body rotation. [58]

Figure 14.28 shows comparisons of three-dimensional predictions of in-cylinder flow fields with data. The computational and experimental geometries have been matched, as have the intake flow velocities through the valve open area, and engine speed. The figure shows predicted and measured mean flow velocities and turbulence intensities within the cylinder, with a conventional inlet port and valve configuration, at 68° ATC during the intake stroke. [59] The experimental values come from LDA measurements (see Sec. 8.2.2). The general features of the mean flow are reproduced by the model with reasonable accuracy, though some details such as the recirculating flow along the cylinder liner toward the head in the symmetry plane are not predicted. The approximate magnitude of the turbulence intensity levels is predicted, but the values within the conical intake jet are underestimated.

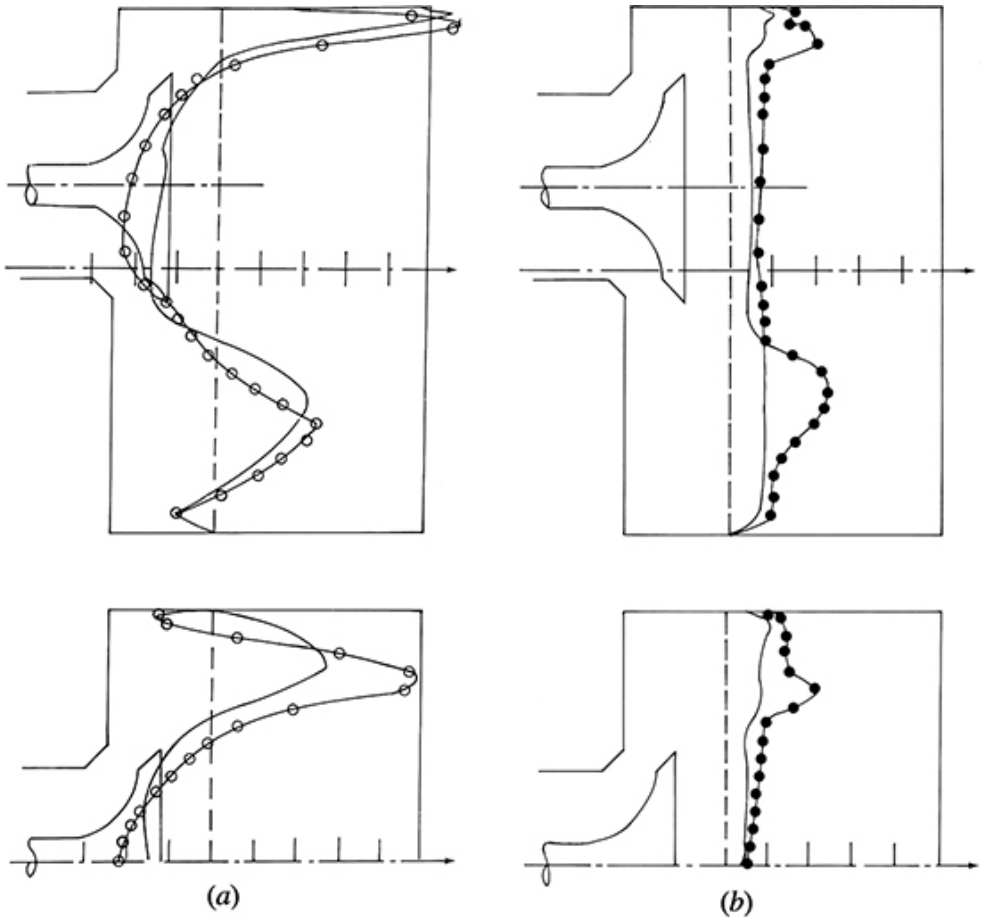


Figure 14.28 Comparison of (*a*) measured and predicted axial velocity profiles and (*b*) measured and predicted turbulence intensity profiles at 68° ATC during the intake stroke. Data: line with points. Predictions: line without points. Each interval on the scale on cylinder axes corresponds to two times the mean piston speed. [59]

Comparative multi-dimensional modeling studies of different turbulence models, [51] differencing schemes, [48, 55] and number of grid points [48] indicate the following. Differences in the form and coefficients of the dilation term in the k-ε turbulence model have only modest effects on flow field predictions. Higher-order turbulence models do provide improved accuracy. [55] Both mesh refinement, more finely spaced grid points, and use of higher-order differencing schemes do improve the accuracy of the predictions, often of course with substantial increases in computing requirements. [48]

Examples of predictions of other types of engine flow processes are the following. Squish flows into bowl-in-piston combustion chambers with swirl have been extensively analyzed. Figure 14.29 shows the flow field into and within a bowl-in-piston high-speed direct-injection diesel. The figure shows the modeled velocity distributions in the radial-axial plane with the shading/color representing the tangential/azimuthal velocity component from the end of compression to end of combustion, −10 to 40 crank-angle degrees (CAD), relative to TC. Combustion commences about 2° ATC and heat release is three-quarters completed by about 15° ATC. The axes of the six fuel sprays are 17.5° downward directed (with piston at TC) into the farthest lower right side of the bowl. The *r*-*z* plane shown corresponds to the axis of one of the fuel sprays.

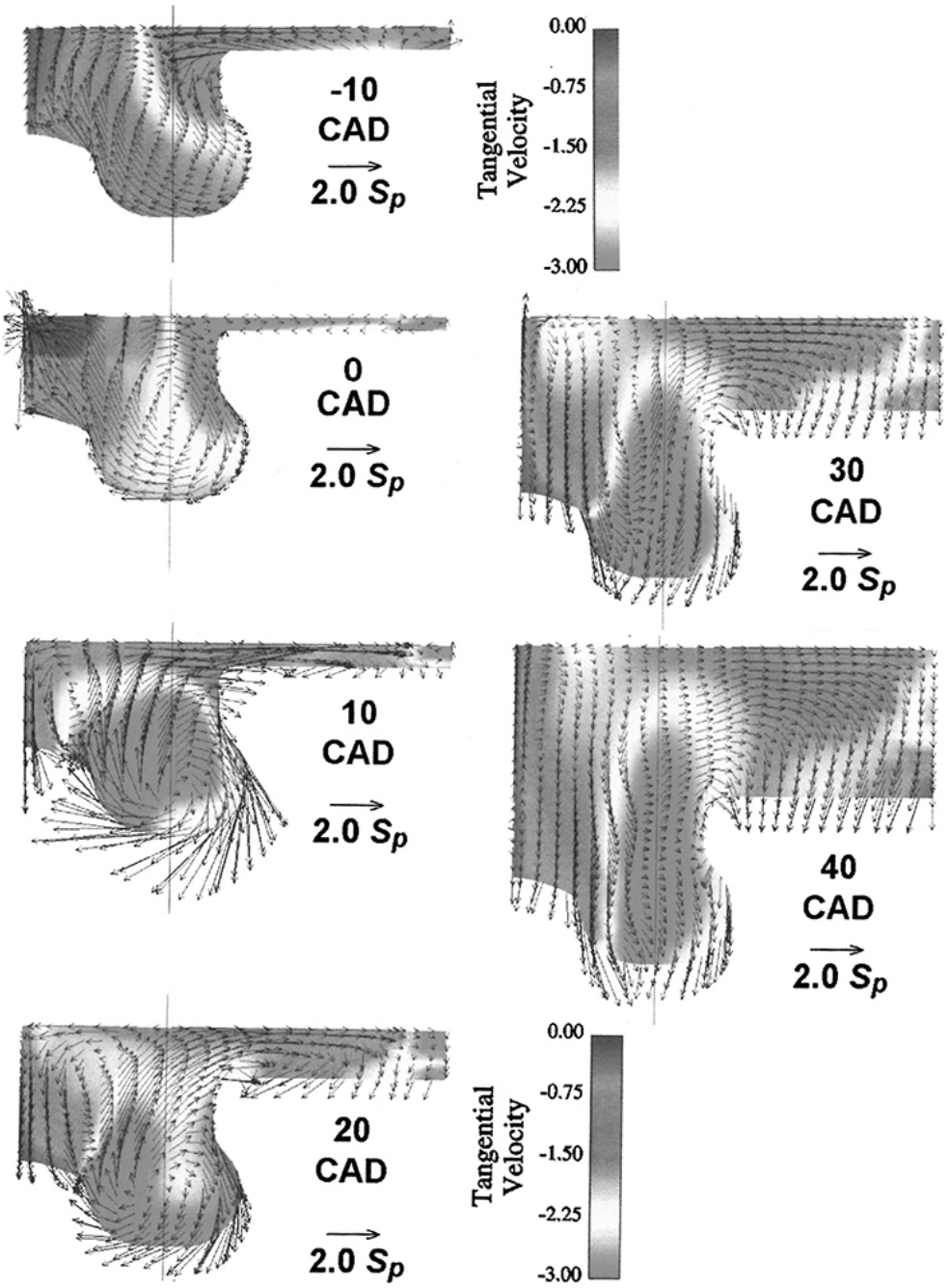


Figure 14.29 Flow velocity vectors in an *r*-*z* plane aligned with one (of six) fuel spray axis. The magnitude of the tangential/azimuthal velocity (in units of $\overline{S}_p$) is shown by the grey scale). Start of fuel injection, 1.5 crank-angle degrees (CAD) before TC; end of injection, 2.5 CAD after TC; start of combustion, 2 CAD after TC; combustion three-quarters complete by 15 CAD after TC, and ends about 35 CAD after TC. [60]

At −10 CAD, the strong radially inward squish motion is clear: at this crank angle, the upward piston-motion generated squish flow reaches its maximum (see Fig. 8.23). This inward flow does not penetrate to the cylinder/piston bowl axis, it is slowed and deflected down, then outward, by the pressure field generated by the high flow swirl within the bowl (some two to three times the mean piston speed). Within the bowl, a rotating flow (clockwise in this representation) in the *r-z* plane is created. With the radially inward squish flow decreasing rapidly as the piston approaches TC, the fuel injection spray supplements this *r-z* plane rotation within the bowl. The fuel spray entrains air and is seen to transport low angular momentum fluid outward from the upper center region of the piston bowl. The combustion-produced gas expansion within the bowl (0 to 10 CD) accelerates the gas rotation and increases the swirl. As the piston travels down the cylinder, the high swirl region within the bowl increases in size: the rotational velocity about cylinder axis of the gases that leave the bowl and move radially outward (the reverse squish flow) significantly decreases since, while their angular momentum is conserved, their radius increases. [60]

The turbulence intensity u' predicted by current CFD engine models generally shows moderate agreement with root-mean-square velocity measurements but with, at times, significant deviations. In the study shown in Fig. 14.29, the rms radial and tangential velocity components during injection and combustion, in the top portion of the bowl, were about 0.6 $\overline{S}_p$ and decreasing modestly. The turbulence was anisotropic. The measurements carried out in parallel with the modeling, showed some evidence of increased rms velocity fluctuations during and after combustion above and within the piston bowl possibly caused by the gas expansion on burning within the diesel diffusion flame. [60] These levels of turbulence intensity are significant, 2 to 3 m/s, would scale with engine speed, and obviously influence mixing rates between fuel and air, and combustion products with the excess air (post combustion) in the cylinder. Turbulence intensity results are often displayed on contour plots. There is a correspondence between high-velocity regions generated by the squish flow (Fig. 14.29) and higher turbulence intensities. A substantial variation in intensity throughout the bowl is usually predicted. Computer-generated three-dimensional perspective views of the velocity field prove valuable.

An alternative method of displaying multi-dimensional model results, especially from three-dimensional calculations, is through particle traces. Infinitesimal particles are placed at key locations in the flow field at a given crank angle (e.g., at the start of the process of interest) and their trajectories are computed from the velocity field as a function of time. The particle traces illustrate the flow streamlines within the intake, port, valve, and cylinder. The example of particle traces (Fig. 14.30) within the cylinder during the intake stroke indicates the complexity of swirling flows with realistic intake port and valve geometries. [61] The figure shows the paths traced out by six particles, initially evenly spaced around the valve curtain area at TC at the start of the intake process, during the intake stroke with a tangentially directed inlet port. While all the particles follow a helical path within the cylinder, the steepness of these paths varies depending on the initial location of each particle.

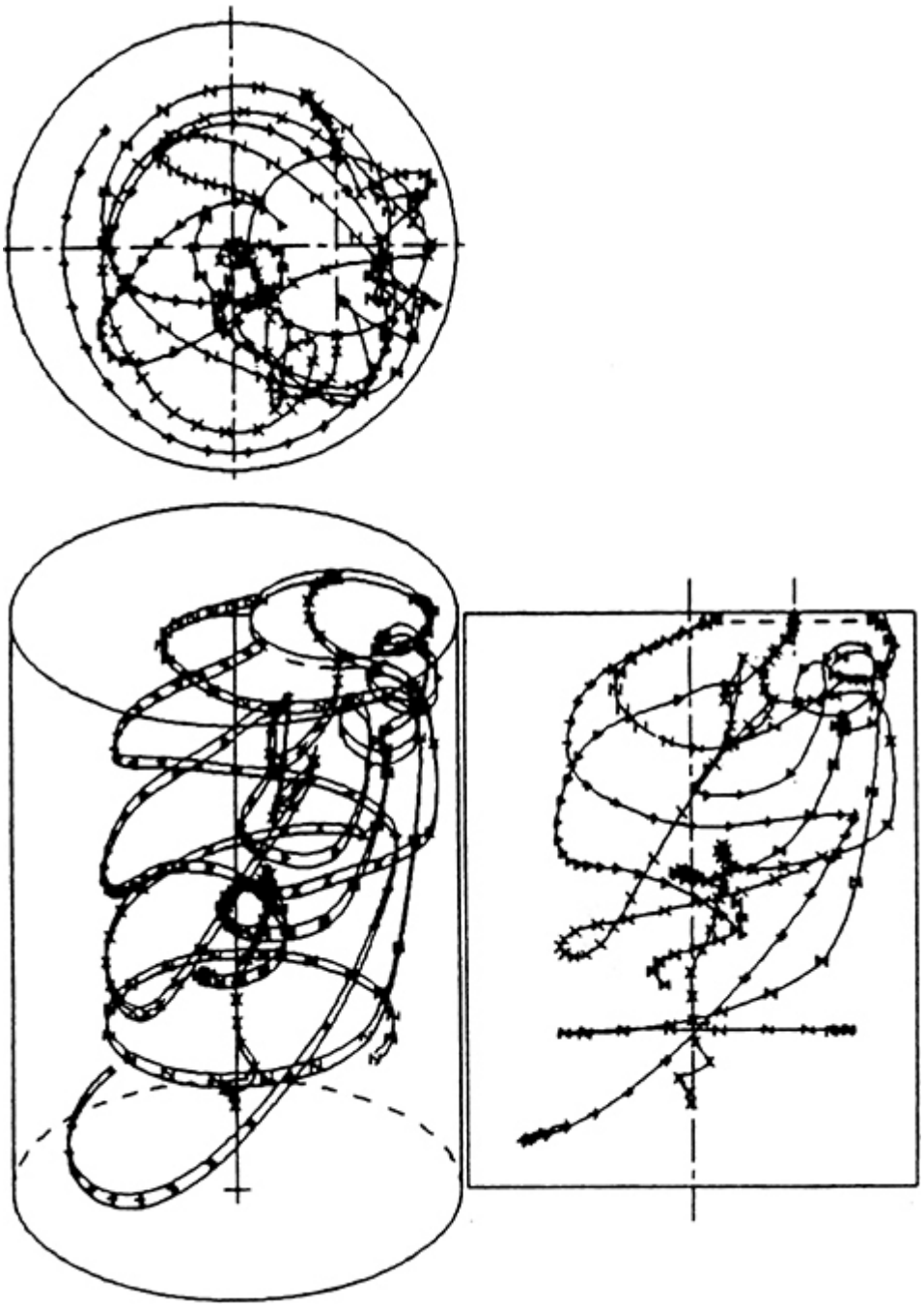


Figure 14.30 Computed trajectories traced out during the intake stroke by six gas particles initially evenly spaced around the valve curtain area at TC at the start of intake process, with a tangentially directed inlet port. Cylinder shown with piston at BC, at the end of the intake stroke. [61]

Multi-dimensional models also provide local mixture composition information. Studies have been done of the mixing of fuel with air in sprays,

of two-stroke cycle scavenging flows (see Ref. 62), and of the mixing between fresh mixture and residual gases in four-stroke cycle engines (e.g., Ref. 63). Figure 14.31 shows how the mixing between fresh fuel and air, and residual gases, proceeds during the intake and compression strokes of a spark-ignition engine four-stroke cycle. Concentrations (defined as fresh mixture mass/ total mixture mass) at different locations within the cylinder are plotted against crank angle ($z = 2$ is near the head, $z = 7$ near the piston; $y = 2$ is near the cylinder axis, $y = 7$ near the cylinder liner). A relatively long time is required for the fresh and residual gases to mix and at 30° BTC there is still several percent nonuniformity. At part load with its higher residual fraction, one would expect these differences to be larger. [63]

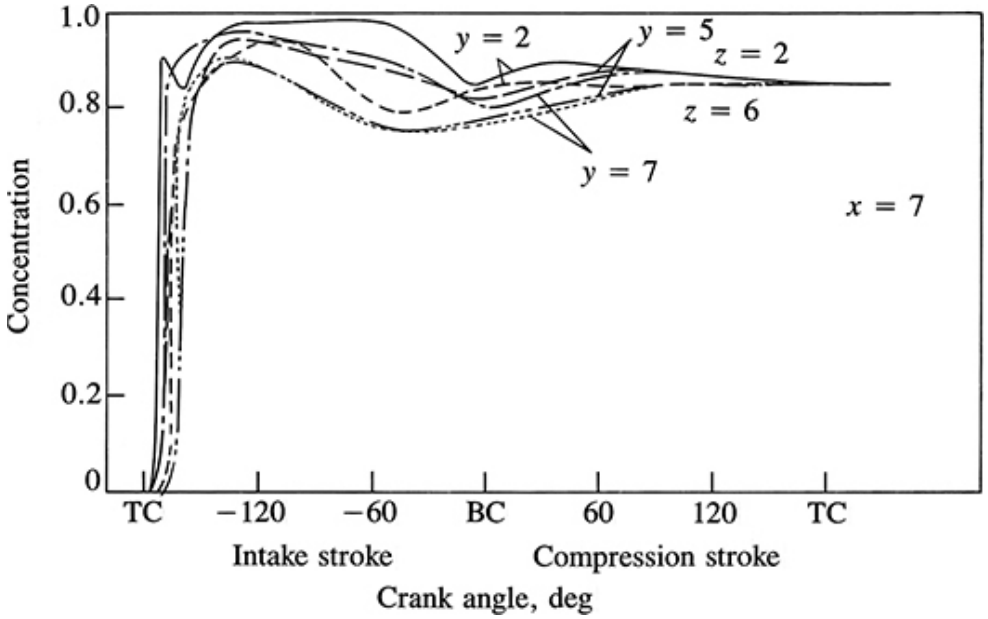


Figure 14. 31 Computed concentration distribution of fresh fuel-air mixture and residual gas within the cylinder during the intake and compression stroke of a spark-ignition engine. Concentration expressed as fresh mixture mass/total mixture mass, $z = 2$ is near the cylinder head, $z = 7$ near the piston; $y = 2$ near the cylinder axis, $y = 7$ near the cylinder liner; $x = 7$ along the radius passing beneath the inlet valve. 2000 rev/min and wide-open throttle. [63]

14.5.5 Fuel Spray Modeling

The physical behavior of liquid fuel sprays when injected into the engine cylinder, as occurs in DI stratified-charge spark-ignition and diesel engines, has already been described in Secs. 9.3.5 and 10.4. Here the current status of models for such spray behavior which are used with multi-dimensional models of gas motion within the cylinder is reviewed. Direct fuel-injected internal combustion engines present a particularly difficult problem for numerical simulation. The fuel spray produces an inhomogeneous fuel-air mixture: the spray interacts with and strongly affects the flow patterns and temperature distribution within the cylinder. The fuel is injected as liquid, it atomizes into a large number of small droplets with a wide spectrum of sizes, the droplets disperse and vaporize as the spray moves through the surrounding air, droplet coalescence and separation can occur, gaseous mixing of fuel vapor and air then takes place, followed finally by combustion. Models which explicitly treat the two-phase structure of this spray describe the spray behavior in terms of differential conservation equations for mass, momentum, and energy.

Two such classes of spray models exist, usually called the continuum droplet model (CDM) and the discrete droplet model (DDM). Both approaches average over flow processes occurring on a scale comparable to the droplet size, and thus require independent modeling of the interactions occurring at the gas droplet interface: typically this is done with correlations for droplet drag, and heat and mass transfer. The CDM attempts to represent the motion of all droplets via a Eulerian partial-differential spray probability equation containing, in its most general case, eight independent variables: time, three spatial coordinates, droplet radius and the three components of the droplet velocity vector. This approach imposes major computational requirements. The DDM uses a statistical approach; a representative sample of individual droplets, each droplet being a member of a group of identical non-interacting droplets termed a "parcel," is tracked in a Lagrangian fashion from its origin at the injector by solving ordinary-differential equations of motion which have time as the independent variable.

This latter type of model is used in engine spray analysis. [64] Droplet parcels are introduced continuously throughout the fuel-injection process, with specified initial conditions of position, size, velocity, number of droplets in the parcel prescribed at the "zone of atomization" according to an assumed or known size distribution, initial spray angle, fuel-injection rate, and fuel temperature at the nozzle exit. The values of these parameters are

chosen to represent statistically all such values within the spray. They are then tracked in a Lagrangian fashion through the computational mesh used for solving the gas-phase partial differential conservation equations.

The equations describing the behavior of individual droplets are [65]

$$\frac{d}{dt}\mathbf{x}_k = \mathbf{u}_k \tag{14.69}$$

$$\frac{d}{dt}(m_k\mathbf{u}_k) = -\frac{m_k}{\rho_k}\nabla p + F_{D,k}(\mathbf{u}-\mathbf{u}_k) \tag{14.70}$$

$$m_k\frac{dh_k}{dt} = \dot{q}_k + (h_v - h_k)\frac{dm_k}{dt} \tag{14.71}$$

where $\mathbf{x}_k$ is the position vector for droplet k and $\mathbf{u}_k$ its velocity, m_k is the droplet mass and ρk the droplet density, $\mathbf{u}$ is the gas velocity, h_k the droplet specific enthalpy, $\dot{q}_k$ the heat-transfer rate from the gas to the droplet, and h_v the specific enthalpy of fuel vapor. $F_{D,k}$ is the droplet drag function:

$$F_{D,k} = \frac{1}{2}\pi r_k^2 \rho C_D\,|\mathbf{u}-\mathbf{u}_k| \tag{14.72}$$

where r_k is the droplet radius, μ and ρ the gas viscosity and density, and C_D is the drag coefficient. $F_{D,k}$ is the sum of the Stokes drag and the form drag, and in the laminar limit where $C_D = 24/\mathrm{Re}$ with $Re = 2r_k\rho|\mathbf{u}-\mathbf{u}_k|/\mu$ it goes to $6\pi r_k\mu$.

An equation for the evaporation rate completes this set: it is usually assumed that the droplet is in thermal equilibrium at its wet-bulb temperature, T_s. Then a balance between heat transfer to the droplet and the latent heat of vaporization carried away by the fuel vapor exists:

$$(h_v - h_k)\frac{dm_k}{dt} = -\dot{q}_k \tag{14.73}$$

While a large portion of the droplet lifetime is spent in this equilibrium, terms can be added to Eq. (14.73) so that it describes the heat-up phase

where the droplet temperature increases from its initial value to T_s.[65] The heat and mass exchange rates are calculated from experimentally based correlations for droplet Nusselt and Sherwood numbers as functions of Reynolds, Prandtl, and Schmidt numbers.[66]

Account must now be taken of the two-way nature of the coupling between the gas and the liquid. The gas velocity, density, temperature, and fuel vapor concentration required for solving the droplet equations are taken from the values prevailing in the grid cell in which the droplet parcel resides. At the same time, a field of "sources" is assembled for the interphase mass, momentum, and energy transfer, and these are subsequently fed back into the gas-phase solution preserving conservation between phases. [67] The gas-phase mass, momentum, and energy conservation equations require additional terms to account for the displacement effect of the particles, the density change associated with mixing with the fuel vapor, the drag of the droplets, the initial momentum difference, and enthalpy difference between evaporated fuel at the drop surface and the surrounding gas, and heat transfer to the droplet. [65]

The above treatment is limited to "thin sprays" where the droplets are sufficiently far apart for interparticle interactions to be unimportant. This assumption is not valid in the immediate vicinity of the injector or in narrow cone sprays. In such "thick sprays" interparticle interactions—collisions which can result in coalescence or in reseparation of droplets—are important.

The most complete models of atomized fuel sprays represent the spray by a Monte Carlo-based discrete-particle technique. [54, 66] The spray is described by a droplet distribution function—a droplet number density in a phase space of droplet position, velocity, radius, and temperature. The development of this distribution function is determined by the so-called spray equation. [66] The distribution function is statistically sampled and the resulting discrete particles are followed as they locally interact and exchange mass, momentum, and energy with the gas, using the above Lagrangian droplet equations. Each discrete droplet represents a group or parcel of droplets. Droplet collisions are described by appropriate terms in the spray equation.

Figure 14.32 shows an example of the results such spray models can generate. The calculation involves fuel-air mixing in a direct-injection gasoline engine. The Eulerian-Lagrangian approach was used to model the

fuel spray emerging from the multi-hole injector. Sub-models were developed and used for fuel injection, spray breakup, droplet-wall interaction and the liquid wall-film dynamics. Reference 67 gives details of these spray models. The DI engine was operated stoichiometric in homogeneous mode, at 1500 rev/min and 14.5 bar bmep. It had four valves per cylinder, displaced volume of 348 cm^3/cylinder, with bore and stroke close to 76 mm. Gasoline was injected into the cylinder at 80 bar injection pressure with injection commencing at 85° ATC of the intake stroke. Figure 14.32 *a* shows the computational mesh set up: dynamic mesh layering is used to optimize cell size during piston motion. Much of the intake and exhaust use fixed cells. Around the valve seats and heads, fixed meshes move with the valves. Deforming cells in the cylinder follow the piston motion. Figure 14.32 *b* shows the computed in-cylinder flow field in the plane of an intake and an exhaust valve stem at start of injection, about halfway down the intake stroke. The intake flow vortices generated by the flow past the intake valve head are apparent: see also Sec. 8.1 . The maximum flow velocity shown is 30 m/s. Figure 14.32 *c* shows paired views of the spray configuration at two crank angles, 100 and 150 degrees ATC of the intake stroke which indicate how the droplets in the spray spread and disperse.[67]

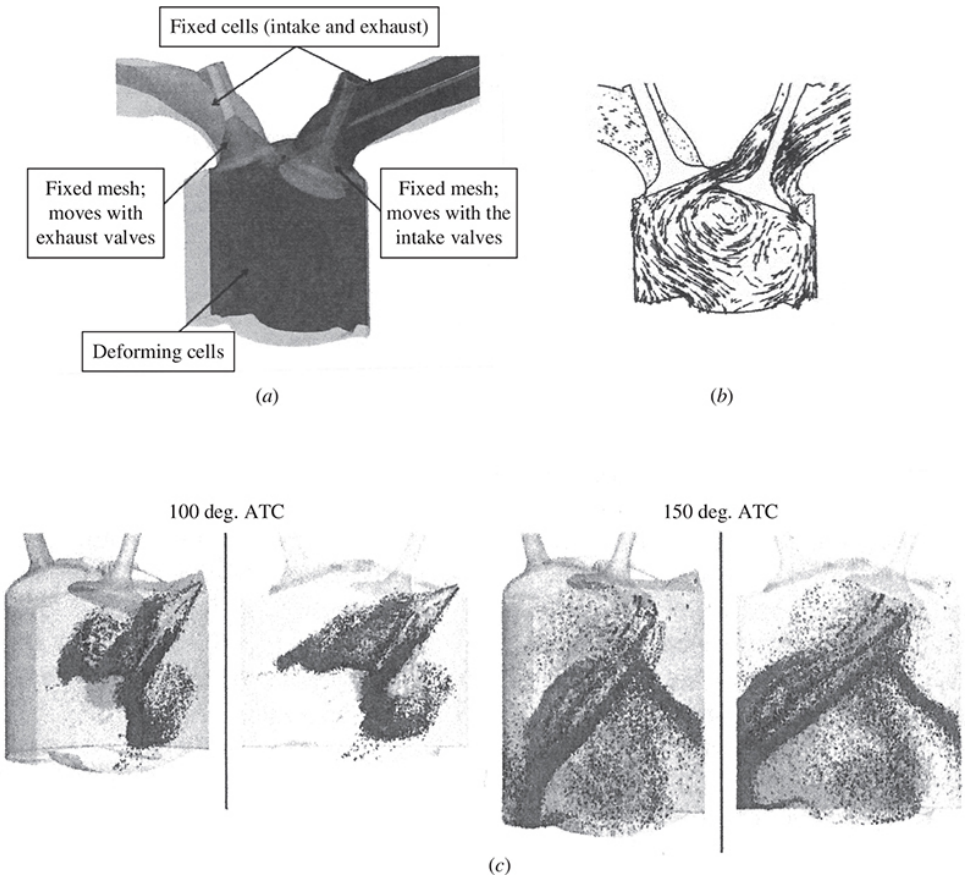


Figure 14.32 Predictions of key in-cylinder flows in a DI gasoline engine: (*a*) fixed mesh set-up which follows the motion of the valves and the piston; (*b*) computed flow field into and within the cylinder at start of fuel injection (maximum velocity is 30 m/s); (*c*) paired images showing the evolution of multi-hole-nozzle fuel sprays at 100 degrees ATC-intake (fuel injection starts at 85 deg. ATC), and at 150 degrees ATC-intake. [67]

14.5.6 Combustion Modeling

In numerical calculations of reacting flows, computer time and storage constraints have, in the past, limited the complexity of the reaction mechanisms that can be incorporated. It is now feasible to include detailed chemical mechanisms for combustion of hydrocarbon-air mixtures in CFD engine simulation codes. Detailed reaction schemes are available for the oxidation of the simpler hydrocarbon fuels such as, methane, n-heptane, isooctane, toluene, and for ethanol and methanol. For higher hydrocarbon compounds, and practical fuels which are blends of a large number of

hydrocarbons, detailed mechanisms have yet to be developed. In the past, multi-dimensional engine calculations have used simplified chemical kinetic schemes, with one or at most a few reactions or empirical equations, to represent the combustion process. While such schemes can be calibrated with experimental data to give acceptable results over a limited range of engine conditions, they lack an adequate fundamental basis.

One common practice has been to assume the combustion process, fuel + oxidizer → products, is governed by a single rate equation of Arrhenius form:

$$R_f = A\rho^2 x_f^a x_{ox}^b \exp\left(-\frac{E_A}{\tilde{R}T}\right) \tag{14.74}$$

where R_f is the rate of disappearance of unburned fuel, x_f and x_{ox} are unburned fuel and oxidizer mass fractions, $\tilde{R}$ is the universal gas constant, and a, b, A, and E_A are constants (usually a and b are taken to be unity, or to be 0.5). Values for the pre-experimental factor A and activation energy E_A are obtained by matching such equations to experimentally determined rates of burning.

While this approach "works" in the sense that, when calibrated, its predictions can show reasonable agreement with data, it has three major problems. The first is the presumption that the complex hydrocarbon-fuel oxidation process can be adequately represented by a single (or limited number of) overall reaction(s). The fact that it is usually necessary to adjust the constants in Eq. (14.74) as engine design and operating parameters change is one indication of this problem. Second, Eq. (14.74) uses local *average* values of concentrations and temperatures to calculate the local reaction rate, whereas the *instantaneous* local values will determine the reaction rate. These two rates will only be equal if the time scale of the chemical reaction is significantly longer than that of the turbulent fluctuations, which is not the case in engine combustion. Third, the implied strong dependence of burning rate on chemical kinetics is at variance with the known experimental evidence on engine combustion (see Secs. 9.3.2 and 10.3). The effects of turbulence on the burning rate, apart from the augmentation of the thermal and mass diffusivities, are not represented by equations of the form of (14.74).

An alternative, also straightforward, approach assumes that turbulent mixing is the rate-controlling process: that is, the kinetics are sufficiently fast for chemical reaction modeling to be unimportant, and reactions proceed rapidly to completion once mixing occurs at a molecular level in the smaller-scale eddies of the turbulent flow: the rate-controlling process is then the communication between and decay of the large-scale eddies. Thus the reaction rate is inversely proportional to the turbulent mixing time $\tau_T (= l_I / u')$ which is equated to k/ε. Whether the fuel or the oxygen concentration is limiting, and the need for sufficient hot products to ensure flame spreading, are also incorporated. For extremely lean (or rich) mixtures, the reaction may become kinetically controlled. A choice between Eq. (14.74) and the above mixing-controlled model can be made depending on whether the ratio of a chemical reaction time to the turbulent mixing time is greater or less than unity.

An example of a three-dimensional calculation of ignition and flame propagation in a spray-guided stratified, lean operating, gasoline DI spark-ignition engine in Fig. 14.33 indicates the sequence of individual process models (fuel spray development and evaporation, ignition, and turbulent flame propagation) as a function of crank angle from the latter part of the compression stroke into the main combustion process. A fuel spray model predicts the penetration and spread of the fuel droplets and vapor, and mixing with air, following the start of injection at 32° CA before TC. The ignition discharge (at 29° CA before TC), within the developing spray, is modeled as a spark channel. Flamelet equations incorporating autoignition are used to transition from the spark discharge to a flame kernel which, if conditions are right, develops into a turbulent premixed propagating flame. The turbulent flame incorporates the flame structure physics, and chemistry (via the laminar flame speed), as described in Ch. 9: see Fig. 9.22 and Sec. 9.3 . In this example, the spray penetrates into the bowl-in-piston combustion chamber after start of injection.) . Following ignition at 29° BTC, the turbulent flame model then takes over as the flame kernel grows. The figure shows the growing turbulent flame propagating outward from the spark location: its shape reflects the impact of the relatively high in-cylinder flow velocities induced by the bowl-in-piston combustion chamber geometry, the squish flows (into and out of the bowl), and the mixture stratification produced by the late-injection fuel spray evolution. This CFD model applied to this specific type of gasoline-fueled DI spark-ignition engine accurately

predicted the engine's performance characteristics (such as cylinder pressure and burn rate profiles).

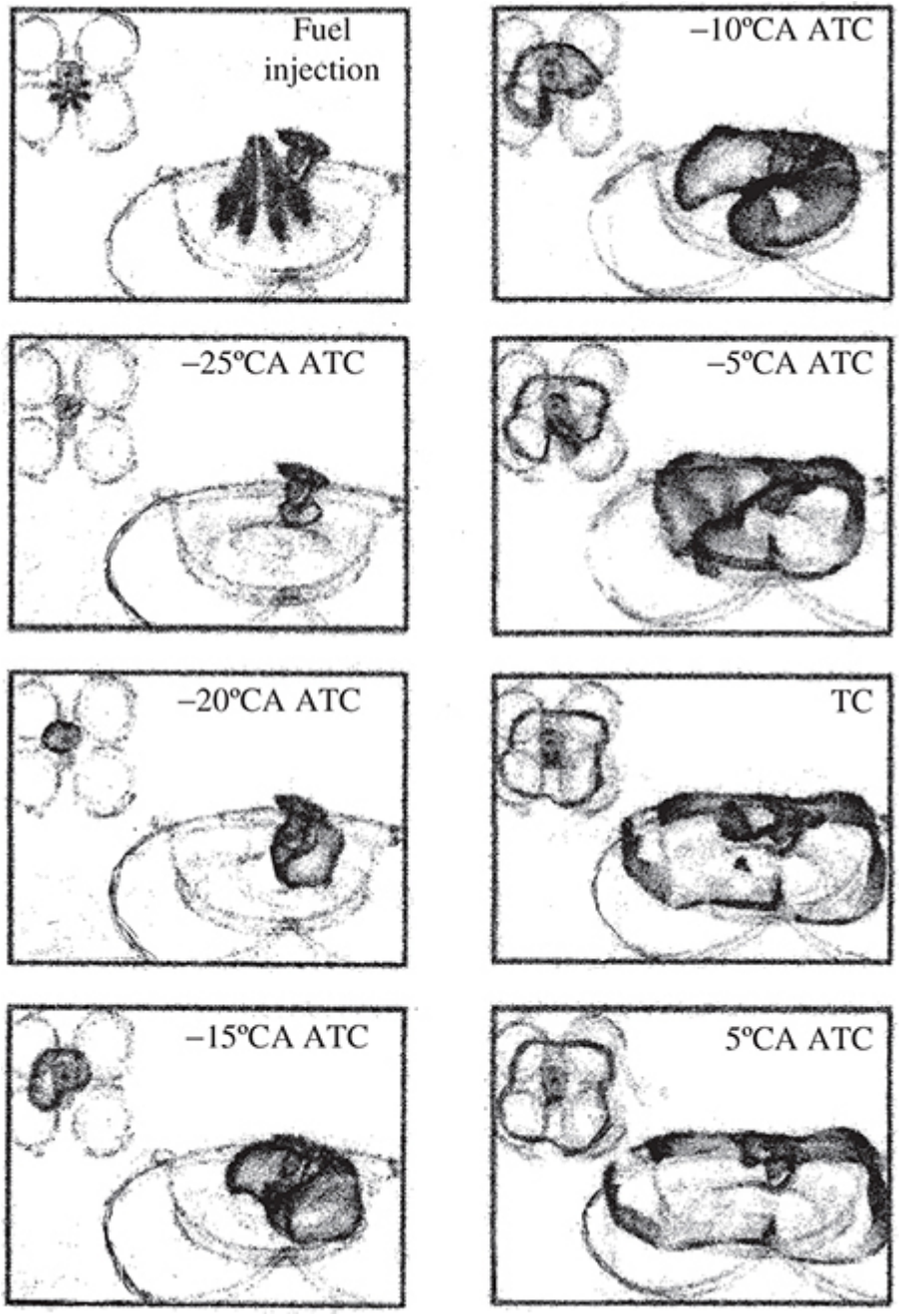


Figure 14.33 Illustration of the combustion process occurring in a spray-guided stratified DI gasoline engine modeled by a 3D CFD-based engine simulation code. The spray behavior is computed with a discrete droplet model using a multi-component fuel evaporation approach, and a model for spray impingement on the piston bowl. Ignition (29° BTC), early flame

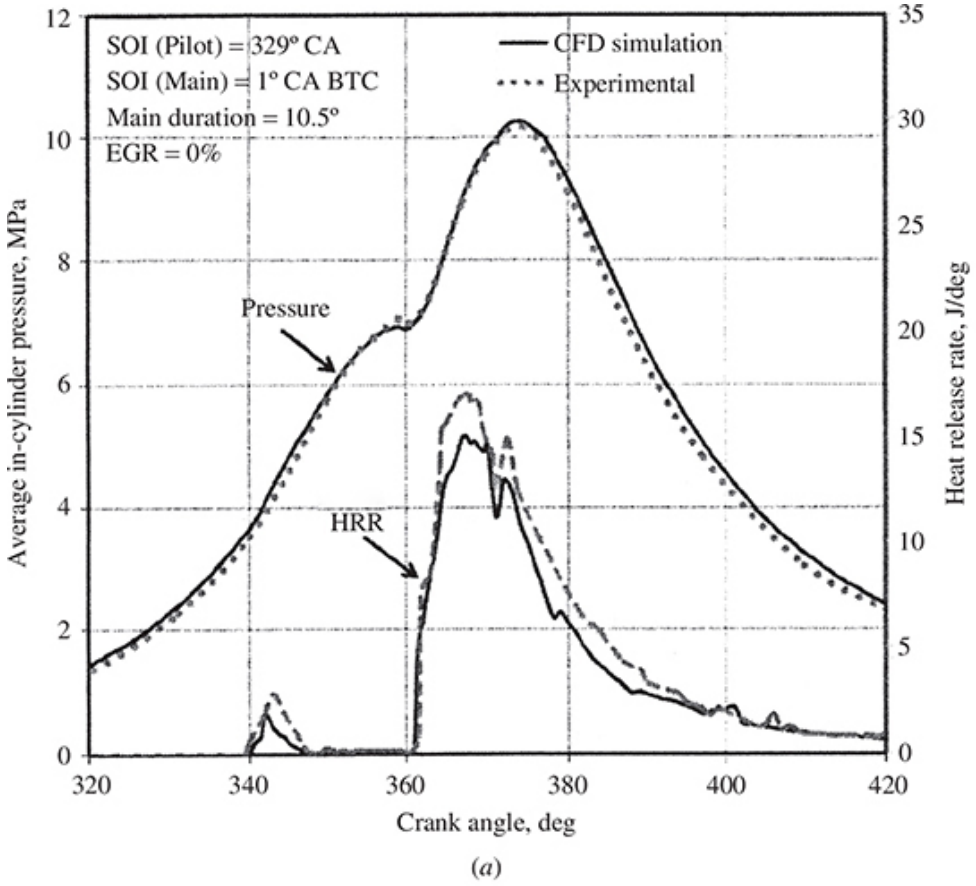
kernel growth, and fully developed turbulent flame front propagation are tracked by detailed fundamentally-based models. The first frame shows the fuel sprays. Subsequent frames show the growing flame boundary as the dark surface within the (outlined) bowl-in-piston combustion chamber, as the flame develops. In the upper left of each frame, the flame outer boundary is shown within a plan view of the full combustion chamber. Text and reference give additional details. [68]

Our final illustration of the use of 3D CFD numerical simulations of engine combustion is in a direct-injection diesel. As the discussion in Ch. 10 (Combustion in Compression-Ignition Engines) explains, the fuel spray development and the three phases of diesel combustion (autoignition, the rapid premixed fuel-rich combustion phase, and the main mixing-controlled diffusion-flame burning process) are individually complex, are linked, and (during combustion) couple turbulent convective processes with the complex HC oxidation chemistry. Models of NO x formation and particulate formation and oxidation are then built upon the diesel combustion models utilized.

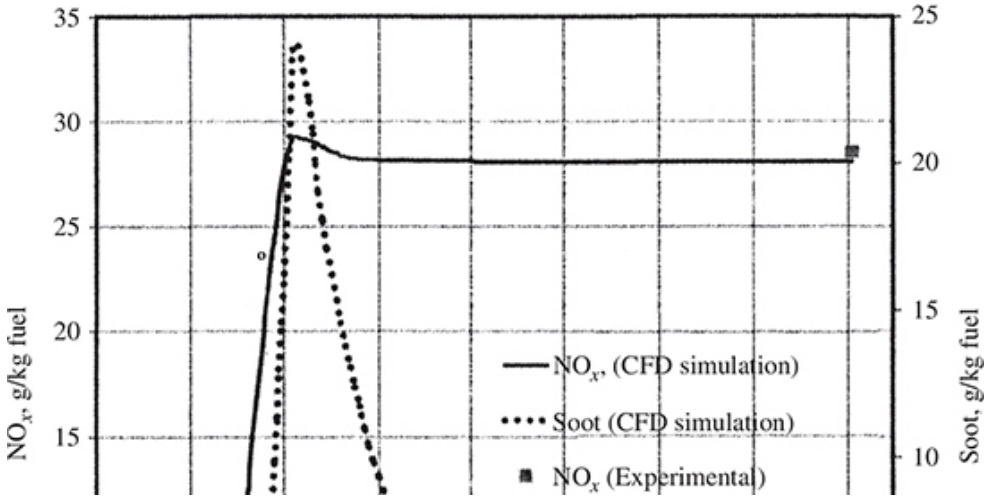
The choices for a diesel combustion model, a key component of any multi-dimensional engine model are between phenomenological models based on simplified descriptions of the underlying physics and chemistry (which are then calibrated against experimental data), or direct integration of the chemical kinetics of detailed (or reduced) chemical mechanisms. The latter approach offers better predictive accuracy and useful intermediate species information during combustion. However, it significantly extends the computational time involved. [g] More broadly, diesel combustion models used with CFD flow simulations require sub-models that predict the evolution of the fuel sprays, autoignition within the spray and the mixing controlled diffusion flames that completes the fuel burning process. Then, additional sub-models are required to connect the NO and soot formation processes to the combustion model.

An example of the overall results from such a diesel combustion model is shown in Fig. 14.34. [69] The engine involved was a 2-liter, four-cylinder (with bore and stroke both 86 mm), compression ratio 18.2:1, high-speed automobile direct-injection diesel engine with a deep re-entrant bowl-in-piston combustion chamber with swirl, and with pilot injection. A detailed fuel spray breakup, atomization, and evaporation model was used. A multi-

step generic chemical model (the Shell model) was used to simulate the ignition delay. The combustion model deployed—the Coherent Flame Sheet model—embodies the autoignition process, and the premixed and diffusion-flame phases of the diesel's combustion process. A kinetic model of the NO formation process (see Sec. 11.2.1) was included, as were phenomenological models for particulate formation and oxidation.[69]



(*a*)



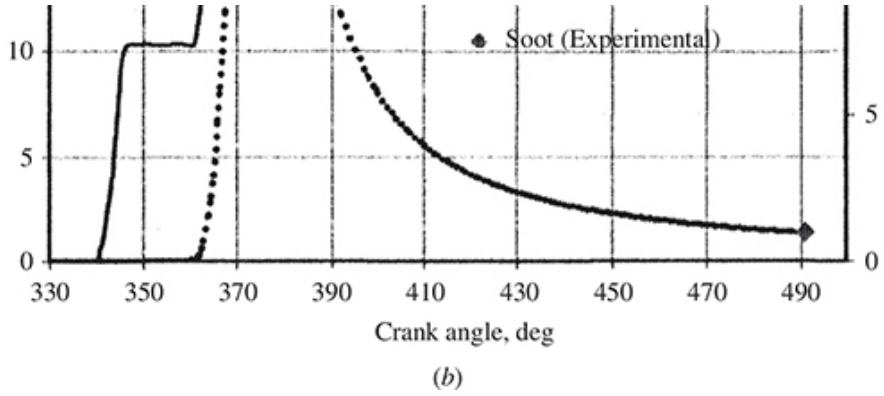


Figure 14.34 CFD engine simulation results of (*a*) cylinder pressure and heat release rate and (*b*) NO*x* and soot emissions, compared with corresponding experimental data: 2-liter four-cylinder high-speed direct-injection diesel, with pilot injection and re-entrant bowl-in-piston combustion chamber, at 1600 rev/min. Details in text and reference. [69]

Figure 14.34 *a* shows the computational results for cylinder pressure and (apparent) heat-release rate profiles with the corresponding experimentally determined profiles. Figure 14.34 *b* shows the simulation's NO*x* and soot predictions, along with the measured engine exhaust gas values. Note that combustion of the pilot-injection fuel (2.5% of the main-injection fuel), presumably due to its close-to-stoichiometric combustion process, produces some one-third of the peak NO *x* levels formed inside the cylinder and, due to the temperature rise it produces, increases the main diffusion-flame NO formation rate. Soot forms rapidly within the fuel-rich sprays once the main-injection fuel starts to burn. Following the peak heat-release rate point, more than 90% of the soot formed within the diffusion-flame-surrounded fuel rich sprays burns up as the soot moves into the flame reaction zone. Good agreement between CFD-based simulation results and measurements is obtained. These results illustrate the value that detailed multi-dimensional CFD models can provide.

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[a] Two- and three-dimensional effects are often important and are modeled with multi-dimensional flow models described in Sec. 14.5 .

[b] Note that the center of this sphere may be convected away from the spark plug location, especially if some swirl is present. However, only strong swirling and squish flows produce major distortions to the flame surface shape.

[c] This initial laminar flame propagation process can be expressed in terms of a flame radius at which the transition to a turbulent flame occurs.[19]

[d] The combustion duration Δ_{comb} is an arbitrary period within which combustion must be completed. A value of 125 degrees CA was used above.

[e] $K_1 = 2 + 1.25 \times 10^{-8} \left(\tau_{id} N\right)^{2.4}$, $K_2 = 5000$, $K_3 = 14.2 / \phi^{0.644}$, $K_4 = 0.79 K_3^{0.25}$ where τ_{id}, the ignition delay, is in milliseconds and N, engine speed, is in revolutions per minute. This study also gave $0.8 < a <$

0.95; $0.25 < b < 0.45$; and $0.25 < c < 0.5$. [30]

[f] Of course, real thermodynamic devices will produce less work than ideal devices.

[g] A detailed discussion of these options, such as the KIVA-Characteristic-time Combustion (CTC) model and KIVA-Chemkin model, can be found in Ref. 53.

CHAPTER 15

Engine Operating Characteristics

This chapter reviews the operating characteristics of the common types of spark-ignition and compression-ignition engines. The performance, efficiency, and emissions of these engines, and the effect of changes in major design and operating variables, are related to the more fundamental material on engine combustion, thermodynamics, fluid flow, heat transfer, and friction developed in earlier chapters. The intent is to provide data on, and an explanation of, actual engine operating characteristics.

15.1 ENGINE DESIGN OBJECTIVES

Internal combustion engines are used in many different applications, though road transportation is the dominant one. The major engine design objectives and operating requirements are common to many of these applications: they are listed in Table 15.1 [1]. These several objectives are important, and meeting them all involves trade-offs.

TABLE 15.1 Engine design objectives [1]

1. High performance (WOT torque, maximum power)
2. Excellent part-load fuel consumption
3. Meets air pollutant emission requirements
4. Good transient response (e.g., in a vehicle context)
5. Quiet operation, low vibration, good sound quality
6. Low weight, bulk volume (packaging)
7. High reliability, little maintenance, low oil consumption
8. Lowest possible cost

Items 1, 2, and 3 in Table 15.1 are obvious design requirements. Wide-open-throttle torque and power, as a function of engine speed, define the capabilities of the engine to produce the desired output under highly loaded operating conditions. In many applications, the time an engine spends at high outputs (and high speeds) is limited, so the engine's fuel consumption and emissions levels are often most critical at torque levels significantly below the maximum, under conditions where the engine operates for a large fraction of the time. [a] In light-duty vehicles, for example, engines operate for much of their "on time" at about one-quarter of the maximum torque, and at relatively low speeds (1500 to 2000 rev/min).

An engine's emissions of HC, CO, NO x, and particulate matter are a result of the within-the-engine processes, and the effectiveness of the engine's exhaust aftertreatment system (if one is included). Normally, the aftertreatment system does more than 90% of the clean-up task. However, the engine-out emissions levels are still important (especially before the exhaust catalysts have warmed-up sufficiently to achieve light off). See Chap. 11 for a detailed discussion of emissions formation and controls.

While much of engine operation can be regarded as quasi-steady, depending on the engine's application area, transient behavior (item 4 in Table 15.1) is usually important also. Items 5, 6, and 7 are obviously critical practical engine design requirements.

Internal combustion engines (spark-ignition engines and diesel) steadily improve over time. [2, 3] They do this as a result of many improvements and changes such as increased understanding of the opportunities and constraints, better integrated design, invention, more sophisticated controls, improved

and new materials, and reductions in manufacturing costs. Strong competition in the engine market pushes these improvements; as also does pressure from alternatives to the petroleum-fueled IC engine. [4] Mainstream internal combustion engines continue to dominate the many application areas of widespread use, in large part due to this steady ongoing improvement in performance and operation. In the performance arena, these improvement in mainstream engine performance parameters are at rates of some 1 to 2% per year. Over decades, this adds up to substantial progress; year to year, average changes are more modest. Note that shifts in focus often occur in parallel (e.g., increasing emphasis on turbocharged and downsized gasoline spark-ignition engines).

In this chapter, our focus is on the operating characteristics of spark-ignition and diesel engines, largely using mainstream petroleum-based fuels, gasoline (petrol), or a distillate diesel fuel, respectively. Note that in many parts of the world, gasoline now contains some 5 to 10% ethanol, and diesel fuel may contain a few percent biodiesel (see Sec. 10.5.5). At these blend levels, the changes in fuel properties are modest, though ethanol has good knock resistance and through E5 or E10 blends improves the octane rating of the base (hydrocarbon) gasoline stock. The future prospects for radically different engine fuels are unclear: [4] this is discussed at the end of the chapter. Natural gas is used as a spark-ignition engine fuel, and its impact on engine behavior will also be reviewed.

15.2 ENGINE PERFORMANCE

15.2.1 Basic Characteristics of SI and Diesel Engines

As will by now be apparent, the ignition and combustion processes in gasoline-fueled spark-ignition engines, and in diesel-fueled diesel engines, are fundamentally different. Gasoline SI engines (unless in lean stratified mode) operate with a largely premixed fuel-vapor/air mixture (usually stoichiometric) that burns through a propagating turbulent wrinkled-laminar flame. The phasing of combustion relative to the piston's motion (determined

by the crank angle of spark discharge) is usually close to that which gives the maximum brake torque (MBT) or minimum fuel consumption. Load (torque) is usually controlled by changing the intake system pressure through a throttle valve upstream of the intake manifold plenum: as the throttle is increasingly closed, the pressure drop across the throttle increases, less air is inducted into the cylinder, and less torque is generated. With turbocharged gasoline engines, with the throttle in its wide-open position, the airflow depends on the level of boost generated in the turbocharger compressor: increasing boost pressure above atmospheric increases the amount of air inducted, the amount of fuel burned can be increased, and thus the engine's torque rises.

Combustion in the diesel engine is significantly different. The diesel operates unthrottled, so the airflow into the cylinder depends on engine speed. When operating turbocharged, in and above the mid-load region, the compressor generates boost (corresponding to the engine's load and speed) that increases the airflow into the engine. Fuel is injected in a relatively short pulse through several nozzle holes directly into the cylinder as each piston approaches TC. Partial entrainment and mixing of air into each vaporizing fuel spray results (after a short delay) in spontaneous ignition of the rich fuel-vapor/air mixture within each spray, rapidly releasing some of the fuel's chemical energy. Then, a diffusion flame develops around each spray and the main diffusion-flame phase of the combustion process occurs as "fuel" (the products of rich-mixture combustion, the first phase of the diesel's combustion process) diffuses from the spray interior into this diffusion flame surrounding the spray. Simultaneously, air diffuses into the flame from outside the spray. Combustion, the release of the fuel's chemical energy, occurs in the diffusion flame at close to stoichiometric conditions. Load (torque level) is controlled by the amount of fuel injected per cycle. (Since diesel engines operate lean, there is always enough air.) In a naturally-aspirated diesel, the airflow changes little as load is changed: in a turbocharged diesel, when the turbocharger is generating boost there is some feedback between engine load and compression boost level, and therefore airflow. At light loads, the turbocharger is largely ineffective and the overall engine fuel/air equivalence ratio is very lean: at maximum torque, the fuel/air equivalence ratio increases to approach 0.7, conditions just below the value of f at which the black smoke combustion limit occurs. Thus the diesel's overall equivalence ratio increases steadily as load (torque) is increased from low to high levels. Combustion retard (in this case controlled by

retarding the start of injection) is used to moderate the in-cylinder peak pressure level, exhaust gas temperature, and engine-out emissions.

Figure 15.1 illustrates these two different combustion processes (SI engine and diesel) at two operating conditions for these light-duty vehicle engines. The cumulative heat release, as a percent of the total injected fuel's chemical energy ($m_f Q_{\mathrm{HV}}$), is plotted versus crank angle. Figure 15.1 *a* compares these combustion profiles at a light-load (260 kPa bmep) and low speed (1500 rev/min). Figure 15.1 *b* shows a similar comparison at full load and 4000 rev/min. At the light load, the diesel combustion process is initially very fast, and then slows down after about half the fuel has burned. The gasoline engine combustion profile exhibits a slower flame development process (some 15 degrees CA) and a comparable ending process (beyond 90% burned). The high load, higher speed comparison indicates that the gasoline-engine burning process shortens in terms of crank angle degrees. However, the burn duration in crank angle degrees ($\Delta\theta_{10\text{–}90\%}$) in gasoline engines varies only modestly across normal load and speed ranges, because the turbulent flame speed scales approximately with gas velocities which scale with engine speed via the piston motion. In the diesel engine, the amount of fuel injected at high load is about five times that at light load, and since engine speed is almost three times higher, the injection duration is much longer which is reflected in the significantly longer (in crank angle degrees) diesel burn profile. Note that while the diffusion-flame burning phase of the diesel combustion process is often described as *mixing controlled* (as it is), it does not scale as well with engine speed as does the spark-ignition engine premixed flame propagation process. Thus diesel engine high-load torque becomes limited at a lower engine speed (4500 rev/min) than does gasoline engine torque (at some 6000 rev/min), due to the slower rate of increase in mixing and thus burning rate as speed increases in the diesel engine.

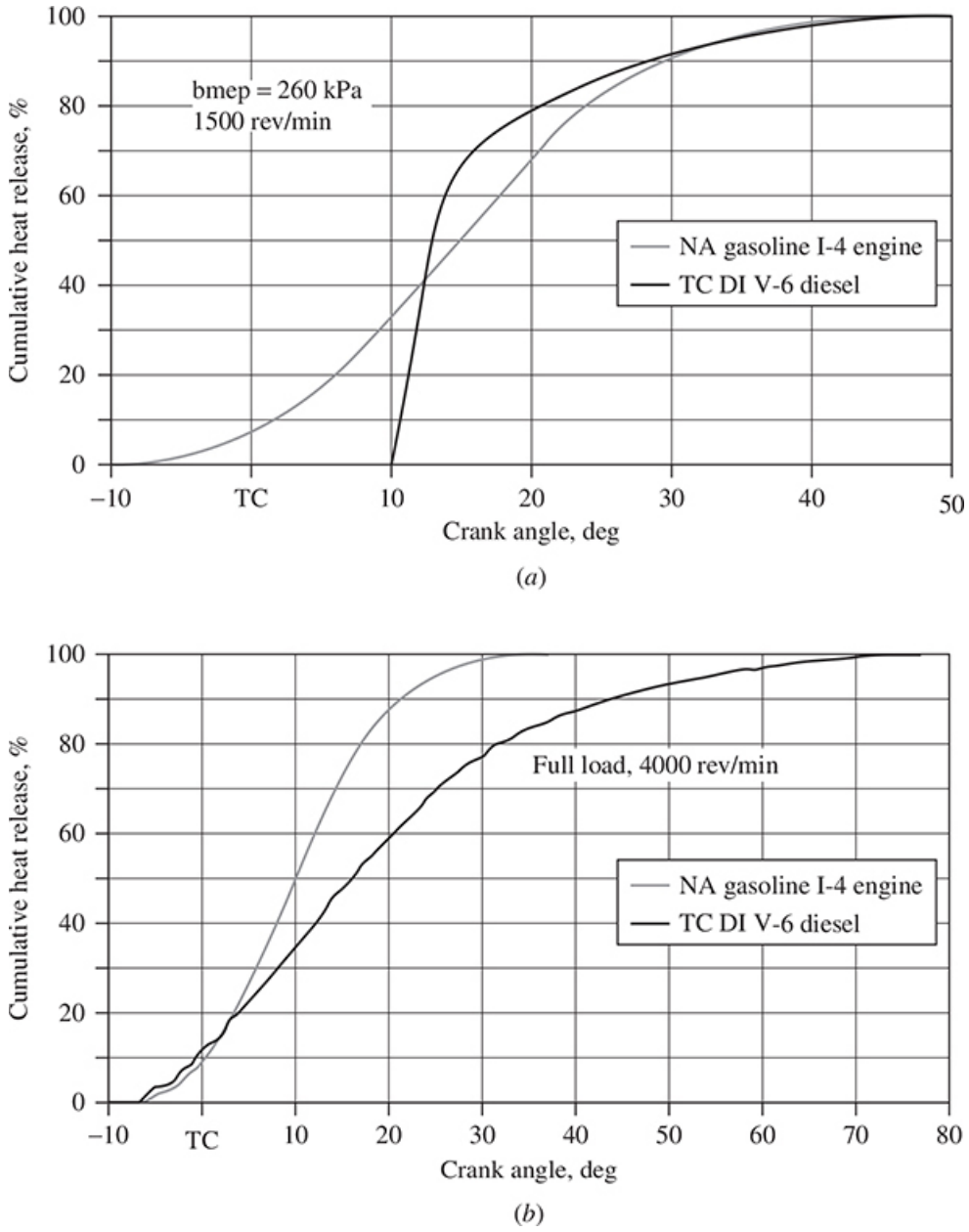


Figure 15.1 Heat release comparison: naturally-aspirated 2-liter four-cylinder SI gasoline engine and 3-liter V-6 turbocharged diesel engine. Upper figure (*a*): both engines are lightly loaded—bmep = 2.6 bar, 1500 rev/min. Lower figure (*b*): both engines at full load and 4000 rev/min. [5]

This combustion summary provides a useful introduction to the differences in performance and operating characteristics of these two types of internal combustion engines.

15.2.2 Characterizing Engine Performance

The practical performance parameters of interest are an engine's maximum power, the maximum torque available as a function of engine speed, and the engines specific fuel consumption. Power and torque are related by

$$\text{Power} = 2\pi \times \text{Torque} \times \text{Speed}$$

Power and torque depend on an engine's displaced cylinder volume. If the engine is turbocharged, they also depend on the air's boost pressure at entry to the engine's cylinders. [b]

In Chap. 2 , a set of normalized and dimensionless performance and emissions parameters was defined to eliminate the effects of engine size. Power, torque, and fuel consumption were expressed in terms of these parameters (Secs. 2.4 and 2.8), and the significance of these parameters over an engine's load and speed range was discussed (Sec. 2.15). The key normalized parameters are mean effective pressure, mep, (see Sec. 2.7) which is the work per cycle divided by the displaced volume,

$$\text{mep} = \frac{Pn_R}{V_d N} = \frac{2\pi n_R T}{V_d} \tag{15.1}$$

where n_R is the number of crank revolutions per power stroke ($n_R = 2$ for four-stroke cycle engines, and 1 for two-stroke cycle engines), and specific fuel consumption

$$\text{sfc} = \frac{\dot{m}_f}{P} \tag{15.2}$$

the fuel flow rate per unit power output. These parameters can have *brake* and net and gross *indicated* values. Torque—the turning moment the drive shaft provides—is a "brake" parameter, so bmep is effectively the engine's torque normalized by displaced volume: see Eq. (15.1) . These normalized parameters allow the effect of engine size to be made explicit. The power P can be expressed as:

$$P = \text{mep}\, A_p \bar{S}_p / 4 \quad \text{(four-stroke cycle)}$$
$$P = \text{mep}\, A_p \bar{S}_p / 2 \quad \text{(two-stroke cycle)} \tag{15.3}$$

The torque T is given by

$$T = \text{mep}\, V_d / (4\pi) \quad \text{(four-stroke cycle)}$$
$$T = \text{mep}\, V_d / (2\pi) \quad \text{(two-stroke cycle)} \tag{15.4}$$

Thus for well-designed engines, where the maximum values of mean effective pressure and piston speed are either flow limited (in naturally-aspirated engines) or stress limited (in turbocharged engines), power is proportional to piston area and torque to displaced volume. Mean effective pressure can be expressed in terms of the dimensionless efficiencies defined in Chap. 2 :

$$\text{mep} = \eta_f \eta_v Q_{\text{HV}} \rho_{a,i} \left(\frac{F}{A} \right) \tag{15.5}$$

for four-stroke cycle engines [Eq. (2.48)], and as

$$\text{mep} = \eta_f \eta_{tr} \Lambda Q_{\text{HV}} \rho_{a,i} \left(\frac{F}{A} \right) \tag{15.6}$$

for two-stroke cycle engines. The importance of high fuel conversion efficiency, breathing capacity, and intake air density is clear. Specific fuel consumption is related to fuel conversion efficiency by Eq. (2.24) :

$$\text{sfc} = \frac{1}{\eta_f Q_{\text{HV}}} \tag{15.7}$$

These parameters have both brake and indicated values (see Secs. 2.3 , 2.4, and 2.5). The difference between these two quantities is the engine's friction (and pumping) requirements, and their ratio is the mechanical efficiency η_m.

The relative importance of these parameters varies over an engine's operating speed and load range. The maximum or rated brake power (see

Sec. 2.1) and the quantities such as bmep derived from it (see Sec. 2.7) define an engine's full potential. The maximum brake torque (and bmep derived from it) over the engine's speed range, indicates the ability of the designer to obtain a high airflow through the engine over the full speed range and use that air effectively. Then over the whole engine operating range, and most especially those parts of that range where the engine will operate for long periods of time, engine fuel consumption and efficiency, and engine emissions are important. Since the operating and emissions characteristics of gasoline spark-ignition engines and compression-ignition diesel engines are substantially different, each engine type is dealt with separately.

15.2.3 Torque, Power, and Mean Effective Pressure

Figure 15.2 shows the wide-open-throttle normalized torque (bmep) and specific power (kW/liter) behavior, as a function of speed, for a naturally-aspirated and a turbocharged gasoline engine, as well as a turbocharged high-speed DI diesel. (Since these are both normalized performance parameters, engine size is not a factor in this comparison.) For the reasons discussed above in Sec. 15.2.1 , such smaller high-speed diesels operate over a narrower speed range than do gasoline engines: maximum power is realized at engine speeds of about 4500 rev/min (mean piston speeds of 13 m/s), where fuel-air mixing rates (in crank angle degrees) become too slow to achieve fast enough burning rates. The naturally-aspirated gasoline engine generates maximum power at around 6000 rev/min, at 17 m/s mean piston speed. Naturally-aspirated gasoline engines are operated up to the intake-flow choking point (see Sec. 6.2.5), to maximize the power realized. With turbocharging, engines are *down speeded*: maximum power is achieved at slightly lower speeds (5500 rev/min, 15 m/s) by increasing boost levels. In Fig. 15.2, both of the turbocharged engines show rapidly rising torque levels as engine speed increases from about 1250 to 2000 rev/min. In naturally-aspirated gasoline engines, normalized torque levels are a factor of two or so lower: such NA engines realize a maximum bmep of 12 bar at about 4500 rev/min, rather than at about 2000 rev/min as with turbocharged engines. The gasoline engines at wide-open throttle operate at stoichiometric or up to about 20% fuel rich (for knock and exhaust gas temperature control, slightly higher power). Thus they fully utilize the air inducted into the cylinders. The

maximum relative fueling rate for diesels is $\phi \leq 0.7$, so full air utilization is not achieved.

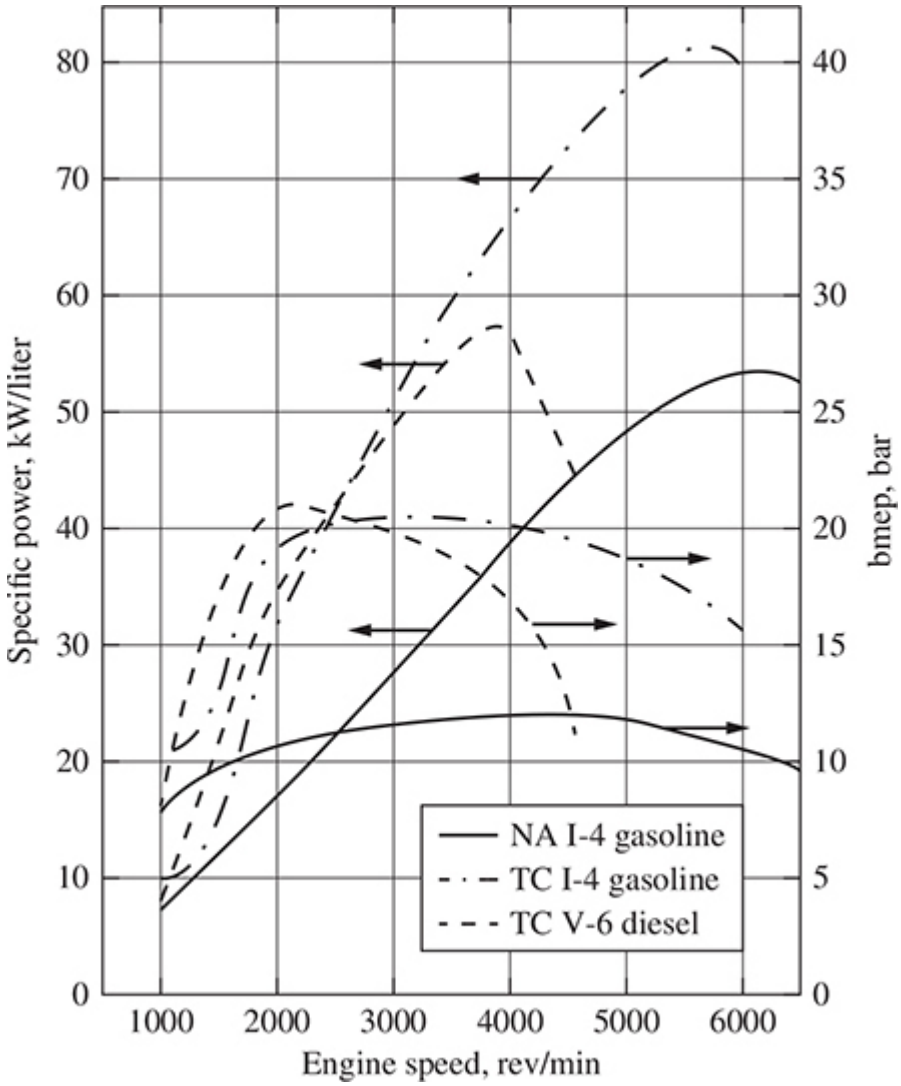


Figure 15.2 Full load bmep (bar) and specific power (kW/liter) versus engine speed for three types of internal combustion engines: (*a*) 2-liter I-4 naturally-aspirated gasoline SI engine; (*b)* 1.4-liter I-4 turbocharged gasoline engine;[6] (*c*) 3-liter V-6 turbocharged DI diesel engine.

Note that turbocharged engines normally use an *intercooler*, a charge air cooler, between the compressor and cylinder head. Reducing the air's temperature before it enters the cylinder has several important benefits. At a given compressor exit pressure, cooling the air increases its density. Also,

lower intake air temperatures reduce the burned gas temperatures and thus the amount of NO x formed within the engine's cylinders, and slow down the autoignition chemistry and thus delay the onset of knock constraints in gasoline engines.

The wide-open-throttle operating characteristics of an automobile spark-ignition engine are shown in Fig. 15.3. The brake power shown is the power the basic engine can provide, including the built-in engine accessories. The indicated (gross) power was obtained by adding the friction power to the brake power: it is the average rate of work transfer from the gases in the engine cylinders to the pistons during the compression and expansion strokes of the engine cycle (see Sec. 2.4). The indicated mean effective pressure shows a maximum in the engine's mid-speed range, just below 3000 rev/min. The shape of the indicated power curve follows from multiplying the imep curve by engine speed. Since the full-load indicated specific fuel consumption (and hence indicated fuel conversion efficiency) varies little over the full speed range, this variation of full-load imep and power with speed is primarily due to the variation in volumetric efficiency, η_v [see Eq. (15.5)]. [c] Since friction mean effective pressure increases with increasing speed, friction power will increase even more rapidly. Hence mechanical efficiency decreases with increasing speed from a maximum of about 0.9 at low speed to 0.7 at 5000 rev/min. Thus bmep peaks at a lower speed than imep. The brake power shows a maximum at about 4300 rev/min in this older gasoline engine, increases in speed above this value result in a decrease in P_b. The indicated fuel conversion efficiency increases by about 10% from 0.31 to 0.34 over the speed range 1000 to 4000 rev/min, due primarily to the decreasing importance of heat transfer per cycle with increasing speed. For a turbocharged gasoline engine, the WOT imep and bmep values will be significantly higher, while the fmep only increases slightly. Thus the mechanical efficiency will be higher (an important benefit of turbocharging).

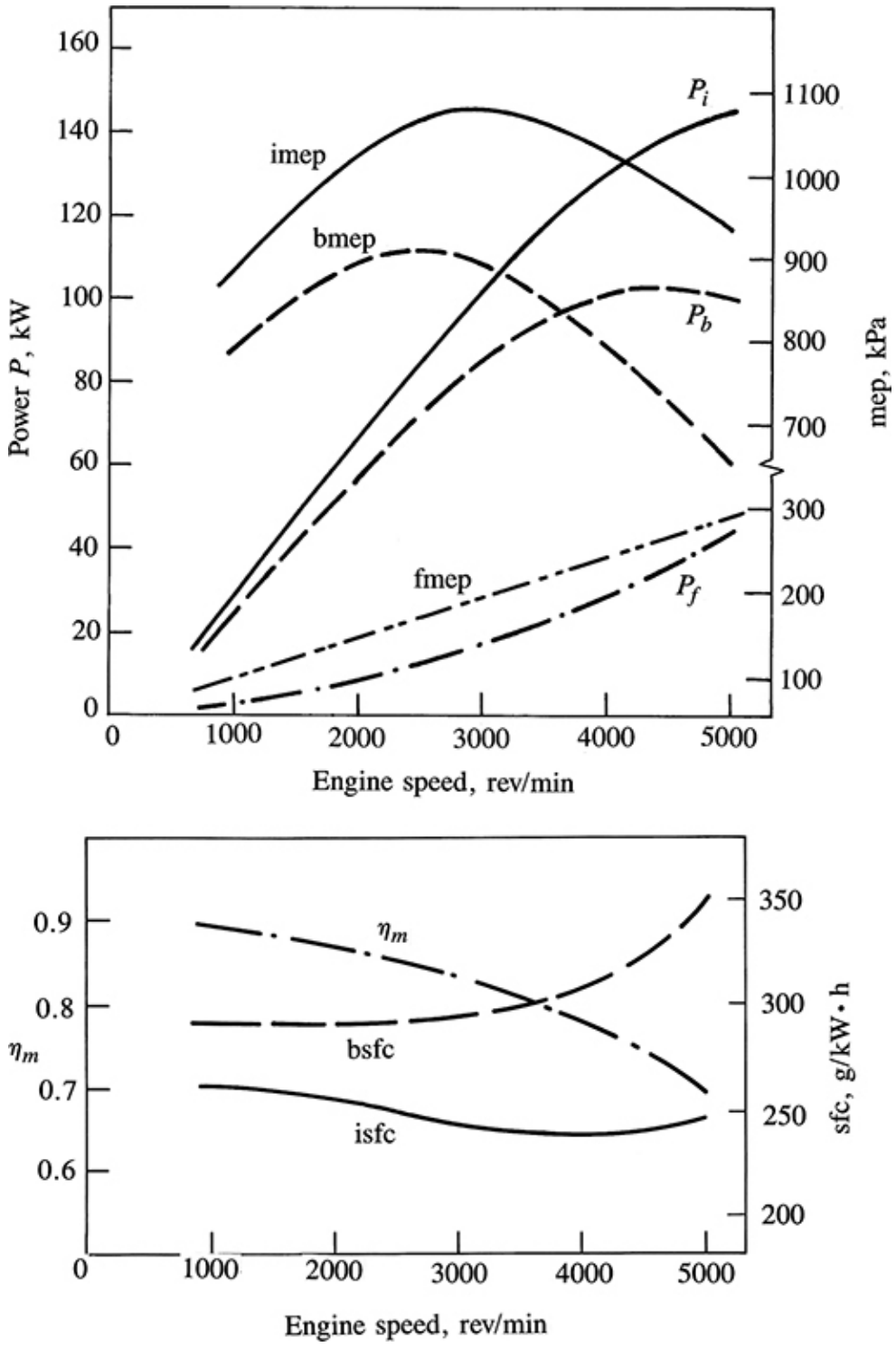


Figure 15.3 Gross indicated, brake, and friction power (P_i, P_b, P_f), indicated, brake, and friction mean effective pressure, indicated and brake-specific fuel consumption, and mechanical efficiency for a 3.8-liter six-

cylinder naturally-aspirated automotive spark-ignition engine at wide-open throttle. Bore = 96.8 mm, stroke = 86 mm, r_c = 8.6. [7]

Figure 15.4 shows full-load indicated and brake power and mean effective pressure for naturally-aspirated DI diesel engine, as a function of speed. Except at high engine speeds, brake torque and mep vary only modestly with engine speed since the intake system of the diesel can have larger flow areas than the intake of SI engines with their throttle and intake-system fuel transport requirements. The part-load torque and bmep characteristics (at fixed amount of fuel injected per cycle) have a similar shape to the full-load characteristics in Fig. 15.2. The decrease in torque and bmep with increasing engine speed is due primarily to the increase in friction mep with speed (see Figs. 13.7, 13.11, and 13.12). Decreasing engine heat transfer per cycle and decreasing airflow rate, as speed increases, have modest additional impacts.

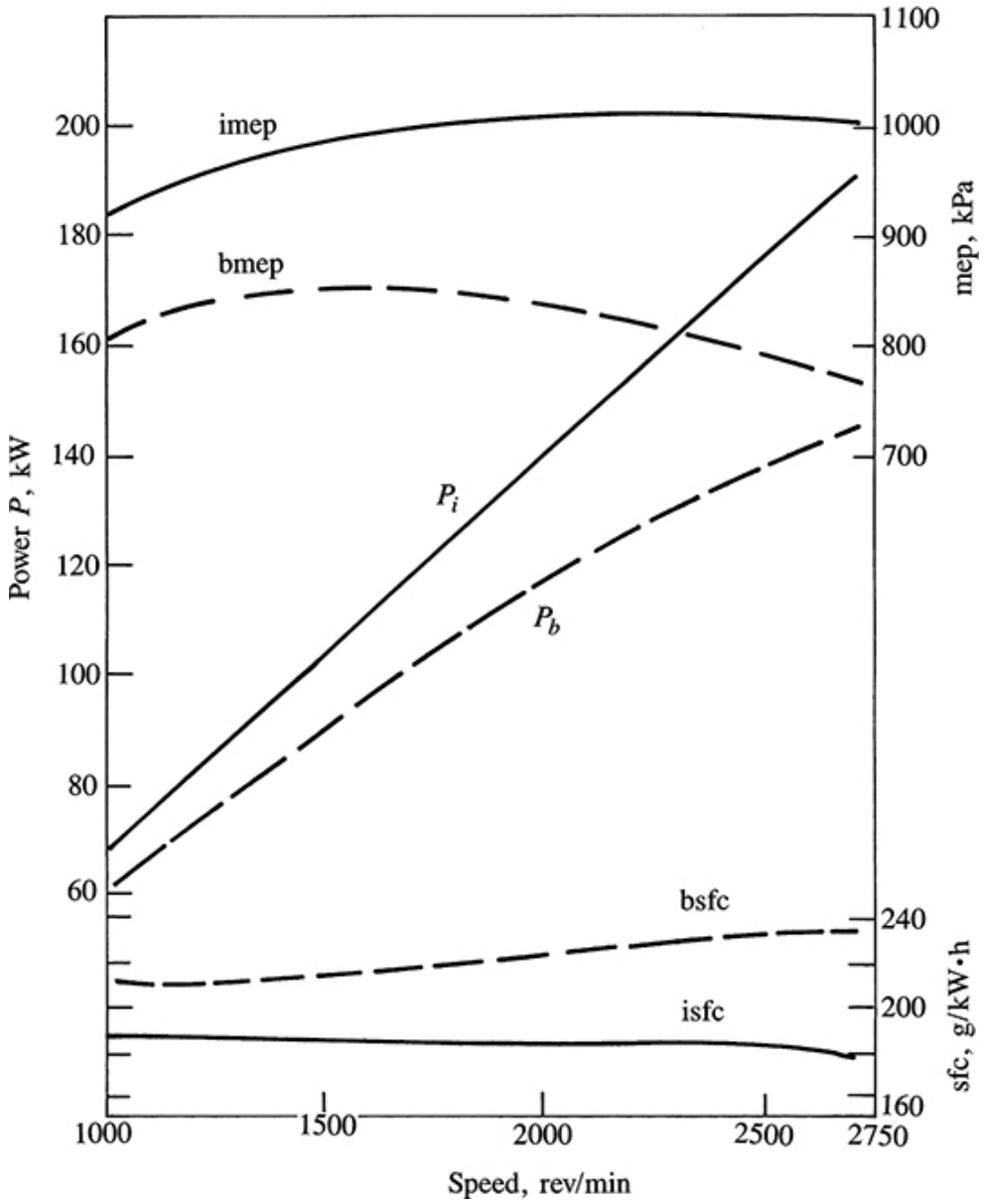


Figure 15.4 Gross indicated and brake power (P_i, P_b), mean effective pressure (imep, bmep), and specific fuel consumption (isfc, bsfc) for a 8.4-liter six-cylinder naturally-aspirated direct-injection diesel engine. Bore =115 mm, stroke = 135 mm, r_c=16 [8].

15.2.4 Engine Performance Maps

One common way to present the operating characteristics of an internal combustion engine over its full load and speed range is to plot contours of

constant brake-specific fuel consumption on a graph of torque (or brake mean effective pressure) versus engine speed. Such a plot is called an engine *performance map*. Operation of the engine, coupled to a dynamometer on a test stand, over its load and speed range, generates the torque and fuel flow-rate data from which a performance map is derived. Equations (15.1) and (15.2), also Eqs. (2.25) , (2.26), and (2.28), relate bmep and bsfc to torque, $\dot{m}_f$, speed, and power. Figure 15.5 shows an example of such a performance map for a naturally-aspirated gasoline spark-ignition engine. The upper envelope of the map is the wide-open-throttle performance curve. Points below this curve define the part-load operating characteristics (realized by steadily closing the air-controlling throttle value). While details differ from one engine to another (and turbocharged engines have significantly higher maximum load versus speed curves; see Fig. 15.2), the overall shapes of these maps are remarkably similar. Performance maps are usually plotted with crankshaft rotational speed as the abscissa: when mean piston speed ($\overline{S}_p = 2\,LN$) is used, the quantitative similarity of such maps for a given type of engine is more apparent.

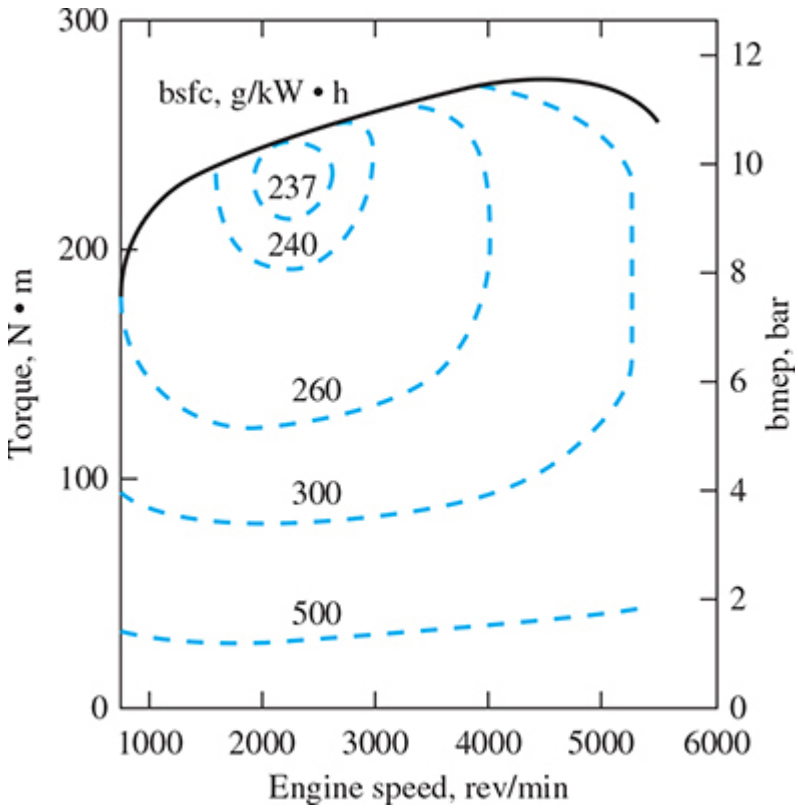


Figure 15.5 Performance map for a naturally-aspirated gasoline SI engine: torque/bmep versus engine speed showing contours of constant bsfc (g/kW·h). 3-liter, V-6, four-valve per cylinder, port-fuel-injected engine, r_c = 10, maximum power 143 kW at 5200 rev/min, bore and stroke 88 and 83 mm, respectively. [9]

Figure 15.5 shows a performance map for a standard naturally-aspirated gasoline engine. The maximum torque (at about 4500 rev/min) corresponds to a bmep of 1200 kPa. The best bsfc value (in the upper left quadrant) of 237 g/kW·h corresponds to a brake fuel conversion efficiency of 35%. In the lower half of the map, the bsfc and efficiency worsens significantly. At a typical engine-in-vehicle light load (bmep about one-quarter the max. bmep and mid engine speed), bsfc is some 380 g/kW·h and $\eta_{f,\,b}$ is 22%.

Figure 15.6 shows a performance map for a turbocharged high-speed automotive diesel engine. The upper bound of the map, the maximum torque curve, has a shape typical of turbocharged engines: a rapid rise in torque/bmep from 1000 to 2000 rev/min; then depending of relative size of turbocharger and engine, approximately constant, declining or slightly rising maximum torque as the engine moves through the mid to high-speed range. The diesel's fuel consumptions are lower than for a (naturally-aspirated) gasoline engine (Fig. 15.5), and brake efficiencies are higher. The most efficient diesel operating conditions (high torque, low- to mid-speed) has bsfc values close to 200 g/kW·h and $\eta_{f,\,b}$ values of 42 to 44%. At typical automotive engine conditions (quarter maximum torque/bmep and mid-speed), the fuel consumption and efficiency worsen to some 260 g/kW·h and 33%, respectively.

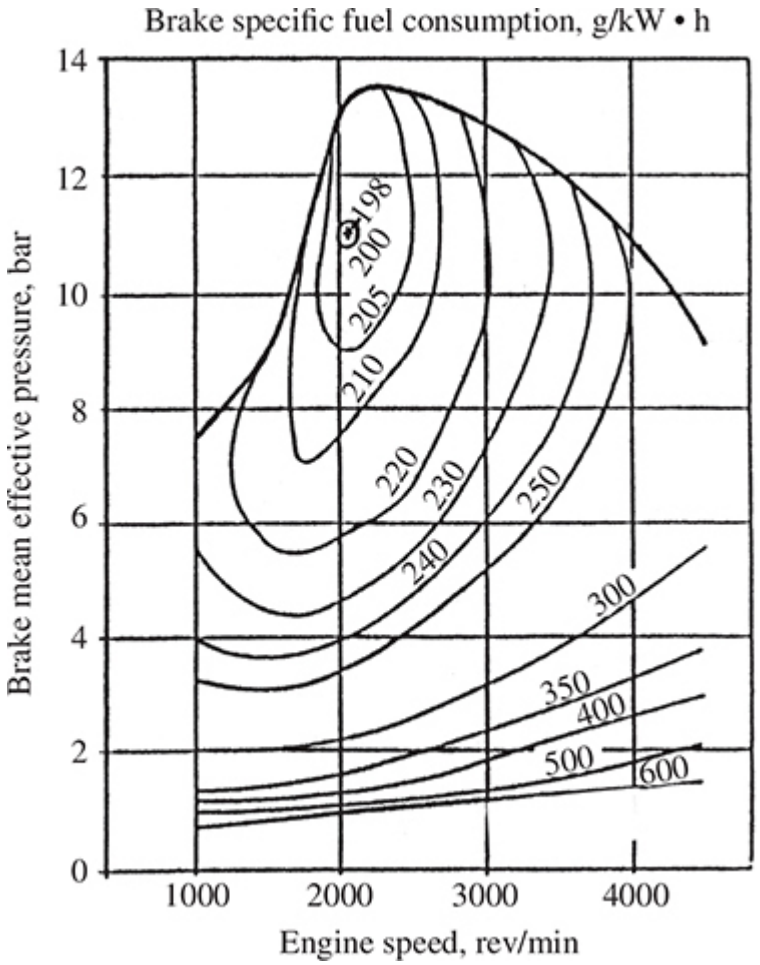


Figure 15.6 Performance map for a turbocharged 2.5-liter DI 5-cylinder diesel engine: bmep versus engine speed, with brake-specific fuel consumption contours (g/kW·h). Torque, in excess of 240 N·m, available over a wide speed range (1500 to 3300 rev/min): maximum torque 265 N·m, maximum bmep = 13.5 bar at 2250 rev/min. Compression ratio = 21, bore and stroke 86 mm. [10]

This comparison, naturally-aspirated gasoline to turbocharged diesel engine shows that at the most efficient operating conditions (high load, lower speed), the ratio of fuel consumptions, diesel:gasoline, is 0.8 to 0.85. At typical light load and low speed, this ratio is 0.65 to 0.7: that is, as load decreases, the diesel's efficiency advantage increases significantly. The reasons behind this difference between diesel and gasoline engines are the following. The diesel operates fuel-lean overall, and has a higher compression (in effect expansion) ratio: both of these directly impact engine

fuel consumption and efficiency. Also, at light load, the naturally-aspirated gasoline engine is substantially throttled: the intake manifold pressure at this typically one-quarter maximum load point is about 0.5 bar (abs.) . This adds significantly to engine friction (see Sec. 13.7).

The overall engine fuel consumption and efficiency behavior over such performance maps is explained by the set of three maps shown in Fig. 15.7. Each map shows lines of constant values of different efficiencies: (*a*) gross indicated fuel conversion efficiency, $\eta_{f\,,\,\mathrm{ig}}$; (*b*) mechanical efficiency, η_m; (*c*) brake fuel conversion efficiency, $\eta_{f,\,b}$. For the gasoline engine (operating stoichiometric), the gross indicated fuel conversion efficiency is almost constant over the operating map (in the 35 to 40% range). The middle map (Fig. 15.7 *b*) shows lines of constant mechanical efficiency (η_m = bmep/imep = 1 - fmep/imep). The mechanical efficiency is about 0.9 at high loads and low speeds. It decreases as speed is increased at constant bmep due to the rise in fmep with speed. Most importantly, at constant engine speed, η_m decreases markedly as load is decreased. Friction torque/fmep does not change that much with load [though in naturally-aspirated engines, where pumping work friction is included in total friction, the increase in pumping work as load decreases (see Fig. 13.30) does increase total friction some]. Thus as load decreases, friction becomes relatively much more important compared to the work delivered to the pistons (tfmep/imep increases substantially). At the one-quarter maximum torque, typical driving engine-torque requirement, mechanical efficiency for a naturally-aspirated gasoline engine is 0.5 to 0.6: at these conditions, almost half the work transferred to the engine's pistons is used to overcome engine friction. (At idle, η_m is zero, by definition.)

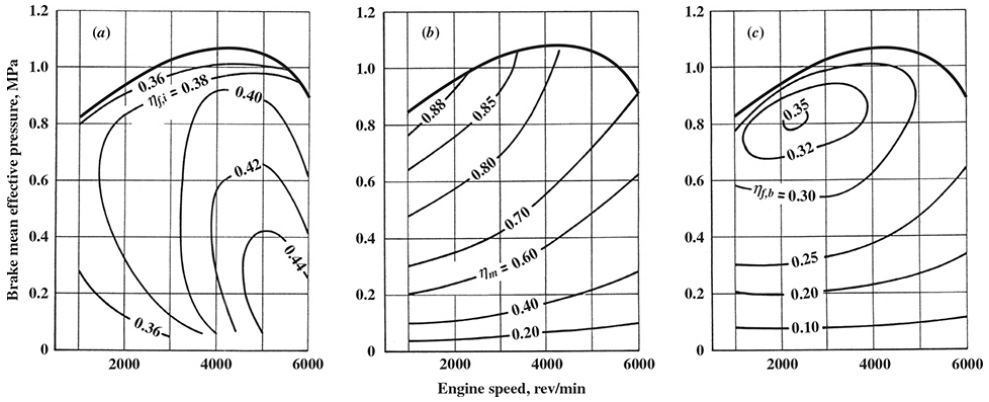


Figure 15.7 Three performance maps for a standard naturally-aspirated gasoline engine showing lines of (*a*) constant gross indicated fuel conversion efficiency, (*b*) constant mechanical efficiency, and (*c*) constant brake fuel conversion efficiency.[11]

This same logic is relevant to turbocharged engines, and diesel engines, with these additional factors. Turbocharging an engine raises its maximum bmep line by about a factor of two. It does not increase friction mep that much. Thus mechanical efficiency values do not decrease anywhere near as fast as load is decreased. If the maximum bmep is doubled, η_m at one-quarter maximum bmep would increase from a value of 0.6 in Fig. 15.7 to about 0.75. This would substantially increase the typical part-load brake-specific fuel consumption and efficiency. In diesels, one important additional factor is that as load decreases so does the engine's overall fuel-air equivalence ratio. Shifting to increasingly leaner mixtures leads to an increasing *indicated* fuel conversion efficiency.

A key question in the discussion of engine performance is engine size (and individual cylinder size). We have shown in this section that normalized performance parameters (which are size independent) are useful and instructive. In practice, engines are sized for specific tasks usually stated in terms of the maximum torque and power required. When comparing different types of engines where specific power and bmep vary, different engine displacements will be required to do the same practical task (stated in terms of the torque and power required). The example below goes through such a comparison: *downsizing* a turbocharged gasoline engine so it matches a larger naturally-aspirated engine.

EXAMPLE 15.1

Determine the displacement of a turbocharged gasoline SI engine that matches the performance of a 5.0-liter naturally-aspirated engine that produces 270 kW (300 hp) brake power and 515 N·m (380 lb $_f$ ft) torque.

The torque, brake mean effective pressure relationship is:

$$\text{bmep} = W_{\text{cycle,brake}}/V_d = 4\pi T/V_d$$

The following table gives the performance characteristics of the 5-liter NA engine and the matching 3.5-liter TC engine:

Performance parameter	Naturally-aspirated engine	Turbo engine
Displaced volume	5.0 L	3.5 L
Maximum power	270 kW	272 kW
Maximum torque	515 N·m	570 N·m
Maximum bmep	1294 kPa	2045 kPa
¼ Maximum bmep	324 kPa	511 kPa
Total fmep	130 kPa	150 kPa
Mechanical efficiency	0.7	0.77
Brake fuel conversion efficiency	25%	28%

The end result: the TC engine due to its lower friction (in absolute and relative terms), at the same maximum power, gives higher torque at lower speed, and 10 to 15% better part-load fuel consumption and brake efficiency.

The maximum bmep levels realized in the different types of gasoline and diesel engines have been steadily increasing over time. This is due to improvements related to in engine breathing and combustion, increases in compression ratio, reductions in friction and, in turbocharged engines, increasing boost levels. Thus the engine displacement required to produce the same torque and power goes down. Engine sizes are moving downward to match this evolution in normalized engine performance with the practical performance requirements. As engine displacement is reduced, the number of cylinders and the size of those cylinders can be reduced. The geometric relationships between cylinder size, number of cylinders, and engine displacement are shown in Fig. 15.8. The lines for 3, 4, 5, 6, 8, 10, and 12-cylinder engines are purely geometric. The data [3] are for 2008 model year North American gasoline engines (almost all are naturally-aspirated). The range in cylinder volume is 330 cm^3 (three-cylinder, 1-liter engine) to 750 cm^3 (large, eight-cylinder engine). Average cylinder size in the United States is 500 to 600 cm^3; it will be about 25% less (400 to 450 cm^3) in most of the rest of the world.

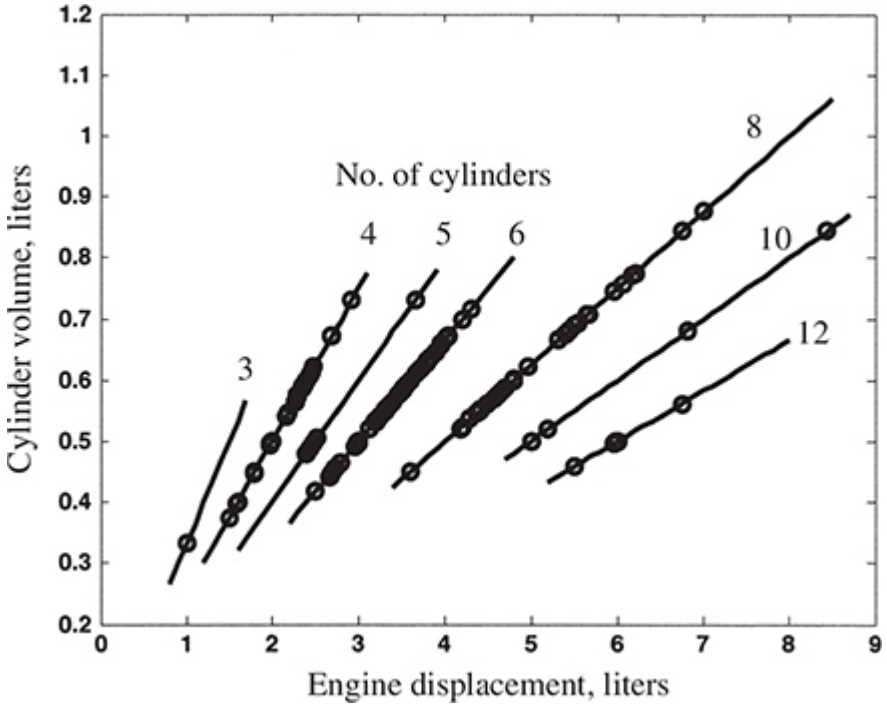


Figure 15.8 Cylinder volume as a function of engine size for 3, 4, 5, 6, 8, 10, and 12-cylinder engines. Data: 2008 model year North American light-duty vehicle gasoline engines. [3]

15.3 OPERATING VARIABLES THAT AFFECT SI ENGINE PERFORMANCE, EFFICIENCY, AND EMISSIONS

The major operating variables that affect spark-ignition engine performance, efficiency, and emissions at any given load and speed are: spark timing, fuel/air or air/fuel ratio relative to the stoichiometric value, and fraction of the exhaust gases that are recycled for NO_x emission control. Load is, of course, varied by varying the inlet manifold pressure. Both naturally-aspirated and turbocharged gasoline SI engines are in common use. In the former, the maximum manifold air pressure (MAP) is just below one atmosphere when the engine's throttle is fully open, and decreases as the throttle plate rotates toward the closed position. In turbocharged gasoline engines, the turbocharger compressor starts to provide boost as the throttle

opens and the intake manifold pressure rises above atmospheric. [d] The effect of these variables will now be reviewed.

15.3.1 Spark Timing

Figure 9.5 and the accompanying text explain how variations in spark timing (and combustion phasing defined by the $\theta_{50\%}$ mass-fraction burned point), relative to top-center, affect the pressure development in the SI engine cylinder. Fundamentally, it is the phasing of the combustion process relative to the piston's motion that determines the amount of work transferred to the piston. If combustion starts too early in the cycle, the work transfer from the piston to the gases in the cylinder at the end of the compression stroke is too large; if combustion starts too late, the peak cylinder pressure is reduced and the expansion stroke work transfer from the gas to the piston decreases. Changes in combustion phasing, affect the relative importance of heat losses, also. There exists a particular spark timing which gives maximum engine torque at fixed speed, and mixture composition and flow rate. It is referred to as MBT—maximum brake torque—timing. This timing also gives maximum brake power and (except in boosted engines) the minimum brake-specific fuel consumption. Figure 15.9 *a* shows the effect of spark advance variations on wide-open-throttle brake torque at selected speeds between 1200 and 4200 rev/min for a production eight-cylinder engine. At each speed, as spark is advanced from an initially retarded setting, torque rises to a maximum and then decreases. MBT timing depends on speed; as speed increases the spark must be advanced to maintain optimum timing because the duration of the combustion process in crank angle degrees increases. Optimum spark timing also depends on load. As intake manifold pressure and load are decreased, the spark timing must be further advanced to maintain optimum engine performance.

The maximum in each brake torque curve in Fig. 15.9 *a* is relatively flat. Thus accurate determination of MBT timing is difficult, but is important because NO and HC emissions vary significantly with spark timing. In practice, to permit a more precise definition of spark timing, the spark is often retarded slightly to give a 1 or 2% reduction in torque from the maximum value.

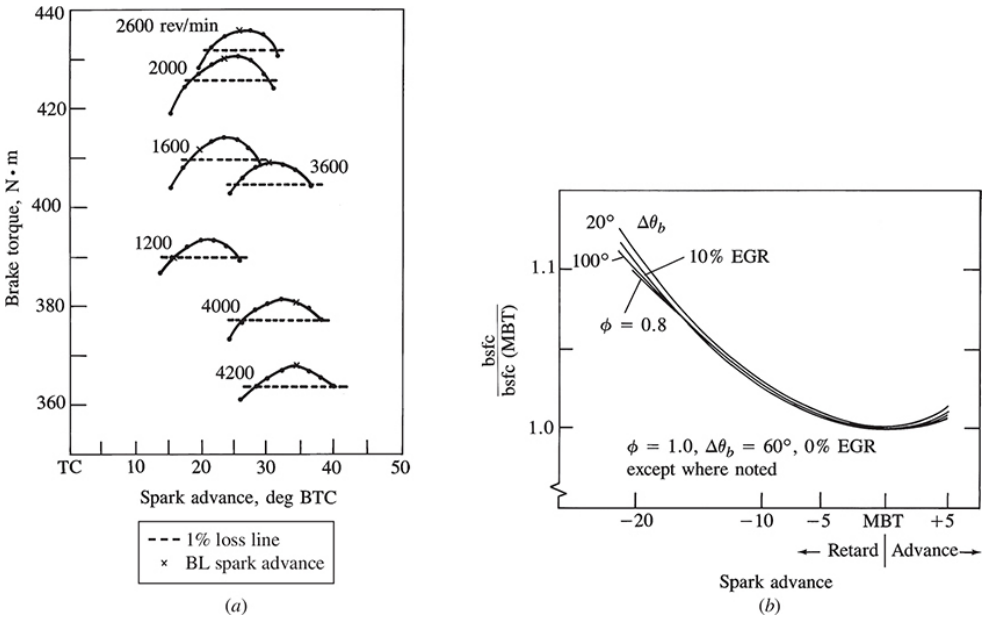


Figure 15.9 (*a*) Variation in brake torque with spark advance, eight-cylinder automotive spark-ignition engine at wide-open throttle, at engine speeds from 1200 to 4200 rev/min. One percent torque loss from MBT and spark advance for borderline (BL) knock are shown. [12] (*b*) Predicted variation in brake-specific fuel consumption (normalized by MBT value) with spark retard at several different part-load engine conditions. [13]

In Fig. 15.9 *a* the mixture composition and flow rate were held constant at each engine speed. If the mixture flow rate is adjusted to maintain constant brake torque, the effect of spark timing variations on fuel consumption at constant engine load can be evaluated. Figure 15.9 *b* shows results obtained with a computer simulation of the engine operating cycle.[13] The curves for several different part-load operating conditions and burn durations (from fast to slow) have been normalized, and fall essentially on top of each other.

The effects of spark retard on net imep, normalized by the maximum (MBT) combustion-timing value are shown in Fig. 9.16 for a wide range of *naturally-aspirated* engine data. When the abscissa is $\theta_{sp} - \theta_{sp,MBT}$ there is a common curve with some data variation: when the abscissa is ($\theta_{50\%} - \theta_{50\%,MBT}$) a tighter universal curve is obtained. [e]

Figure 15.10 shows data for a turbocharged gasoline engine at various boost levels. On the retard side of MBT timing, as boost levels rise, so do

the normalized net imep values. This occurs because at constant λ, as combustion is retarded the exhaust energy increases. This increased exhaust energy/temperature drives the turbocharger more strongly, increasing the compressor airflow and outlet pressure. Thus the nimep decrease with spark retard under these (higher) boosted conditions is reduced. Note, however, that the fuel consumption does not benefit: the higher airflow (at constant stoichiometry) requires an increase in fuel flow, and with retarded combustion, the engine is less efficient.

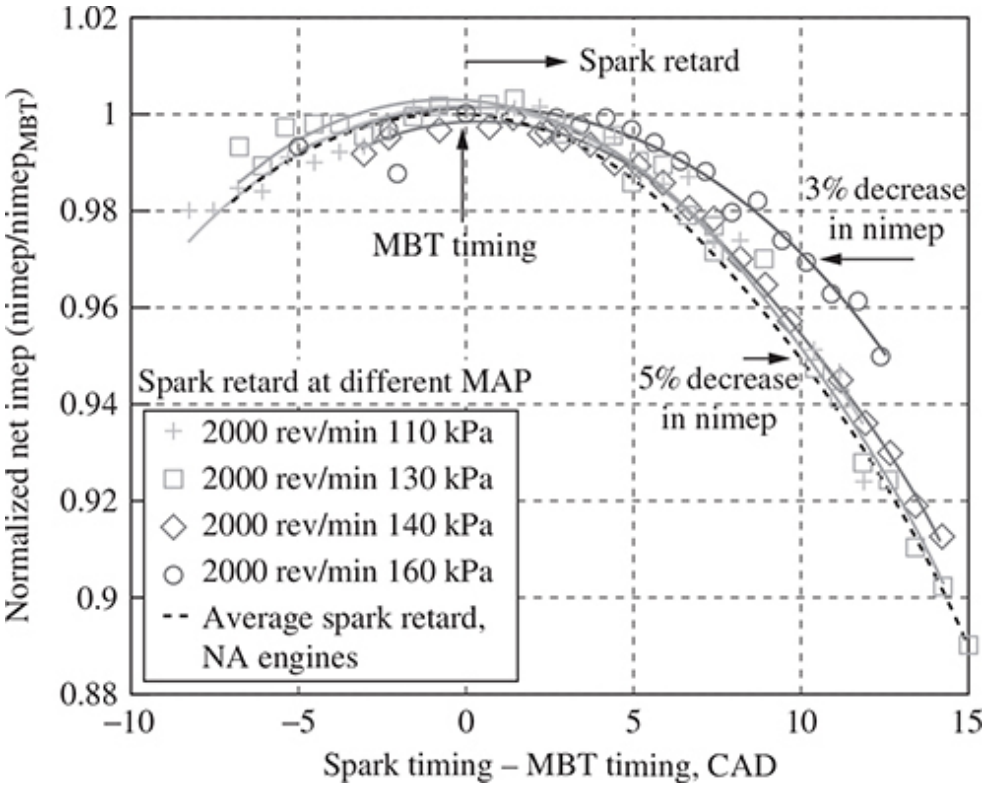


Figure 15.10 Effect of spark retard from MBT spark timing on normalized net mean effective pressure, for several intake manifold pressures (absolute values), for naturally-aspirated and turbocharged SI engines, E25 fuel; 3 and 5% decrease from net imep at MBT timing indicated.[14]

Spark timing affects the value of the peak cylinder pressure, and therefore affects peak unburned and burned gas temperatures (see Sec. 9.2.1). Retarding spark timing from the optimum reduces these variables. Retarded timing is used therefore for NO x emission control (see Fig. 11.12 and accompanying text) and to avoid knock (see Sec. 9.6.1). The exhaust

temperature is also affected by spark timing. Retarding timing from MBT increases late-expansion-stroke exhaust temperatures, so both engine efficiency and heat loss to the combustion chamber walls (see Fig. 12.12 *b*) are decreased. Retarded timing may be used at engine idle to bring the ignition point closer to TC where conditions for avoiding misfire are more favorable.

Retarding spark reduces peak cylinder pressures, and thus peak burned gas temperatures so it impacts engine-out emissions. While the effect on CO emissions is small (see Sec. 11.3), retarding spark timing from MBT timing usefully reduces NO*x* and HC emissions. Figure 11.12 shows the effect on NO*x*: retarding spark 5 degrees CA from MBT timing reduces engine-out emissions by about 20% at stoichiometric light-load conditions. Retarded timing also reduces engine-out HC emissions, as shown in Fig. 15.11 for a range of *A/F*, rich to lean. The primary effect of spark retard on HC emissions is that it raises late-expansion-stroke in-cylinder and exhaust gas temperatures, and thus increases the amount of afterburning of HC left unburned at the end of the normal combustion process. The increase in exhaust gas temperature is a consequence of the reduced engine efficiency. Note that the exhaust gas temperature level impacts the light-off and steady-state catalyst temperature, and CO and HC oxidation efficiency.

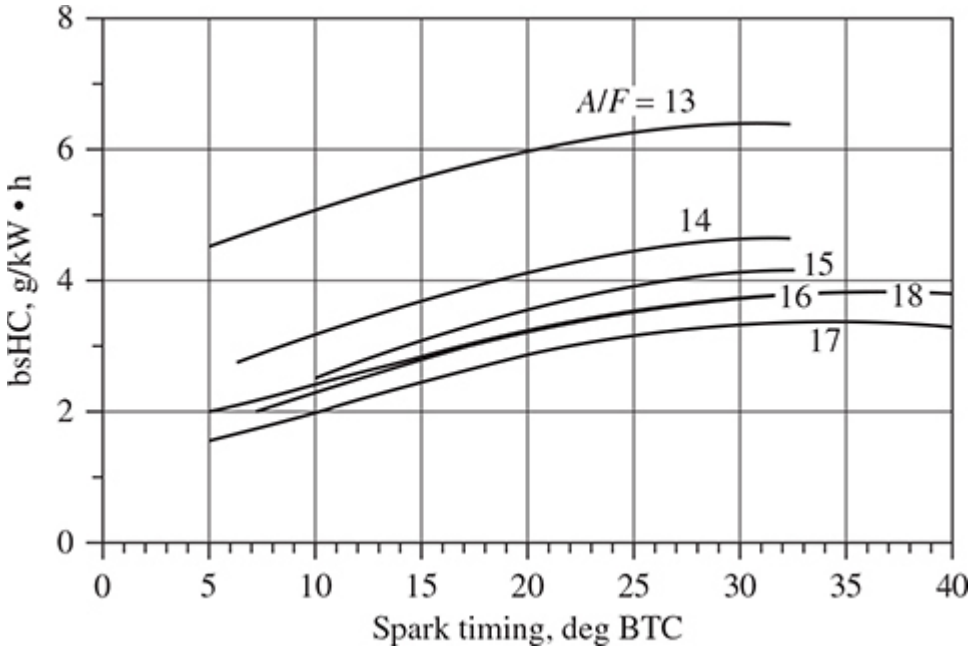


Figure 15.11 Brake-specific hydrocarbon engine-out emissions for a 5-liter gasoline V-8 spark-ignition engine as a function of spark advance, for a range of *A/F* ratios. Stoichiometric *A/F* = 14.6. Operating conditions: bmep = 2.6 bar, speed = 1500 rev/min, without EGR.

15.3.2 Mixture Composition

The unburned mixture in the engine cylinder consists of fuel (normally vaporized), air, and burned gases. The burned gas fraction is the residual gas plus any recycled exhaust used for NO *x* control. Mixture composition during combustion is most critical, since this determines the development of the combustion process which governs the engine's operating characteristics. While efforts are made to produce a uniform mixture within the cylinder, some nonuniformities remain (see Sec. 9.4.2). In a given cylinder, cycle-by-cycle variations in average charge composition exist. Also, within each cylinder in a given engine cycle, the fuel, air, EGR, and residual gas are not completely mixed, and composition nonuniformities across the charge may be significant.[f] These together produce variations in composition at the spark plug location (the critical region since the early stages of flame development influence the rest of the combustion process) which can be of order ±10% peak-to-peak. In addition, in multicylinder engines, the average air, fuel, and EGR flow rates to each cylinder are not identical. Typical cylinder-to-cylinder variations have standard deviations of ±5% of the mean for airflow rate and fuel flow rate (giving up to ±7% variation in the air/fuel ratio) for steady-state engine operation. EGR cylinder-to-cylinder flow rates may have higher variability. Under unsteady engine operating conditions these variations could be higher.

It is necessary to consider the effect of mixture composition changes on engine operating and emissions characteristics in two distinct regimes: (1) wide-open throttle (WOT) or full load and (2) part throttle or load. At WOT, the engine airflow is the maximum that the engine will induct.[g] At wide-open throttle, fuel flow can be varied, but airflow is set by engine design variables and speed. At part throttle, airflow, fuel flow, and EGR flow can be varied. Evaluation of mixture composition changes at part load are best done at fixed (brake) load and speed, that is, under conditions where the engine provides the desired torque level at the specified speed. To maintain torque (or bmep)

constant as mixture composition is varied normally requires changes in throttle setting (and if EGR is varied, changes in EGR flow-control valve setting). This distinction between part-load comparisons at specified torque or bmep, rather than at constant throttle settings (which gives essentially constant airflow), is important because the pumping work component of engine friction will vary at constant engine load as mixture composition changes. At constant throttle setting and speed, the pumping work remains essentially unchanged.

Relative Air/Fuel or Equivalence Ratio

Mixture composition effects are usually discussed in terms of the air/fuel ratio (or fuel/air ratio) because in engine tests, the air and fuel flow rates to the engine can be measured directly and because the fuel system is designed to provide the appropriate fuel flow for the actual airflow at each speed and load. However, the relative proportions of fuel and air can be stated more generally in terms of the fuel/air equivalence ratio ϕ [the actual fuel/air ratio normalized by dividing by the stoichiometric fuel/air ratio; see Eq. (3.8)] or the relative air/fuel ratio λ [see Eq. (3.9)]. The combustion characteristics of fuel-air mixtures and the properties of combustion products, which govern engine performance, efficiency, and emissions, correlate best for a wide range of fuels relative to the stoichiometric mixture proportions. Either the relative air/fuel ratio or the equivalence ratio can be used as the defining parameter: see Eq. (7.1) . With gasoline engines, the relative air/fuel ratio λ is more commonly used.

The theoretical basis for understanding the effect of changes in the equivalence ratio is the fuel-air cycle results in Figs. 5.9 and 5.10, where the indicated fuel conversion efficiency and mean effective pressure are shown as a function of the fuel/air equivalence ratio, ϕ. The mean effective pressure peaks slightly rich of stoichiometric, between $\phi = 1$ and 1.1. Due to dissociation at the high gas temperatures following combustion, molecular oxygen is present in the burned gases under stoichiometric conditions so some additional fuel can be added and partially burned. This increases the temperature and the number of moles of the burned gases in the cylinder. These effects increase the pressure to give increased power and mep. Fuel conversion efficiency decreases approximately as $1/\phi$, as the mixture is richened above stoichiometric ($\phi > 1$) due to the decreasing combustion efficiency associated with the richening mixture.

For mixtures lean of stoichiometric, the theoretical fuel conversion efficiency increases linearly as ϕ decreases below 1.0. Combustion of mixtures leaner than stoichiometric produces products at lower temperature, and with less dissociation of the triatomic molecules CO_2 and H_2O, and with excess oxygen and more nitrogen (both diatomic molecules). Thus the *fraction* of the chemical energy of the fuel which is released as sensible energy near TC is greater. As a consequence, a greater *fraction* of the fuel's energy is transferred as work to the piston during expansion, and the fraction of the fuel's available energy rejected to the exhaust system decreases since, with higher ratio of specific heats, γ the expansion stroke work as a fraction of the fuel's chemical energy increases (see Sec. 5.7). There is a discontinuity in the fuel conversion efficiency and imep curves at the stoichiometric point: the burned gas composition is significantly different on the rich and the lean sides of $\lambda = 1$.

Summarizing this behavior, Figure 15.12 shows gross indicated specific fuel consumption data for a six-cylinder spark-ignition engine at wide-open throttle and 1200 rev/min, along with values of gross indicated mean effective pressure and fuel conversion efficiency derived from the isfc data. In these engine tests, the fuel-air mixture was prepared with a heated vaporizing tank to ensure intake-mixture uniformity. Use of a fuel vaporizing and mixing tank removes any mixture nonuniformity and extends the lean misfire limit to about $\phi = 0.7$, $\lambda = 1.4$. However, $\eta_{f,\,i}$ starts to decrease before this point. The reasons are that cycle-to-cycle pressure variations and the total duration of the burning process increase as the mixture becomes leaner: both these factors degrade engine efficiency. Since the spark advance is set for the average cycle, increasing cycle-to-cycle dispersion produces increasing imep losses in "non-average" cycles due to non-optimum spark timing. The lengthening burn duration directly decreases efficiency, even in the absence of cyclic variations.

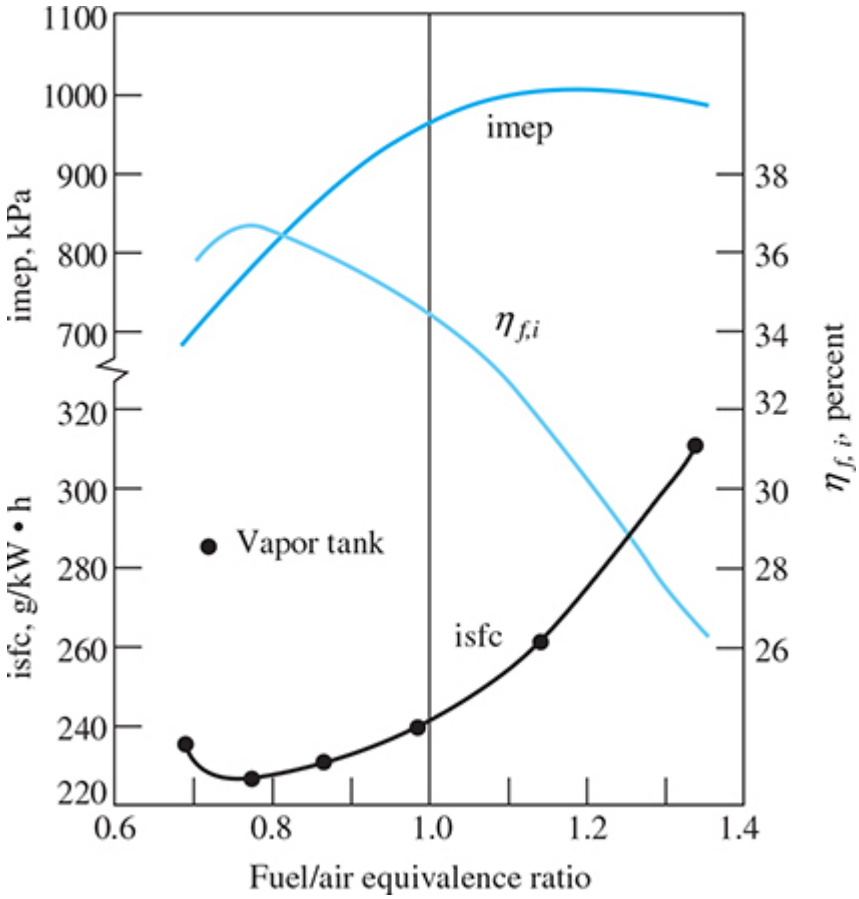


Figure 15.12 Effect of the fuel/air equivalence ratio on indicated mean effective pressure, specific fuel consumption, and fuel conversion efficiency of a naturally-aspirated six-cylinder spark-ignition engine at wide-open throttle and 1200 rev/min. Data from a gasoline-fueled SI engine equipped with vapor tank. [15]

Engine fuel consumption and efficiency well lean of stoichiometric depend strongly on the engine's combustion or burn rate. Figure 15.13 *a* shows bsfc results obtained from a thermodynamic-based computer cycle simulation of the spark-ignition engine operating cycle (at 325 kPa bmep and 1400 rev/min).[13] The engine's fuel consumption depends on the combustion characteristics of the chamber. A slow-burning chamber ($\Delta\,\theta_b = 100°$) shows much higher bsfc: more moderate burn rates (60°) significantly improve bsfc. Very fast burning chambers further improve bsfc but not by that much. As the mixture is leaned out from stoichiometric, the burn rate lengthens, but

initially only slowly. Thus improvements in bsfc slow down, bsfc reaches a minimum, and then rises as burn duration increased at an accelerating rate.

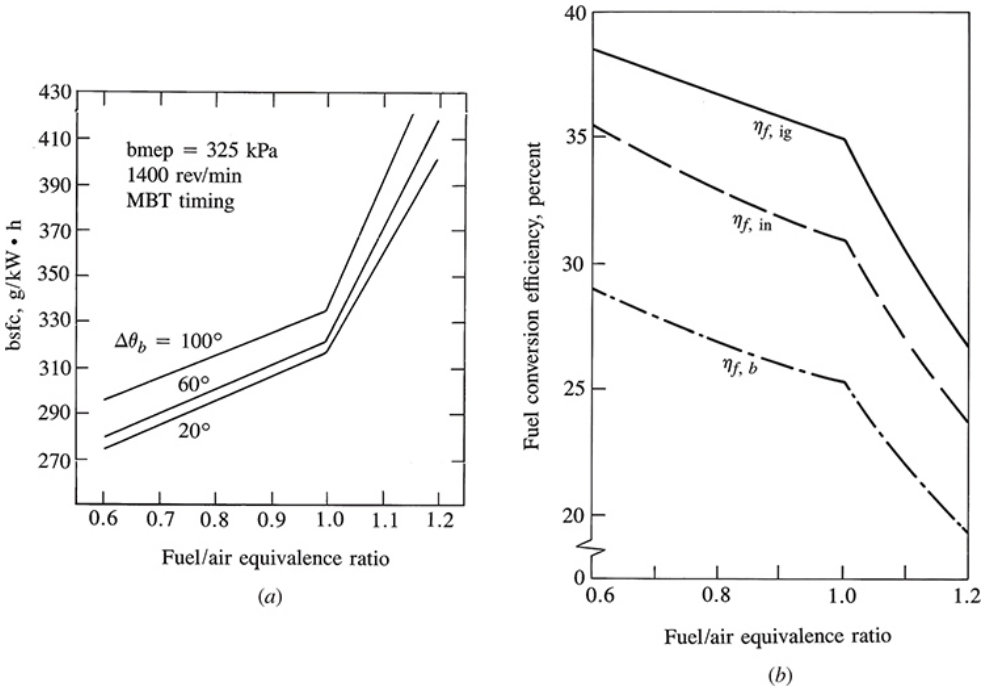


Figure 15.13 (a) Effect of combustion-chamber burn rate on spark-ignition engine brake-specific fuel consumption; predictions from a thermodynamic-based computer simulation of engine cycle for 5.7-liter eight-cylinder engine at bmep of 325 kPa and 1400 rev/min with MBT spark timing. (*b*) Gross and net indicated, and brake, fuel conversion efficiencies (also predicted by a thermodynamic-based cycle simulation) at constant part-load bmep (325 kPa) and speed (1400 rev/min) for a fixed burn duration (0 to 100%, 60 degrees CA). [13]

At part load, as the air/fuel ratio is varied at constant brake load, the pumping work varies: this also contributes to the brake-specific fuel consumption and efficiency variation with equivalence ratio. Figure 15.13 *b* shows the gross and net indicated fuel conversion efficiencies, and brake efficiency, as a function of equivalence ratio at a part-throttle constant load and speed point (325 kPa bmep and 1400 rev/min), again calculated using a thermodynamic-based computer simulation of the engine's operating cycle. The difference between the net and gross indicated curves illustrates the

magnitude of the effect of the pumping work changes. As a naturally-aspirated engine is leaned out, the pumping work impact is reduced.

Note that engine data usually show a smooth transition between the rich and lean characteristics as the stoichiometric point is crossed, whereas the calculated sfc and efficiency characteristics show a discontinuity in slope. The difference is due to cylinder-to-cylinder and cycle-by-cycle mixture composition variations and to cycle-by-cycle cylinder pressure variations which exist (though to a lesser extent) even in the absence of these mixture variations. Averaging over these variations smooths out the theoretical discontinuity in slope at λ or $\phi = 1.0$.

Relative air/fuel ratio effects on a spark-ignition engine's behavior over the full load and speed range have been explained above from the point of view of performance and efficiency. However, since engine-out emissions also depend on λ, emission control requirements may dictate a different engine calibration. The mixture requirements are usually discussed in relation to *steady* and to *transient* engine operation. Steady operation includes operation at a given speed and load over several engine cycles with a warmed-up engine. Transient operation includes engine starting, engine warm-up to steady-state temperatures, and changing rapidly from one engine load and speed to another. The mixture requirements of the engine as defined by the composition of the combustible mixture at the time of ignition, while they vary somewhat with speed and load, are essentially the same for all these operating modes. [h] However, the methods used to prepare the mixture prior to entry to the cylinder must be modified in the transient modes when liquid fuels are used, to allow for variations in the liquid fuel flow and fuel evaporation rate in the intake port and valve region as the airflow varies and as the pressure in the manifold, and port and valve and temperature change. The transient fuel metering requirements for adequate mixture preparation are discussed in Chap. 7 .

The exhaust gas temperature varies with relative air/fuel. The exhaust gas temperature also varies continuously as the gas leaves the engine cylinder and flows through the exhaust port, and the manifold and pipe (see Sec. 6.5), so an appropriate definition of an average exhaust gas temperature should be used in quantifying this variation. However, time-averaged thermocouple measurements from specific locations in the exhaust system can provide useful information on trends. Figure 14.10 shows examples of predictions of the enthalpy-averaged exhaust gas temperature at the exhaust port exit as a

function of equivalence ratio compared with time-averaged measurements. The enthalpy-averaged temperature is defined by Eq. (6.22) . These are typically 50 to 100 K higher than time-averaged measurements. The exhaust temperature peaks at the stoichiometric point and decreases as the mixture is richened and leaned on either side.

The relative air/fuel ratio is an important parameter controlling spark-ignition engine emissions. The critical factors affecting engine-out emissions, that are governed by the relative air/fuel ratio, are the oxygen concentration and the temperature of the burned gases. Excess oxygen is available in the burned gases lean of stoichiometric. The maximum burned gas temperatures occur slightly rich of stoichiometric at the beginning of the expansion stroke, and at the stoichiometric composition at the end of expansion and during the exhaust process. Figure 11.2 illustrates the general trends in engine-out emissions with equivalence ratio which have already been discussed. Of course, the behavior of the exhaust aftertreatment system largely controls the overall engine emissions. That behavior is influenced by the composition and temperature of the engine-out exhaust. (See Sec. 11.6.1 .)

Historically, gasoline engines have been operated fuel rich at high loads. There are several reasons why. Air utilization is maximized, since all the oxygen is used up. Evaporation of the excess fuel provides additional charge cooling. The working fluid's ratio of specific heats, γ, is changed advantageously: during compression of the unburned mixture γ is lower so the temperature rise is less; during the burned gas expansion, γ is higher (less CO_2 and H_2O) so expansion stroke work is larger. With direct-injection, the evaporative cooling impact is significantly greater. Richer operation also reduces the exhaust gas temperature, which is important with turbocharged gasoline engines. However, for emissions control and fuel economy reasons, stoichiometric operation all the way up to wide-open throttle is to be preferred.

Figure 15.11 also shows the effect of variations in equivalence ratio, for rich through stoichiometric to lean mixtures, on unburned hydrocarbon emissions. For rich mixtures, engine-out HC emissions are significantly higher. This is primarily due to the lack of oxygen for afterburning of any unburned hydrocarbons that escape the primary combustion process, within the cylinder and the exhaust system. HC emissions decrease as the stoichiometric point is approached: increasing oxygen concentration and increasing expansion and exhaust stroke temperatures result in increasing HC

burnup. For moderately lean mixtures, HC emission levels vary little with relative air/fuel ratio. Decreasing fuel concentration and increasing oxygen concentration essentially offset the effect of decreasing bulk gas temperatures. As the lean operating limit of the engine is approached, combustion quality deteriorates significantly and HC emissions start to rise again (e.g., *A/F*, 17 to 18) due to the occurrence of occasional partial-burning cycles. For still leaner mixtures, HC emissions rise more rapidly due to the increasing frequency of partial-burning cycles, and even the occurrence of completely misfiring cycles (see Sec. 9.4.3). The equivalence ratio at which partial-burning and misfiring cycles just start to appear depends on details of the engine combustion and fuel preparation systems, as well as the load and speed point.

The effect of equivalence ratio variations on CO emissions has already been explained in Sec. 11.3 (see Fig. 11.19). For rich mixtures, CO levels are high because complete oxidation of the fuel carbon to CO_2 is not possible due to insufficient oxygen. For lean mixtures, CO levels are approximately constant at a low level of about 0.5% or less.

The above discussion suggests that if an engine can be designed and operated so that its stable operating limit under the appropriate part-load conditions is sufficiently lean, excellent fuel consumption and substantial control of engine-out NO, HC, and CO emissions can be achieved. Such an approach requires good control of mixture preparation and a fast-burning combustion chamber design (see Sec. 15.4.1). However, this lean-engine approach is not compatible with the three-way catalyst system (see Sec. 11.6.3) which, with close-to-stoichiometric mixtures, achieves very substantial reductions in NO, HC, and CO emissions.

Exhaust Gas Recycle

Exhaust gas recycle (EGR) is the principal technique used for control of SI engine in-cylinder NO *x* formation (see Sec. 11.2.3). A fraction of the exhaust gases are recycled through a control valve from the exhaust to the engine intake system. The recycled exhaust gas is usually mixed with the intake air just downstream of the throttle valve. EGR acts, at part load, as an additional diluent in the unburned gas mixture, thereby reducing the peak burned gas temperatures and NO formation rates. Note that it is the total burned gas fraction in the unburned mixture in the cylinder that acts as a diluent. These burned gases are comprised of both residual gas from the

previous cycle and exhaust gas recycled to the intake. As described in Sec. 6.4 , the residual gas fraction is influenced by load and valve timing (especially the extent of valve overlap) and, to a lesser degree, by the air/fuel ratio and compression ratio. The total burned gas mass fraction is given by Eq. (4.3) . Since the burned gases dilute the unburned mixture, the absolute temperature reached after combustion varies inversely with the burned gas mass fraction. Hence increasing the burned gas fraction reduces the rate of formation of NO emissions.

Figure 11.9 shows the effect on engine-out NO emissions of increasing the burned gas fraction by recycling exhaust gases to the intake system. Substantial reductions in NO concentrations are achieved with 10 to 25% EGR. However, EGR also reduces the combustion rate which makes stable combustion more difficult to achieve (see Sec. 9.4.3 and Fig. 9.36). The amount of EGR a particular combustion chamber will tolerate depends on its combustion characteristics, the engine speed and load, and the relative *A/F* ratio. EGR percentages in the 15 to 25 range are about the maximum amount of EGR a spark-ignition engine will tolerate under normal part-throttle conditions. Faster-burning engines will tolerate more EGR than slower-burning engines. Because of the decrease in burn rate and increase in cycle-by-cycle combustion variations, hydrocarbon emissions increase with increasing EGR, as shown in Fig. 11.31. At first, the increase in HC is modest and is due primarily to decreased HC burn-up due to lower expansion and exhaust stroke gas temperatures. The HC increase becomes more rapid as slow combustion, partial burning, and eventually misfire, in turn, occur with increasing frequency. EGR has no significant effect on engine CO emissions.

The effect of EGR on engine performance and efficiency, for mixtures with $\lambda \geq 1.0$, is similar to that of adding excess air. Both EGR and excess air dilute the unburned mixture. In practice, EGR is only used at part-throttle conditions, and since three-way catalysts are widely used where NO x emission constraints are severe, greatest attention has been focused on dilution with EGR at $\lambda \approx 1.0$. Figure 15.14 shows the effect of increasing EGR on bsfc and enthalpy-mean exhaust temperature [defined by Eq. (6.22)] at constant bmep, predicted by using a thermodynamic-based computer simulation of the engine's operating cycle. Predictions made for different burn durations are shown, at MBT spark timing, for a stoichiometric mixture. At constant burn duration, bsfc and exhaust temperature decrease with

increasing EGR. Only for very long combustion processes is the burn rate especially significant. This improvement in fuel consumption with increasing EGR is due to three factors: (1) reduced pumping work as EGR is increased at constant brake load (fuel and airflows remain almost constant; hence intake pressure increases); (2) reduced heat loss to the walls because the burned gas temperature is lower; and (3) decreased amount of dissociation in the lower temperature burned gases which allows more of the fuel's chemical energy to be converted to sensible energy near TC (the value of $\gamma = c_p/cv$ is increased). These different effects are comparable in magnitude.

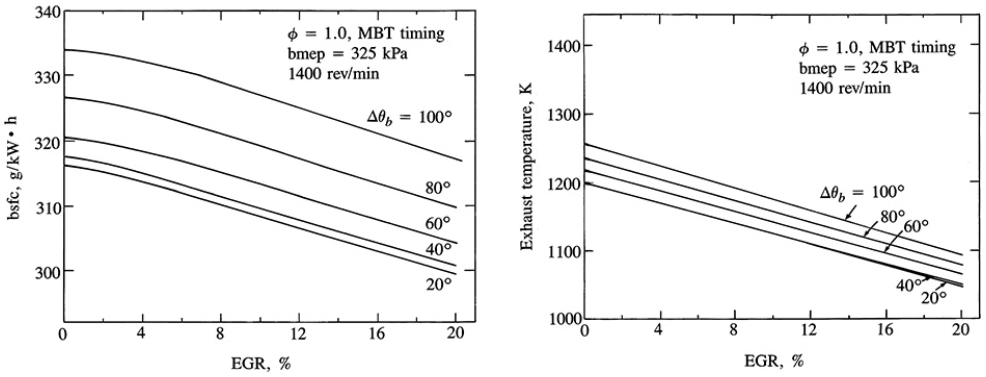


Figure 15.14 Effect of recycled exhaust gas on brake-specific fuel consumption and exhaust gas temperature at constant bmep and speed, stoichiometric mixture, and various burn durations (20 to 100°). Predictions from thermodynamic-based cycle simulation. [13]

Addition of EGR to the intake mixture lengthens both the flame development and propagation processes (and thus requires increasing spark advance to maintain the MBT timing requirement). Since faster-burning combustion chambers provide significant bsfc reductions until, approaching 15 to 20% EGR, the burn rate and combustion quality deteriorate much more rapidly. For slower-burning combustion chambers, the tolerance to dilution with EGR is much less. The benefits of faster-burning combustion systems are discussed in Sec. 15.4 .

Lean Operation and EGR

Lean operation of spark ignition engines has long been viewed as an attractive approach from the perspective of improved fuel consumption. It is primarily a part-load opportunity. At high load, lean operation does not fully

utilize the air inducted into the engine cylinder: so maximum torque and power levels in naturally-aspirated engines of a given size are reduced. In turbocharged engines, this less-than-full intake air utilization could be offset by using higher compressor boost pressures. The advantages of lean operation are higher engine indicated efficiencies (lower fuel consumption) and, potentially, lower engine-out emissions (though this depends on the details). The disadvantages are primarily worse combustion quality (slower burn rates, higher cyclic variability), and the likely need for a highly effective lean NO x catalyst (the very efficient three-way catalyst technology cannot be used with lean engine operation). Assessments of the potential benefits have been made. Of course, they depend on the base case assumed and what is held constant in such a comparison.

Both excess air (with lean operation) and EGR are diluents. They can be used separately, or they can be combined. One approach, extensively explored, is a DI gasoline engine that operates stratified (and very lean) at light load with some EGR, then homogeneous lean with EGR at mid-load, and stoichiometric at high load is illustrated on the performance map shown in Fig. 15.15. The EGR is added (in moderate amounts) to reduce peak burned gas temperatures, especially when stratified at lighter loads, when a "premixed" flame propagates from the spark plug, through the close-to-stoichiometric mixture around the plug.

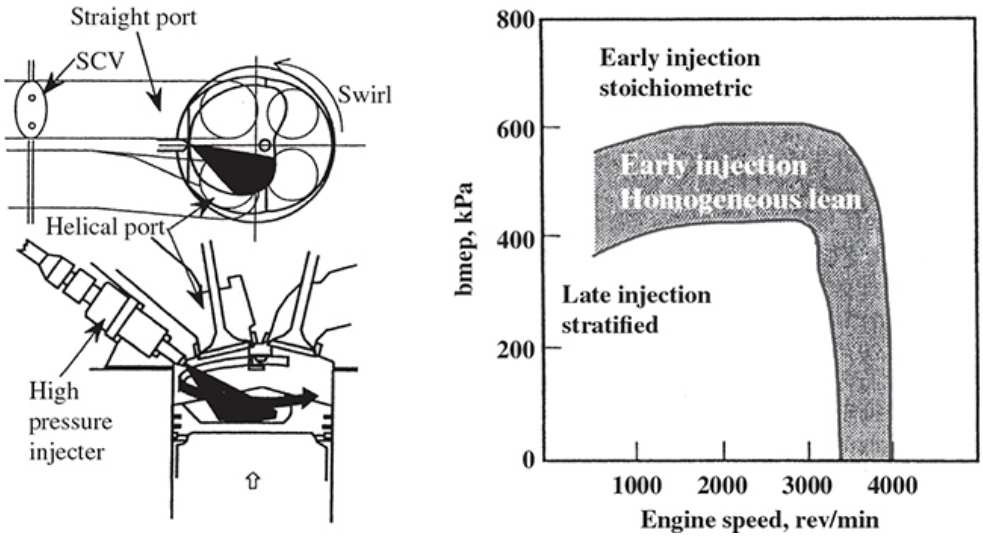


Figure 15.15 Illustrative performance map for a stratified DI gasoline engine showing the different regimes for lean stratified, lean homogeneous (both with EGR), and stoichiometric (higher load) regimes.[16]

The effects of increasing dilution with excess air, and with recycled exhaust with stoichiometric operation, on combustion stability, fuel consumption, and engine-out HC and NO *x* emissions are shown in Fig. 15.16. EGR (open circles) is highly effective at reducing NO *x* emissions: 15% EGR reduces NO *x* by about a factor of 3. However, combustion stability (COV imep) worsens significantly at these steady state operating conditions as EGR levels move beyond 20%, and bsfc goes to a minimum. At the same combustion stability level (COV imep below 1%), lean operation out to 40% dilution with excess air gives significantly higher bsNO *x*, slightly higher bsHC, but almost doubles the bsfc benefit (from about 6% to about 10%). Thus, at this 3-bar bmep light-load operating condition, the benefit of lean dilution relative to EGR dilution, at comparable combustion stability limits is an additional 6 or so percent lower bsfc. This example illustrates the approximate magnitude of the potential fuel consumption (and efficiency) benefit from lean gasoline engine operation.

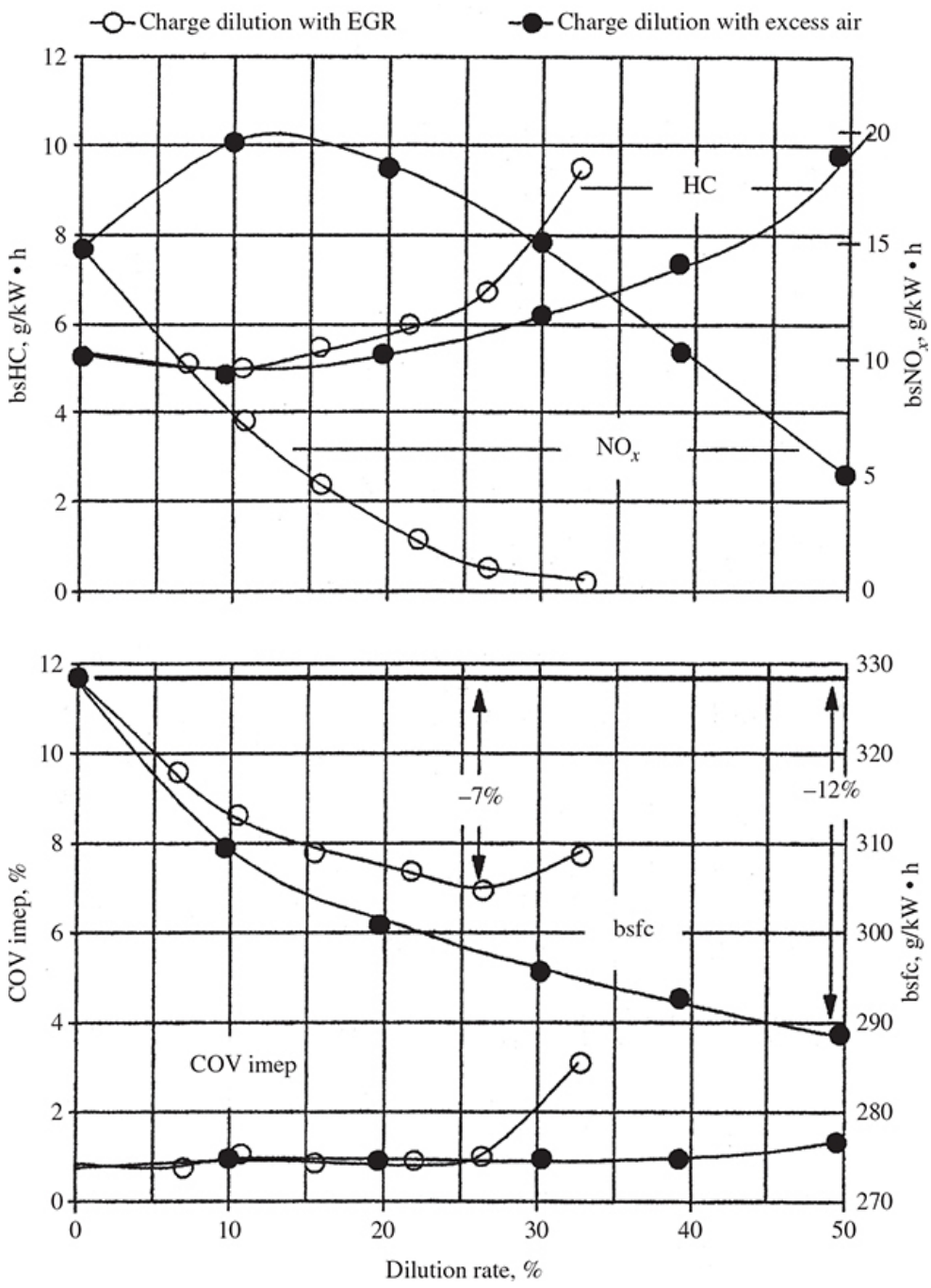


Figure 15.16 Comparison of effect of charge dilution with excess air or with EGR, on gasoline engine combustion stability, bsNO*x* and bsHC emissions and bsfc: 2-liter four-valve I-4 engine, operated at 2000 rev/min and 2 bar bmep, with MBT spark timing. EGR data with stoichiometric operation. [17]

15.3.3 Load and Speed

The operating characteristics of an internal combustion engine over its full load and speed range are often plotted as lines of constant brake-specific fuel consumption on a graph of torque or brake mean effective pressure versus engine speed. Such *performance maps* have already been introduced in Sec. 15.2.3 . Here we explore the specific effects of load and speed on the fuel consumption, efficiency and emissions of spark-ignition engines. While details differ from one engine to another, the overall shapes of these maps for gasoline engines are remarkably similar. Figure 15.17 shows the map for a 1.6-liter four-cylinder naturally-aspirated gasoline engine over its full load and speed range.

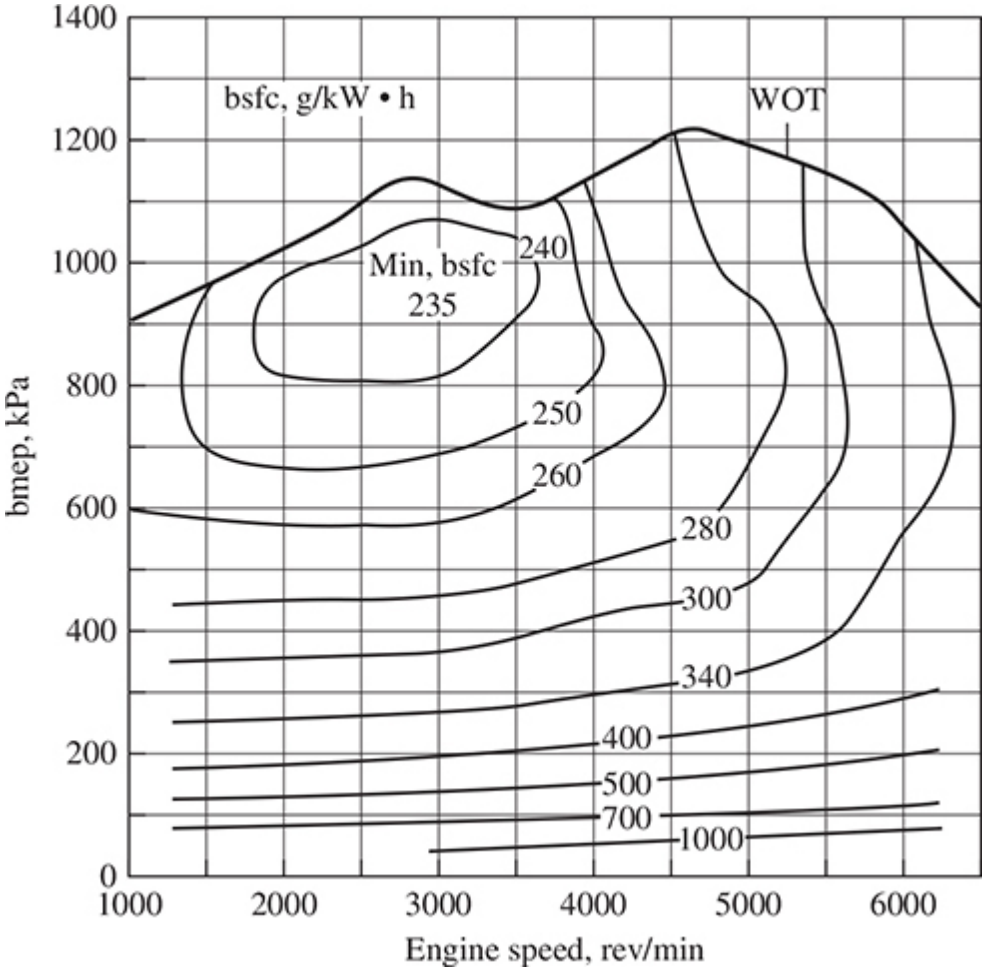


Figure 15.17 Performance map for 1.6-liter four-cylinder naturally-aspired SI engine showing contours of constant bsfc in grams per kilowatt-hour.[18]

The wide-open-throttle bmep curve versus speed has the shape expected for a naturally-aspirated SI engine, with two tuning peaks (see Fig. 6.12). Maximum bmep (about 1200 kPa) occurs at 4500 rev/min; the minimum bsfc island is located at a lower speed, just below the WOT line. These map characteristics can be understood in terms of variations in volumetric efficiency η_v, gross indicated fuel conversion efficiency $\eta_{f,\text{ig}}$, and mechanical efficiency η_m, relative A/F, EGR (if used), and the importance of heat losses and friction, via Eqs. (15.5) and (15.7). The maximum bmep curve reflects the variation with speed of η_v, the decrease of η_m as $\overline{S}_p$ increases, and the increase of $\eta_{f,\text{ig}}$ as $\overline{S}_p$ increases due to decreasing importance of heat transfer per cycle.

The bsfc contour trends can be explained as follows. Starting at the minimum bsfc point, increasing speed at constant load increases bsfc due primarily to the increasing friction mep at higher speeds (which decreases η_m). While $\eta_{f,\text{ig}}$ increases modestly as speed increases, friction increases dominate. Decreasing speed at constant load increases bsfc due primarily to the increasing importance of heat transfer per cycle (which decreases $\eta_{f,\text{ig}}$). Friction also decreases, increasing η_m, but this is secondary. Any mixture enrichment required to maintain a sufficiently repeatable combustion process and avoid knock at low engine speeds contributes too. Increasing load at constant speed from the minimum bsfc point increases bsfc if mixture enrichment is used to increase torque as the engine becomes increasingly airflow limited, or to avoid knock. With or without such enrichment, spark retard worsens bsfc modestly, also. Decreasing load at constant speed increases bsfc due to the increased magnitude of friction (due to increased pumping work), the increased importance of friction mep relative to imep, and the increasing importance of heat transfer (which decreases $\eta_{f,\text{ig}}$).

The effects of speed and load variations on NO and HC emissions are shown in Fig. 15.18. [19] NO concentrations increase moderately with increasing speed at constant load. At lower loads, the proportional increase in NO is greater than at higher loads. [13] The residual gas fraction decreases as speed increases, this effect being greater at lower inlet manifold pressures (lighter loads): see Fig. 6.26. Also, the relative importance of heat transfer per cycle is less as speed increases (see Fig. 12.25), which would also be expected to increase NO concentrations. With increasing load (at constant

speed), NO concentrations also increase. Again, as inlet manifold pressure and load increase, the residual gas fraction decreases (Fig. 6.26); also, the relative importance of heat transfer per cycle decreases with increasing load (Fig. 12.23).

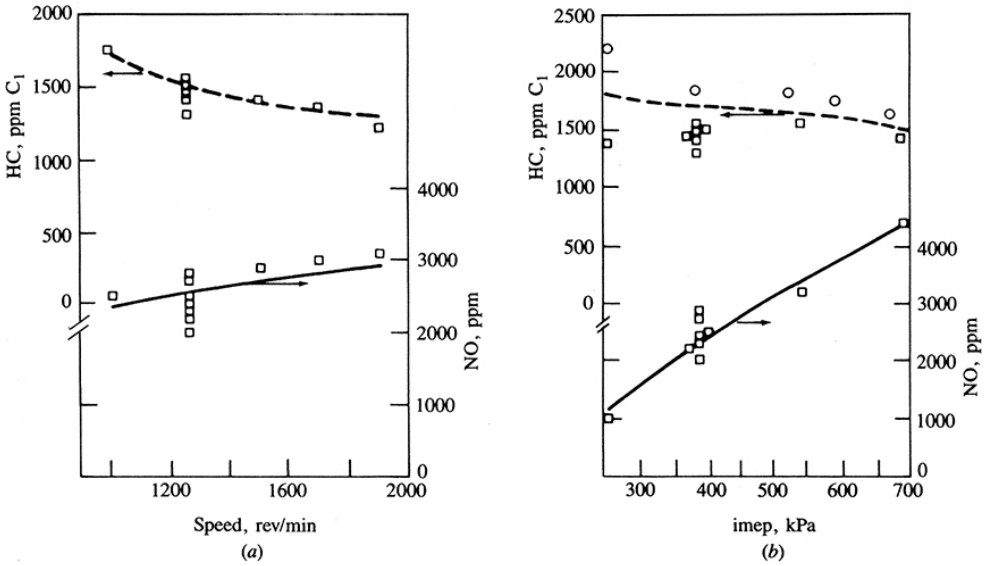


Figure 15.18 Variation in spark-ignition engine HC and NO engine-out emissions with (*a*) engine speed at 330 kPa imep and (*b*) load (or imep) at 1250 rev/min. Relative air/fuel ratio = 1.1, MBT spark timing, $r_c = 7$. [19]

The hydrocarbon concentration trends with speed and load changes are the opposite of the NO concentration trends. As indicated in Table 11.8, speed and load are likely to affect several of the HC formation mechanisms such as the in-cylinder mixing of unburned hydrocarbons which escape normal combustion with the bulk gases, and the fraction of the in-cylinder HC which escape into the exhaust. If oxygen is available, oxidation of unburned hydrocarbons both within the cylinder and in the exhaust system will be significantly enhanced by increases in speed since the expansion stroke and exhaust process gas temperatures increase due to the reduced significance of heat transfer per cycle with increasing speed. This more than offsets the reduced residence time in the cylinder and in the exhaust. Measurements of the percent HC reacted in the exhaust port as a function of engine speed show the same proportional reduction as in the exhaust emissions data in Fig. 15.11. [20] The rationale behind the variation of HC with load is less clear. As

load increases at constant speed, expansion and exhaust stroke temperatures increase, and the in-cylinder oxidation rate, if oxygen is available, will increase. However, as the exhaust gas flow rate increases, the residence time in critical sections of the exhaust system decreases and a reduction in exhaust port HC oxidation occurs. [21] The net trend is for the engine-out HC concentration to decrease modestly as load is increased.

We now compare turbocharged gasoline engine behavior with naturally-aspirated engine behavior. With a turbocharger, at higher loads when MAP increases above one bar, the density of the air entering the cylinder is increased, so imep will increase as given by Eq. (15.5) . Thus, to replace a naturally-aspirated engine and provide the same torque, the turbocharged engine can be downsized—its displaced volume significantly decreased. Figure 15.19 compares the performance maps of a 1.8-liter turbocharged direct-injection spark-ignition (DISI) engine with a 2.6 liter V-6 naturally-aspirated port-fuel-injected (PFI) engine. Both have the same maximum torque (about 260 N·m), but different bmep values (TC DISI 1850 kPa, NA PFI 1200 kPa). Their low-speed torque levels, 1250 to 2500 rev/min, are significantly different.

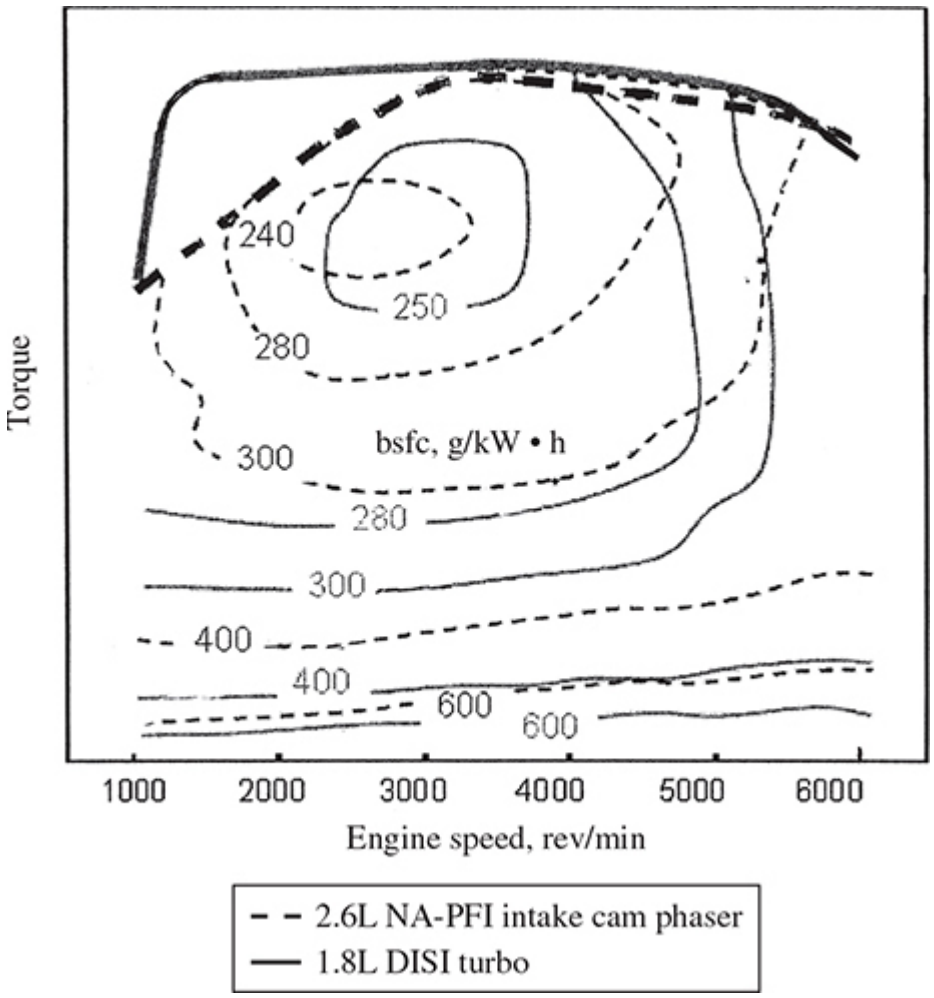


Figure 15.19 Comparison of performance maps of a 2.6-liter naturally-aspirated port-fuel-injected gasoline V-6 engine (with cam phaser), and a 1.8-liter I-4 turbo direct-injection SI engine which have the same maximum torque and power. Wide-open-throttle lines (solid TC engine; dashed NA engine) match from about 3000 to 5500 rev/min. Lines of constant bsfc (g/kW·h) are shown (solid lines TC engine; dashed lines NA engine). [22]

The minimum bsfc of the turbo DISI engine is about 245 g/kW·h close to that of the naturally-aspirated engine (235 g/kW·h). However, when lightly loaded, at about one-quarter the maximum torque and 2000 rev/min, the TC engine has significantly lower bsfc (300 g/kW·h) than the larger-displacement NA engine (about 365 g/kW·h), an 18% reduction.

Several factors contribute to these differences, turbocharged (DI) versus naturally-aspirated (PFI). The compression ratios are similar: 10.5 for the DI

turbocharged engine, 10.0 for the PFI NA engine. Turbocharging worsens the high load knock problem; direction-injection of fuel through evaporative cooling of the in-cylinder charge helps here. The biggest factor in this fuel consumption difference is the reduced impact of engine friction. The total friction mep values for the two types of engines are comparable: at part load, the NA engine has higher pumping mep, but the higher pressures (at the same torque) in the smaller displacement TC engine increases the mechanical friction mep, modestly. The much larger effect is the reduced size of the TC engine (downsized by 30%) which decreases the *magnitude* of the TC engine's friction *torque* by the same percentage. Since the bmep of the TC engine is increased by some 40% but the fmep changes little, the mechanical efficiency (η_m) increases significantly. The friction related benefits of turbocharging on part-load bsfc and efficiency, due to the major engine downsizing that usually occurs, are very significant.

Figure 15.20 shows the brake fuel conversion efficiency of a 2.0-liter 4-cylinder turbocharged DI gasoline engine, as a function of load at 2000 rev/min. Data for two fuels (for stoichiometric mixtures of each) are shown. Both the ethanol (E85) and the gasoline behaviors are essentially the same, with brake fuel conversion efficiencies at 2000 rev/min above 32% for bmep values above about 600 kPa. Efficiencies are still close to 30% for typical automotive part loads of about a quarter the maximum bmep. Note for these two fuels the brake efficiency contours on their performance maps are essentially the same. Both these fuels have high octane ratings, so engine performance at high output is not constrained by knock. Thus the performance map is essentially "fuel neutral" because the chemical energy content of their stoichiometric mixtures is almost the same, and modest differences in their burn rates in these fast-burning engines (with optimized MBT spark timing) have little effect on engine efficiency. Note however that specific fuel consumption numbers (g/kW·h) for these two fuels are significantly different because the lower heating value, Q_{HV}, of E85 is only about 70% of gasoline's heating value. [i] Thus, until knock onset occurs at high load, these performance maps for a given type of engine, expressed in brake fuel conversion efficiency at a given relative air/fuel ratio, are essentially independent of fuel.

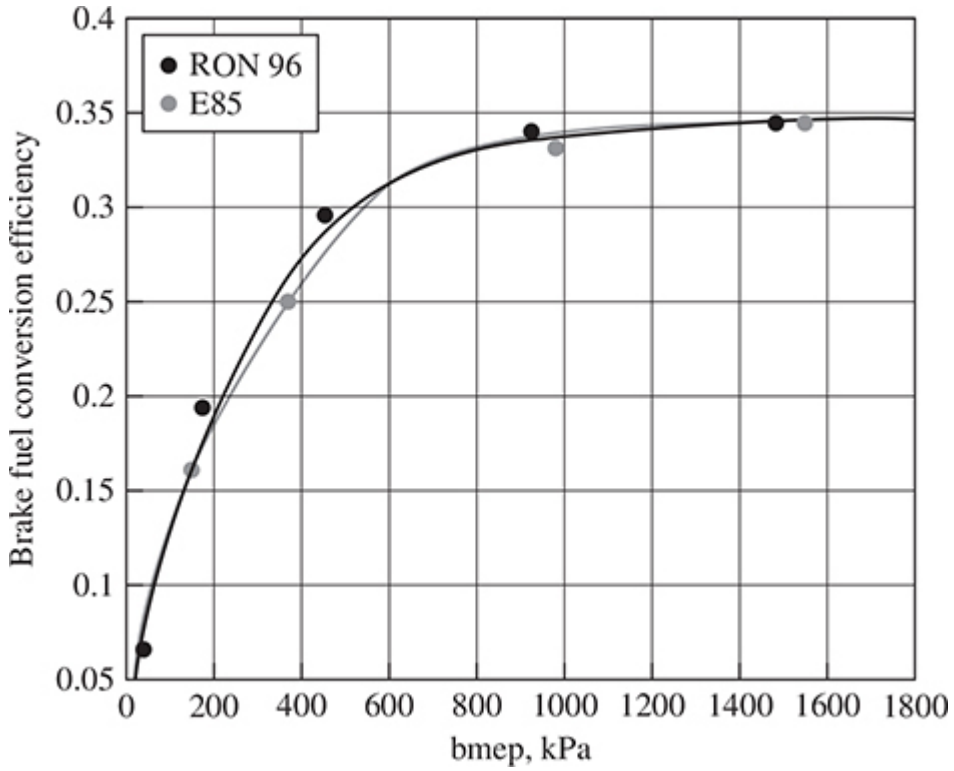


Figure 15.20 Brake fuel conversion efficiency of a 2.0-liter four-cylinder turbocharged DI spark-ignition engine as a function of load (bmep) at 2000 rev/min with gasoline (RON 96) and with ethanol (E85).[23]

Figure 15.21 compares the bsfc and the brake fuel conversion efficiency of a DI turbocharged 3.56-liter V-6 gasoline engine, with a 10.5:1 compression ratio with a 5.7-liter V-8 naturally-aspirated PFI engine over the torque range of these engines, at 2000 rev/min, which produce comparable maximum power (DI boost 283 kW, PFI NA 298 kW) and torque (DI boost 560 N·m, PFI NA 536 N·m). The maximum bmep values for the two engines are 1980 and 1180 kPa, for the DI boost and PFI NA engines, respectively. Note that the brake efficiencies at full load are comparable (36%); however, the more rapid decline in brake efficiency (below about half maximum torque for the NA engine) is delayed significantly with the boosted engine.

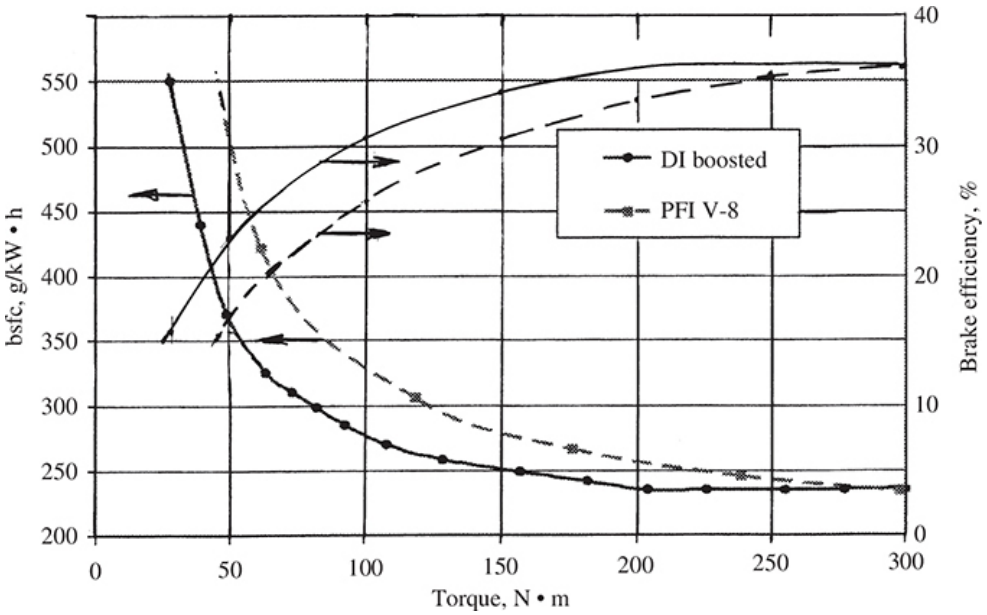


Figure 15.21 Brake-specific fuel consumption and fuel conversion efficiency for a turbocharged and a naturally-aspirated gasoline engine, of comparable torque and power: 3.5-liter V-6 boosted DI TC engine and a 5.7-liter V-8 PFI NA engine. Their maximum power and torque are closely comparable. Stoichiometric operation at 2000 rev/min. [24]

Note that in turbocharged gasoline engines, maximum power and torque are often limited by maximum cylinder pressure constraints or knock. To implement these limits (since the exhaust gases contain plenty of thermal energy to drive the turbocharger to give really high boost), a turbine bypass valve or *wastegate* is employed to limit air compression in the compressor. Section 6.8.5 discusses engine/turbocharger matching and illustrates the total system configuration with a wastegate.

The above discussion illustrates how both the practical engine performance parameters (torque and power) and the normalized parameters (bmep, kW/liter, bsfc) need to be used together to explore and compare boosted gasoline engines (turbocharged and supercharged), and naturally-aspirate gasoline engines, with their different displaced volumes (for the same practical performance). It also indicates the value of calculating the important dimensionless engine efficiencies (gross and net indicated fuel-conversion efficiency, mechanical efficiency, and brake fuel-conversion efficiency) and using these efficiencies to explore the opportunities for improving engine efficiency and understanding engine performance

opportunities and constraints, over the full performance map. It is useful in examining the role of individual engine variables on performance and efficiency to ask which of the various efficiencies (including volumetric efficiency) that particular variable impacts from a fundamental perspective. For example, compression ratio, which we examine next, directly affects the indicated work transfer to the piston and the indicated fuel conversion efficiency. Since it only modestly affects engine friction, so its impact on brake efficiency is not as direct.

15.3.4 Compression Ratio

The ideal cycle analysis in Chap. 5 showed that indicated fuel conversion efficiency increased continuously with compression ratio, according to Eq. (5.31) . With $\gamma = 1.3$, this relation also matches closely the fuel-air cycle predictions with $\phi \approx 1.0$. However, in an actual engine other processes which influence engine performance and efficiency vary with changes in compression ratio: namely, combustion rate and stability, heat transfer, and friction. Over the load and speed range, the relative impact these processes have on power and efficiency varies also. Hence, the applicability of Eq. (5.31) is open to question. Also, while the geometric compression ratio (ratio of maximum to minimum cylinder volume) is well defined, the actual compression and expansion processes in engines depend on valve timing details and the importance of flow through the valves while they are opening or closing (which depends on engine speed). Of course, our ability to increase the compression ratio is limited by the octane quality of available fuels and knock (see Sec. 9.6.1).

Note also that for this ideal cycle, the expansion ratio (TC to BC) equals the compression ratio. It is plausible that the term compression ratio became standard because, long ago, poor quality fuels severely constrained the amount of compression that could be realized before knock occurred. However, the magnitude of the expansion-stroke work in this ideal cycle is about five times the compression-stroke work. So it is the *expansion ratio* and not the compression ratio that has the primary effect on engine efficiency. In actual engines the effective compression ratio and expansion ratio are not necessarily the same: they depend on intake-valve-closing and exhaust-valve-opening crank angles. Also, with modern engines, variable valve timing is a common feature. While the average intake valve closing is about

50 degrees after BC, and exhaust valve opening 55 degrees before BC, variations of some ±20 degrees about these mean values now occur. The terms *effective* compression and expansion ratio are often used to better define the actual engine's compression and expansion processes. These are defined by the cylinder volumes at intake valve closing and exhaust valve-opening crank angles, respectively.

Results from the ideal-gas constant-volume cycle and the fuel-air cycle for various compression ratios, specific heat ratios, and equivalence ratios are shown in Fig. 15.22. These results show that *indicated* fuel conversion efficiencies are expected to increase, though at a decreasing rate, with increasing compression ratio. However, analysis of extensive SI engine data shows a different behavior. There is a significant spread in the efficiency values across different engine designs and operating conditions and, as would be expected, the engine indicated efficiencies are lower than the ideal-cycle values by some 15%. In these several experiment studies, where compression ratio was changed in a given engine by decreasing the clearance height of the pistons (thus, at constant engine displaced volume), while efficiency increases with increasing compression ratio, it does so at a decreasing rate and reaches a maximum at a (geometric) compression ratio of about 15:1.

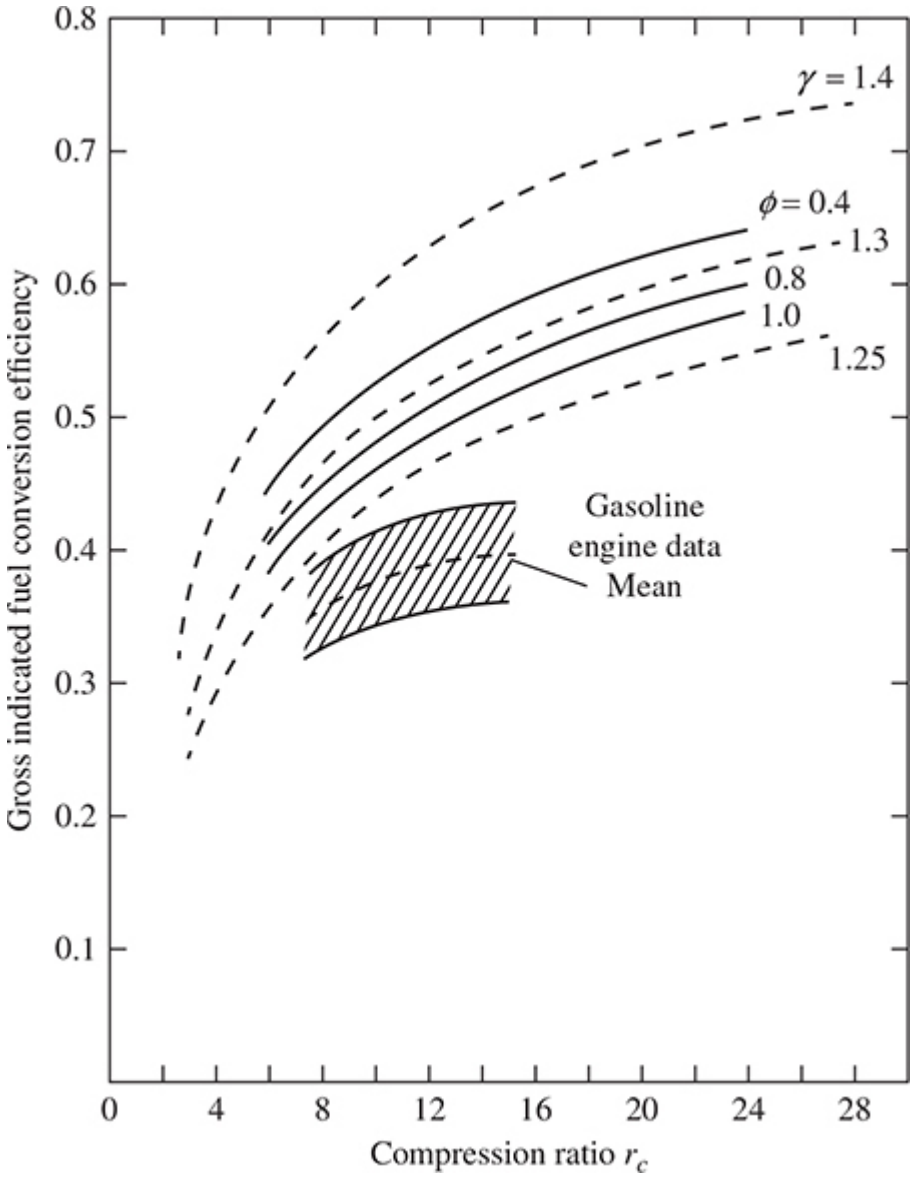


Figure 15.22 Gross indicated fuel conversion efficiency as a function of compression ratio for the ideal gas constant-volume cycle with different values of g (dashed lines), the fuel-air cycle (see Sec. 5.5.1) for different values of f (the fuel-air equivalence ratio), and the available modern gasoline SI engine data.[25]

That engine efficiency data are lower than the ideal cycle values, and at higher compression/expansion ratios have a different trend, should not be surprising. As this text has discussed, several "losses" contribute:

- Changing gas properties (the ratio of specific heats, γ) decreases with increasing temperature.
- The combustion process does not release all the fuel's chemical energy.
- Combustion, though fast, takes up a finite portion of the compression and expansion strokes.
- Significant heat loss from the burned gas reduces the gas pressure.
- The blowdown process when the exhaust valve opens before BC loses a fraction of the full expansion stroke work.

Note that the above losses occur over the compression and expansion strokes, and thus impact the gross indicated performance and efficiency of the engine. The net indicated output $\int pdV$ over the full four strokes of the cycle) additionally includes the pumping work (losses in a naturally-aspirated gasoline engine; a gain in a turbocharged gasoline engine operating with intake pressures above 1 bar). The brake performance and efficiency are additionally impacted by the magnitude of engine friction (its mechanical and accessory components: pumping work is included in the net indicated values).

Studies that have examined the effect of compression ratio on spark-ignition engine performance and efficiency over a wide range of compression ratios are limited. Figure 15.22 shows results from a recent broadly based analysis.[25]

An important question is the effect of compression ratio variations for different size engine cylinders. Improving engine performance and turbocharging/downsizing are prompting reductions in engine size. Figure 15.23 shows brake fuel-conversion efficiency data from engines of different cylinder volume. Both the compression ratio for maximum efficiency, and the value of maximum efficiency, depend on cylinder size. The wide-open-throttle and road-load data (top two curves [26]) confirm that the increase in efficiency with an increase in the compression ratio at part load apparently depends on the details of engine operation to a significant degree also. For the important compression ratio range of 9 to 11, this data set indicates that the relative efficiency improvement is between 1 and 3% per unit of compression ratio increase, depending on cylinder size and operating conditions. Naturally-aspirated gasoline engine cylinder sizes are about 500

cm ^{3} per cylinder (in 2015). Smaller turbocharged engines are becoming common (e.g., 1-liter three-cylinder engines with 333-cm ^{3}-size cylinders).

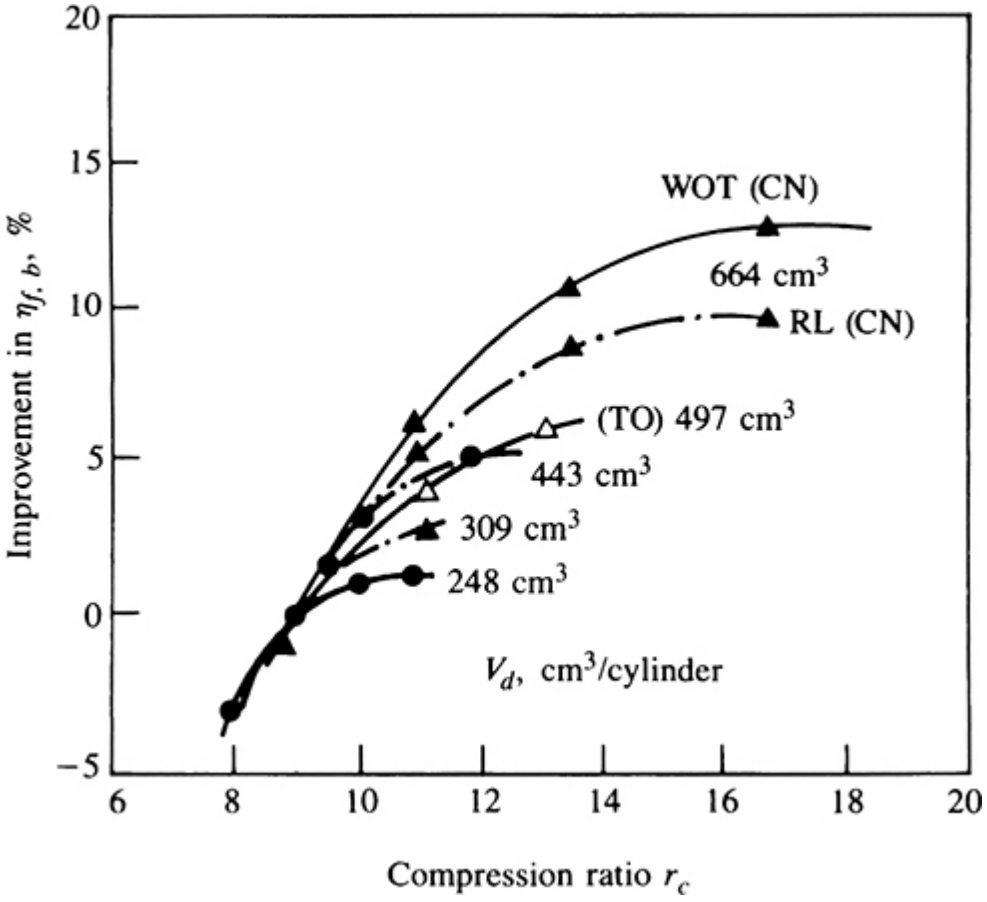


Figure 15.23 Relative brake fuel conversion efficiency improvement with increasing compression ratio of spark-ignition engines of different displaced volume per cylinder: top curve at wide-open throttle, CN;[26] road load (RL), CN;[26] other curves at part throttle, MTJ;[27] TO.[28]

The exhaust temperature decreases as compression ratio and efficiency increase until the compression ratio corresponding to maximum efficiency is reached. It has also been shown that heat losses to the combustion chamber walls, as a fraction of the fuel's chemical energy, also decrease as the compression ratio and efficiency both increase. [26]

The effect of compression ratio changes on NO emissions is small. Some studies show a modest increase in specific NO emissions as the compression ratio increases at constant load and speed; other studies show a slight

decrease. Increasing the compression ratio increases exhaust hydrocarbon emissions. Several trends could contribute: increased importance of crevice volumes at high r_c; lower gas temperatures during the latter part of the expansion stroke, thus producing less HC oxidation in the cylinder; decreasing residual gas fraction, thus increasing the fraction of in-cylinder HC exhausted; lower exhaust temperatures, hence less oxidation in the exhaust system.

The overall impact of increasing compression ratio on engine efficiency can be summarized as follows. Compression ratio values are currently about 10:1 for naturally-aspirated PFI gasoline engines, 9:1 for turbo PFI engines, and 10.5 for turbo DI engines. The higher pressures in turbocharged engines at high load make knock onset (autoignition of the fuel-air mixture ahead of the propagating flame) more likely. The cooling of the in-cylinder charge through evaporation of the direct-injected liquid gasoline spray delays knock onset and allows use of higher compression ratios. These compression ratio values have been steadily rising over time, [3] a trend that is expected to continue and is enabled by increasing use of variable intake valve closing, more uniform cylinder head cooling, and more sophisticated spark retard strategies based on better knock sensing and control.

Figure 15.24 shows the relative change in indicated and brake efficiency (expressed as a percentage) as a function of compression ratio, normalized to $r_c = 10$. The difference between the two curves (each an average of several data bases) is modest. (The changes in brake efficiency are slightly higher since the normalizing number—the brake efficiency at 10:1 compression ratio—is slightly lower.) The differences between these efficiencies at low (1 to 3 bar bmep), mid (4.5 to 7 bar bmep), and high (7.5 to 9.5 bar bmep) are also small. Overall, we see that at *constant engine displaced volume*, raising r_c from 10 to 11 gives a 2% efficiency improvement, from 12 to 13 a 1.3% improvement, and above 15:1 essentially no increase. Thus, future compression ratio increases offer decreasing benefits, with an upper limit at about 15:1.

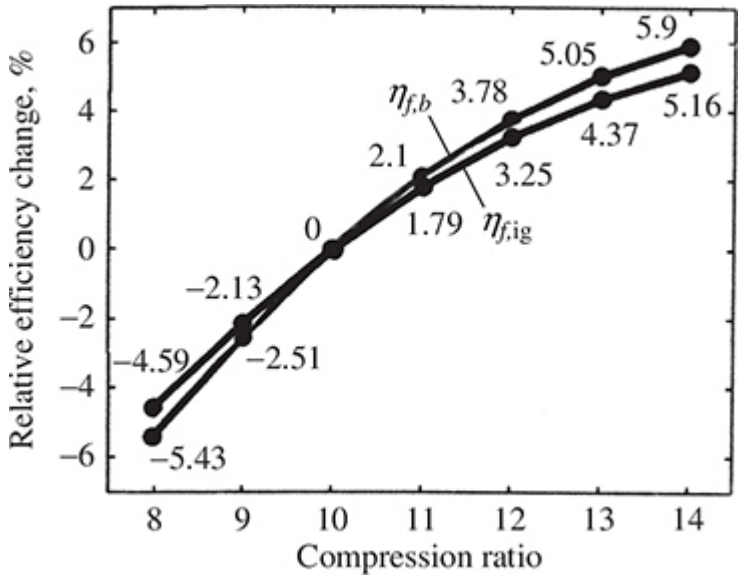


Figure 15.24 Average relative gross indicated and brake fuel conversion efficiency changes versus compression ratio, gasoline SI engines. Relative efficiency changes from unit compression ratio changes are noted (as percentages): r_c normalized at 10:1. [25]

However, as compression ratio is increased, so is the engine's maximum torque. Thus the engine's displacement can be reduced so the torque remains constant (as the higher-compression ratio engine does the same task). With engine downsizing, the benefits of raising compression ratio are significantly increased as shown in Fig. 15.25. Without engine downsizing, the increase in brake efficiency as r_c increases from 9.8 to 13.4 is close to 6% (close to the value, from a different study, in Fig. 15.24). With adjustments in engine size made to maintain constant maximum torque, the efficiency increase is higher. This constant torque comparison, with appropriate engine downsizing, is the comparison of greatest interest. Effectively, the benefits presented in Fig. 15.24 at constant engine displacement, for practical applications where downsizing would occur, understate the real potential. [29] The additional gain depends on whether the engine is naturally-aspirated or turbocharged. It also depends on the rate of change in efficiency with load at the bmep level at which this downsizing effect is being evaluated. At lighter loads, the multiplying factor (shown in Fig. 15.25) is higher: in the 1.3 to 1.6 range for naturally-aspirated gasoline engines. At higher loads, this enhancement factor is lower: it is about 1.1 for turbocharged gasoline engines.

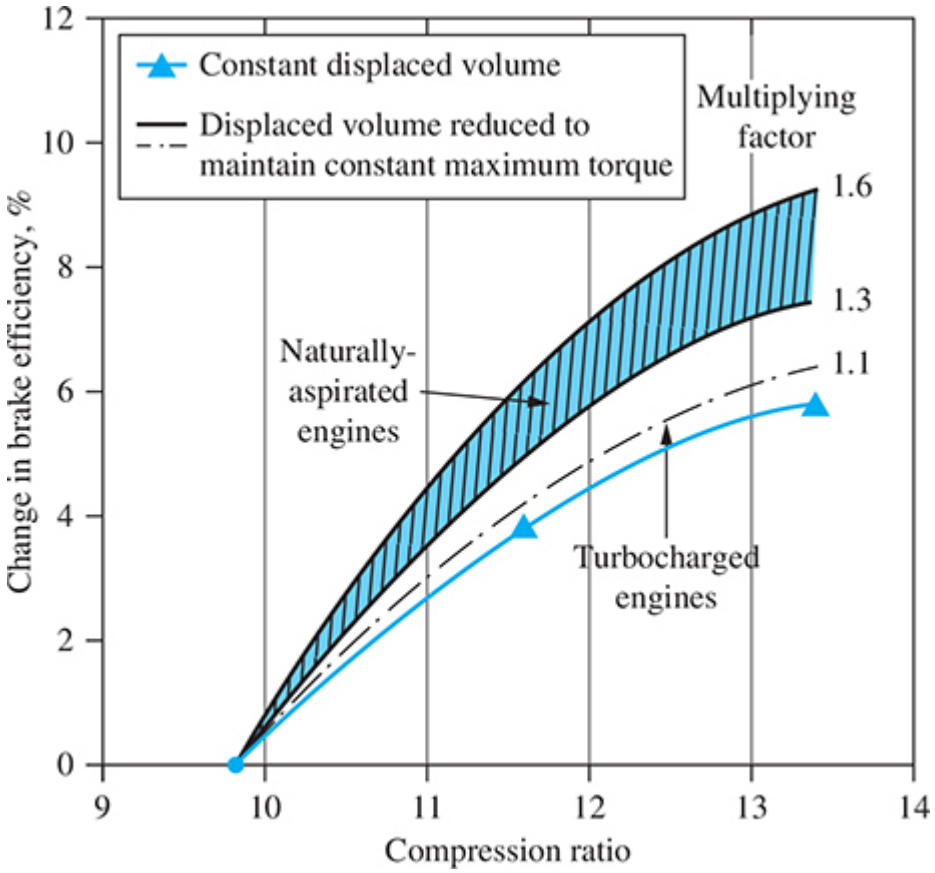


Figure 15.25 Estimated increase in light- to mid-load brake fuel-conversion efficiency of gasoline engines, with increasing compression ratio: bottom curve, at constant displaced volume; upper curves, with engine downsizing to maintain constant maximum engine torque. Efficiency benefit multiplying factor for naturally-aspirated and turbocharged engines shown on right. (Multiplying factor is change in efficiency with downsizing relative to change in efficiency at constant displaced volume.) [29, 30]

15.4 SI ENGINE COMBUSTION SYSTEM DESIGN

15.4.1 Objectives and Options

SI engine combustion-chamber design continues to evolve as requirements and our knowledge base changes. There are several options for cylinder head

and piston crown shape, spark plug location, size and number of valves, and intake port design. Involved are questions such as chamber compactness, surface/volume ratio, flame travel length, how and where the fuel is injected, and use of swirl, tumble, and squish types of mixture motion. Figure 15.26 shows examples of several common combustion chamber shapes. A consensus has developed which favors faster-burning combustion-chamber designs. A chamber design where the fuel burning process takes place faster (i.e., occupies a shorter crank-angle interval at a given engine speed) produces a more robust and repeatable combustion process that provides emission control and efficiency gains simultaneously. A faster-burning chamber with its shorter burn time permits operation with substantially higher amounts of EGR, or with leaner mixtures, within the normal constraints of engine smoothness and response. Thus greater emissions control within the engine can be achieved and, at part load at this higher level of dilution, a faster-burning chamber shows an improvement in fuel consumption due to the reduced pumping work, reduced heat transfer (due to lower burned gas temperatures), and higher g due to reduced temperatures and amount of dissociation in the burned gases.

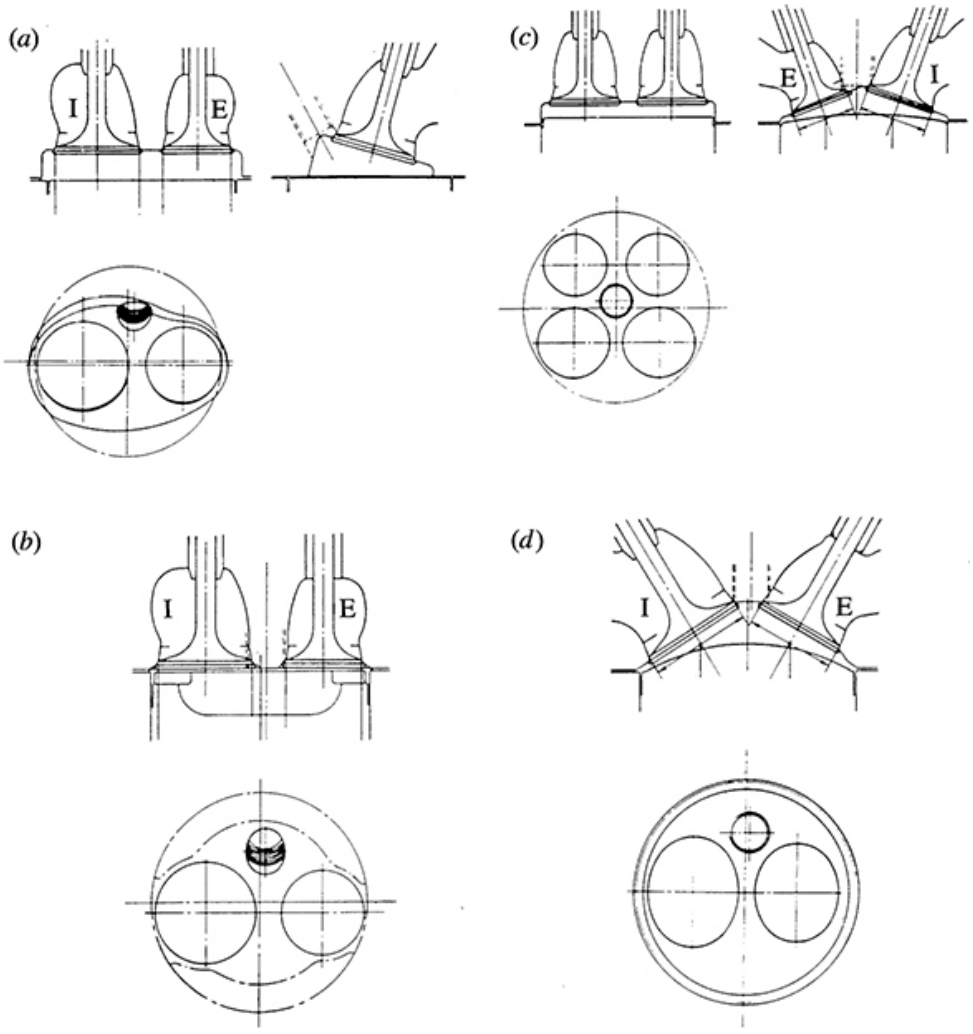


Figure 15.26 Examples of different spark-ignition engine combustion chamber shapes: (*a*) bathtub and wedge; (*b*) bowl-in-piston; (*c*) four-valve pent roof; and (*d*) hemispherical.[31]

The major combustion chamber design objectives that relate to engine performance and emissions are:

- A fast combustion process, with low cycle-by-cycle variability, over the full engine operating range.
- High volumetric efficiency at wide-open throttle.
- Minimum heat loss to the combustion chamber walls.
- A low fuel-octane requirement.
- Low engine-out emission levels (especially when the engine is cold).

Many methods for producing "fast burn" have been proposed. These include ways of making the combustion chamber shape more compact, moving the spark plug to a more central location within the chamber, using two plugs, and increasing in-cylinder gas motion by creating a tumbling or swirling flow in the cylinder during the induction process.

A faster combustion process relative to slower burn-rate older engines does result in a direct engine efficiency gain, other factors being equal. The magnitude of this direct gain is relatively modest. Experimental studies of the effect of an increase in burn rate from slow to moderate on engine efficiency is significant: from moderate burn to fast burn at constant engine load, speed, and mixture composition, the burn rate effect is several percent but not more. [32] (From fast to very fast burn achieves a smaller percentage still.) Computer simulations of the engine operating cycle confirm these experimental observations: while a decrease in total burn duration from 100 to 60° does result in a 4% decrease in bsfc, a decrease in burn duration from 60 to 20° gives only a further 1.5% bsfc decrease. [13]

Of greater importance is the fact that the faster burn process is more robust and results in the engine being able to operate satisfactorily with much more EGR, or much leaner, or with greater spark retard, without a large deterioration in combustion quality. Faster burning chamber designs exhibit much less cycle-by-cycle variability. This ability to operate with greater dilution at part load while maintaining a short burn duration and low cycle-by-cycle combustion variability, permits greater control of NO x within the engine with some 15% EGR without any substantial increase in HC emissions (see Fig. 11.31), or permits much leaner operation. In both cases the efficiency gain relative to moderate burn-rate engines, which must operate with less dilution, is sizeable. [33]

High volumetric efficiency is required to obtain the highest possible power density. The size and shape of the cylinder head affects the sizes of the valves that can be incorporated into the design. Effective valve open area, which depends on number of valves, valve diameter and lift, directly affects volumetric efficiency. Tumble and swirl are components in chamber designs that speed up the burning process and achieve greater combustion stability. Induction-generated tumble (see Sec. 8.4) is a particularly effective in-cylinder flow for increasing the burn rate. It results in higher turbulence inside the chamber during combustion thus increasing the rate of flame

development and propagation. Generating tumble during the intake process does decrease volumetric efficiency.

Heat transfer to the combustion chamber walls has a significant impact on engine efficiency. It is affected by cylinder head and piston crown surface area, by the magnitude of in-cylinder gas velocities during combustion and expansion, by the gas temperatures and the wall temperatures. The heat-transfer implications of a combustion chamber should be included in the design process.

Knock effectively limits the maximum compression ratio that can be used in any combustion chamber; it therefore has a direct impact on efficiency. Knock is affected by most of the combustion chamber features discussed above. It is the hardest of all the constraints to incorporate into the design process because of its complexity.

Knowledge of the fundamentals of spark-ignition engine combustion, in-cylinder gas motion, and heat transfer has developed to the point where a rational procedure for evaluating these factors for optimum combustion chamber development and design can be defined. The next two sections develop this logic.

15.4.2 Factors That Control Combustion

Our understanding of the structure of the spark-ignition engine flame as it develops and propagates across the combustion chamber (see Secs. 9.3 and 9.4) allows us to relate the physical and chemical factors that control this process to the relevant combustion system design and operating parameters. The following factors affect the flame development and propagation processes:

- *Geometry*. Combustion chamber shape and spark plug location.
- *Flow field characteristics*. Mean velocity, turbulence intensity, and characteristic turbulence length scale in the unburned mixture during the combustion process.
- *Unburned mixture composition and state*. Fuel, relative air/fuel ratio, burned gas fraction, mixture pressure and temperature.

Geometry primarily affects combustion through the flame-front surface area. It has a lesser effect on combustion development through its influence on in-cylinder motion. Geometric calculations (see Sec. 14.4.2), based on

the assumption that the front surface of the flame can be modeled as a portion of a sphere centered at the spark plug, provide data on flame front area and the volume behind the flame front surface (the enflamed volume), contained within the combustion chamber, as a function of flame radius and piston position.

Flame area varies significantly from one chamber shape to another for a given enflamed volume. In the example shown in Fig. 14.7, the bowl-in-piston chamber gives flame surface areas 30 to 45% larger than those for the disc chamber under equivalent conditions, around top center. Hemispherical and open or clamshell chambers showed gains of about 30% relative to the equivalent disc configuration. For a given chamber shape, flame area depends even more significantly on plug location. Figure 14.7 also shows that shifting the plug from a side to a center location for the bowl-in-piston chamber increased the peak flame area by 150%. For hemispherical and open chambers, the increases for a similar shift in plug location were 75 and 90%, respectively. [34]

Maps of flame area as a function of radius at different crank angle locations indicate the following pattern. For chamber geometries with side ignition, as flame radius increases the flame area first rises slowly, then remains approximately constant, and then decreases slowly to zero. In contrast, chambers with central ignition show, as flame radius increases, a rise in flame area to a peak during the major part of the flame travel followed by a rapid decrease as the flame encounters the chamber walls. Moving the plug location toward the center of the chamber produces a larger increase in flame front area than does making the chamber shape more compact (though this has a positive impact too).

The effect of chamber geometry on burn rate has been examined using thermodynamic-based engine cycle simulations with various types of combustion model (e.g., the type developed by Keck and coworkers; see Sec. 14.4.2). Figure 15.27 shows results from one such study.[34] The combustion characteristics of ten different chamber geometries were compared at fixed part-load engine operating conditions. The flame development and propagation phases were separated into 0 to 10% and 10 to 90% mass fraction burned times. These were then normalized by the equivalent burn times of the slowest burning chamber—the disc with side ignition. Chamber geometry has the greatest impact on the 10 to 90% burn time; its effect on 0 to 10% time is significant but substantially smaller. Total burn times can be

reduced by between 20 and 30% by optimizing spark plug location—comparing worst to best location for each chamber shape. Comparing worst and best chamber shapes, total burn time with fixed plug location can be reduced by about 10%.

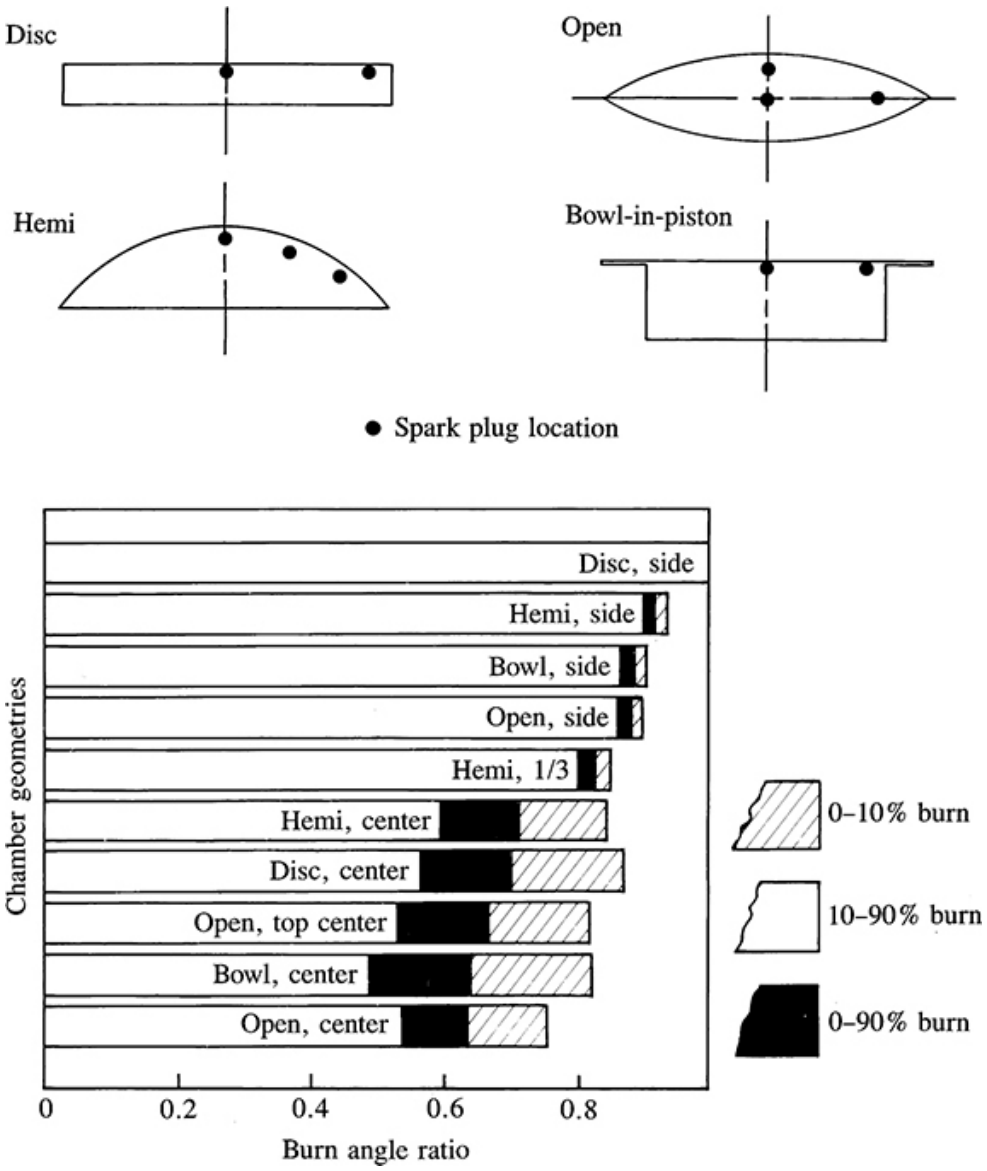


Figure 15.27 Comparison of burn angles (0 to 10% burned, 10 to 90% burned, 0 to 90% burned (see Fig. 9.15) for ten different spark-ignition engine combustion chamber geometries and spark plug locations. Burn angles are normalized by the burn duration angle for the slowest burning chamber: disc with side plug. [34]

Increased turbulence in the unburned mixture at the time of combustion increases the burning rate. Turbulence is usually increased by generating swirl during the induction process (see Sec. 8.3.2 and below). Cycle simulation studies [34] indicate that both the duration of the early stage of the burning process, and of the main stage, decrease when the turbulence levels during combustion are increased. The faster combustion process comes primarily from the higher turbulence intensity increasing the rate of flame propagation; however, the decreased characteristic turbulence scale that accompanies the increased turbulence is also significant since it results in a shorter characteristic burning time [see Eq. (14.38) , and the accompanying text]. It is important to note that the fuel conversion efficiency of higher-turbulence chambers at the same operating conditions are not necessarily higher than for normal chambers, despite the faster burn rates, due to the higher heat transfer rates that accompanies the higher in-cylinder velocities. For example, predictions based on the combustion model defined by Eqs. (14.33) to (14.35), where the characteristic mixture convection speed u_T was increased by a factor of two, showed that the 0 to 10% and 10 to 90% burn durations decreased by about one-third. However, the indicated fuel conversion efficiency decreased by about 6% due to the predicted 15% increase in heat transfer per cycle. [34]

Mixture composition and state affect the burn rate through the dependence of laminar flame speed on temperature, pressure, fuel/air equivalence ratio, and burned gas fraction (residual gas and EGR): see Sec. 9.3.3 and Eqs. (14.33) to (14.35). Table 15.2 compares the burn durations for a stoichiometric mixture, a lean mixture with $\phi = 0.8$, and a stoichiometric mixture with 20% EGR. The values of the laminar flame speed at the time of spark are also given (conditions at spark as well as composition are different in each case). The longer burn durations of the more dilute mixtures are clear. Note that EGR as a diluent has a much more deleterious effect on combustion than does air at these approximately equal levels of dilution.

TABLE 15.2 Effect of excess air and recycled exhaust on burn duration

λ	EGR, %	θ_s, deg	Burn durations, deg		S_L at θ_s, cm/s
			0–10%	10–90%	
1.0	0	340	22	17	75
1.25	0	336	26	21	52
1.0	20	324	31	28	23

400 cm^3 per cylinder displaced volume, 80 mm bore, compression ratio 8.5, disc chamber, center plug location. 1500 rev/min, stoichiometric operation, θ_s = spark timing (MBT), inlet pressure 0.5 atm, inlet temperature 350 K, S_L = laminar flame speed.[35]

All the above-described factors—flame geometry, fluid motion, and mixture composition—can vary cycle-by-cycle, and therefore contribute to combustion variability (see Sec. 9.4). Cyclic differences in gas motion in the vicinity of the spark plug result in differences in motion of the flame during its early stages of development. Differences in turbulence result in differences in the rates at which the initially smooth surface of the flame kernel becomes wrinkled and convoluted by the flow. Also, different initial flame center motions change the geometrical interaction of the flame front with the combustion chamber walls later in the flame propagation process. Differences in the amount of fuel, air, and EGR which enter each cylinder cycle-by-cycle, the nonuniformity in composition of the entering charge, and any incomplete mixing of that entering charge with the residual gases in the cylinder, also contribute to combustion variability. These composition nonuniformities lead to differences in the early stages of flame development. The variations in the amounts of fuel, air, and EGR that enter each cylinder cycle-by-cycle and in the uniformity of that mixture are factors over which the engine designer has some control.

A fast combustion process reduces cyclic combustion variability through the factors summarized in Table 15.3. With a faster burn, optimum spark timing is closer to top-center: mixture temperature and pressure at the time of spark are higher, so the laminar flame speed at the start of combustion is greater. This, combined with the higher turbulence of fast-burn combustion systems, results in faster flame kernel development. More rapid initial flame growth results in less variation in flame center motion during the critical flame-development phase. The resulting geometric variations in the flame front/chamber wall interaction are therefore reduced; this decreases the variations in burn rate that result from these geometric variations. Also, the

faster burning process ends earlier in the expansion stroke. Thus the problem of occasional slow burning cycles, partial burning cycles, and eventually misfire, which occurs with dilute mixtures under normal burning conditions due to quenching of the combustion process as gas temperatures fall during expansion, is largely avoided (see Sec. 9.4.3).

TABLE 15.3 Causes of SI engine cycle-to-cycle combustion variations

Phases of combustion
1. Early flame development
2. Flame propagation
3. Late stage of burning
Factors affecting SI engine cycle-to-cycle variations
(a) Spark energy deposition in gas (phase 1)
(b) Flame kernel motion (phase 1)
(c) Heat losses from kernel to spark plug (phase 1)
(d) Local turbulence characteristics near plug (phase 1)
(e) Local mixture composition near plug (phase 1)
(f) Overall charge components – air, fuel, residual, EGR (phases 2, 3)
(g) Average turbulence in the combustion chamber (phases 2, 3)
(h) Large-scale features of the in-cylinder flow (phase 3)
(i) Flame geometry interaction with the combustion chamber walls (phase 3)

15.4.3 Factors That Control Performance

Volumetric Efficiency

Combustion chamber shape affects volumetric efficiency through its constraints on maximum valve size and through the degree of tumble or swirl (if any) that the chamber and port designs produce to achieve the desired combustion characteristics. To obtain maximum performance and to reduce pumping losses, two intake valves (four-valve, three-valve cylinder heads), with the size of the valve heads as large as is practical. The valve sizes that can be accommodated depend on cylinder head layout. Table 6.1 lists the typical maximum valve sizes that can be accommodated into several common chamber shapes (see Fig. 15.26). The approximate mean piston speed at maximum power is a measure of the maximum airflow that each engine

design can pump. Note that cylinder head configurations with inclined valve stems permit greater maximum airflow because, for a given bore, the valve and port open areas are larger. The four-valve pent-roof design, through its four valves and inclined valve stems, is the best of those listed since it accommodates the largest valve and port areas.

Table 15.4 illustrates the impact of the number of valves per cylinder (vpc), and fixed and variable valve timing, on the key engine performance parameters with results from a study of some 200 gasoline engines in the 2008 model year North American light-duty vehicles. The various configurations are listed in order of increasing airflow. With naturally-aspirated engines (the first four lines in Table 15.4), maximum bmep, $P_{b,\max}/V_d$ and $\overline{S}_p$ at $P_{b,\max}$ all increase as the cylinder-head intake flow resistances are decreased. With boosting (supercharging and turbocharging) output "jumps" significantly, and with turbocharging the maximum mean piston speed is reduced to limit the increasingly negative effect of engine friction at very high engine speed: additional boost can be used to offset this *downspeeding*.

TABLE 15.4 Normalized performance: 2008 North American engines [3]

Valve configuration, etc.	Number of engines, 203 total	Max. bmep (max torque), kPa	Specific power ($P_{b,\max}/V_d$), kW/liter	bmep at max power, kPa	bmep at max power/max bmep	Max $\overline{S}_p$ at $P_{b,\max}$, m/s
12 vpc	42	1082	42.9	970	0.90	15.8
3 vpc	7	1162	45.7	1018	0.88	17.2
4 vpc, fixed	47	1177	51.6	1039	0.88	17.3
4 vpc, variable	81	1230	56.9	1109	0.9	18.1
4 vpc, superch.	4	1734	75.6	1487	0.85	18.6
4 vpc, turboch.	22	1817	80.2	1688	0.93	16.9

Tumble and Swirl

Moderate levels of *tumble* (or *swirl*) are used in gasoline SI engines to achieve fast-enough burn rates through realizing higher turbulence levels in the in-cylinder fluid (see Figs. 8.4 and 8.19). To achieve a tumbling in-cylinder flow, the cylinder head is designed with the intake port axes lowered so as to bring the airflow entering the cylinder largely past the top of the intake valve head and across the cylinder as shown in Fig. 8.19. As the piston moves down the cylinder, this flow moves downwards, then back across the cylinder and sets up the rotating flow illustrated. As the piston

moves up after BC, this tumbling flow is "squashed" and breaks up during the compression stroke into first, smaller eddies, and then turbulence, in time to speed up the flame propagation process.

Typical impacts of tumble and swirl on spark-ignition engine's combustion characteristics are shown in Figure 15.28. Increasing the tumble ratio speeds up the burning process significantly: both the flame development angle ($\Delta\theta_{\text{spark–5\%}}$) and the burn duration ($\Delta\theta_{\text{5–90\%}}$) decrease. The port-deactivation swirl-generating system is intermediate in its impact between the medium and high tumble cases. This is in line with the broader assessment that moderate tumble is usually the appropriate choice to achieve stable SI engine combustion: it provides a fast-enough burn rate to realize most of the burn-rate efficiency benefits, while maintaining a relatively high flow coefficient at high speeds and loads. Typically, flame development and burning angles below about 20 and 25 degrees, respectively, are needed at these light-load low-speed conditions.

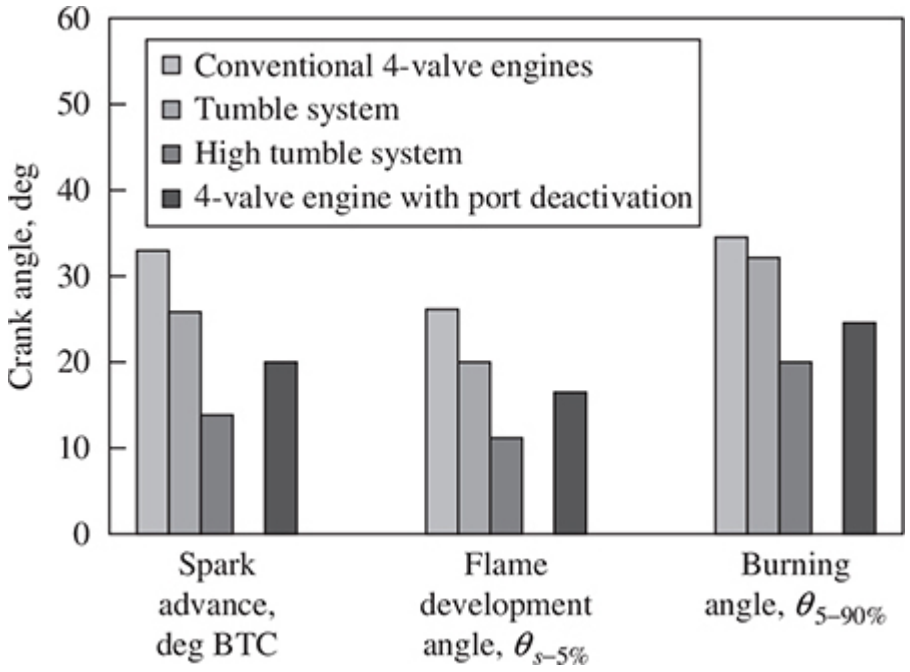


Figure 15.28 Influence of different types and levels of in-cylinder charge motion on SI engine flame development and propagation. Stoichiometric operation, at 3 bar bmep and 2000 rev/min. Conventional, four-valve pent roof combustion system (see Fig. 15.26): tumble systems ranged from tumble ratios of 3 to 6; port deactivation (see Fig. 8.28) generated a swirl ratio of about 2. [36]

As an alternative to tumble, *swirl* can be generated during the intake process through suitable port, valve, and head design. It requires either that the flow through the intake valve be directed tangentially into the cylinder so that gas flows through one side of the valve opening preferentially (e.g., through the use of masks to restrict flow at the mask location, or through the use of a tangentially-directed port, or a flow deflector in the port just upstream of the valve), or requires the use of a helical intake port that generates rotation of the flow about the valve stem before the air enters the cylinder. In either case the flow enters the cylinder with higher velocity than it would have in the absence of swirl: hence the pressure drop across the port and valve is increased, and the maximum airflow through the cylinder is reduced. Well-designed helical swirl-generating ports (see Sec. 8.3.2) appear to be the best way to create swirl. In gasoline engines geometric and production constraints often prevent the incorporation of such ports into the cylinder head design, and other swirl-generating methods must be used. The engine maximum-power penalty associated with generating significant swirl is of order 5%.

Since tumble or swirl is required at part-throttle operation when enhancement of the burn rate is most critical and is not usually needed at full throttle when the flow restriction penalty is most significant, induction systems with a modified passage for the part-throttle airflow are an attractive option. However, the gains in volumetric efficiency are offset by a higher cost due to the additional complexity in port and manifold of the double passage and the individual throttle valves required in each port for flow control.

In smaller-size DI diesel engines, swirl is intensified during compression with bowl-in-piston combustion chambers by decreasing the moment of inertia of the in-cylinder charge as the piston moves toward top-center, thereby increasing its angular velocity (see Sec. 8.3.3). An advantage here is that the swirl level generated during induction is less than would be required without the compression-produced radially inward motion of the charge. The impact of piston-motion generated radially-inward squish motion (see Sec. 8.5) on in-cylinder turbulence, and hence combustion, is complex. Chambers with significant squish are more compact; for this reason alone, they would be faster burning.

Heat Transfer

The convective heat transfer to the combustion chamber walls is described by equations of the form of Eq. (12.2) . The heat-transfer coefficient is usually correlated by expressions of the form of Eq. (12.3) , which relates the Nusselt, Reynolds, and Prandtl numbers (see Sec. 12.4). Thus combustion chamber surface area, and especially the surface area in contact with the burned gases, is an important parameter. The in-cylinder gas velocity is also important: it influences the heat-transfer rate through the Reynolds number. Various characteristic velocities have been used in the Reynolds number to scale heat transfer: mean piston speed, mean in-cylinder gas velocity, turbulence intensity, either individually or in combination. Both of these variables, area and velocity, are affected by combustion chamber design.

Studies of engine performance using thermodynamic-based simulations of the engine's operating cycle (see Sec. 14.4) provide data that indicate the importance of changes in heat transfer. At part-throttle operating conditions, such simulation calculations show that a 10% change in combustion chamber heat losses results in a change of between 2 and 5% in brake-specific fuel consumption; an average fuel consumption (relative) change of about one-third the magnitude of the relative heat-transfer change (and of opposite sign) is an appropriate rule of thumb. [34,37] At wide-open throttle, the effect on mean effective pressure is comparable: a 10% change in heat transfer results in about a 3% change in bmep.

This impact of heat transfer on engine efficiency and performance underlines the importance of combustion chamber details that affect heat transfer. For the chamber shapes shown in Fig. 15.27, the total heat losses as a fraction of the fuel's energy, at fixed engine speed and intake conditions, were also calculated. Both chamber shape and spark plug location affect heat losses since together these govern the surface area of the hot burned gases in contact with the walls. The open and hemispherical chambers had least heat transfer. Geometries such as the bowl-in-piston which obviously have a higher surface area had about 10% higher heat transfer. The effect of shifting the plug from a side location to a center location depended on chamber shape. Open and bowl-in-piston chambers showed little change; the hemispherical chamber showed a 4% reduction. Given an overall chamber-shape choice, the details of the actual design are also important; it is easy to add substantial surface area with piston cutouts, plug bosses, cylinder head

masking, and squish regions which can deteriorate chamber performance to a measurable degree.

Higher in-cylinder velocities affect heat-transfer rates through the Reynolds number term in the heat-transfer coefficient correlation. Tumble, swirl, and squish-generated flows increase in-cylinder gas velocities and will, therefore, increase heat-transfer rates.

15.4.4 Chamber Octane Requirement

The abnormal combustion phenomenon called knock limits a spark-ignition engine's compression ratio, and hence its performance and efficiency. The more fundamental aspects of knock are reviewed in Sec. 9.6.2 . Knock occurs when the fuel-air end-gas mixture ahead of the flame ignites spontaneously prior to its being burned up by the normal flame-propagation process. The tendency to knock depends on engine design and operating variables which influence end-gas temperature, pressure, and time spent at high values of these two properties, before flame arrival.

The presence or absence of knock in an engine also depends on the antiknock quality of the fuel, which is defined by the fuel's *octane number* (see Sec. 9.6.3). That parameter determines whether or not a fuel will knock in a given engine under given operating conditions: the higher the octane number, the higher the resistance to autoignition and knock. The *octane number requirement* of an engine is defined as the minimum fuel octane number that will resist knock throughout its speed and load range. The following factors affect an engine's octane requirement: (1) composition of the fuel; (2) chamber geometry and size; (3) charge motion; (4) spark-advance; (5) intake air, intake port and valve, and water jacket temperatures; (6) air/fuel ratio; and (7) ambient air conditions—pressure, temperature, and relative humidity—during the requirement determination.

The interaction between fuel octane rating and engine operating variables at higher loads, illustrated in Fig. 15.29, shows the relation between spark advance, torque, and speed in an engine operating at wide-open throttle. The dashed lines, determined with a fuel of sufficiently high octane rating to avoid knock, show MBT timing as a function of speed, along with the spark-advance limits for constant specified percentage torque reductions. The upper solid line traces the spark advance for borderline knock with a particular commercial gasoline. To avoid knock onset with this fuel, the

spark advance was set to lose 1% of engine torque at 800 rev/min, with the torque loss diminishing to zero at 1200 rev/min. Above that speed this particular fuel allows operation at MBT timing without knocking. The lower solid curve represents a typical spark-advance schedule at WOT. It lies below the borderline knock advance (and results in an additional torque loss) for the following reasons. One is that different commercial gasolines with the same research octane number can respond differently to variations in engine operating conditions. Calibrating the engine (i.e., specifying the schedules for spark advance, *A*/ *F*, and EGR) must be done with a sufficient margin to avoid objectionable knock with the normal range of commercial gasolines over the full operating conditions of the engine. A second reason is engine-to-engine production variability despite the close dimensional tolerances of modern production engineering. For example, the effective compression pressure in each cylinder of a multi-cylinder engine is not identical, due to geometric and ring-pack differences. The cylinder with the highest compression pressure is most knock-prone. Allowing for corresponding effects of cylinder-to-cylinder variations in *A*/ *F*, EGR rates, and spark timing, it is obvious that for a given operating condition in a multi-cylinder engine, one cylinder is more likely to knock than the others. It is that cylinder which limits the spark advance.[j] A third reason for the discrepancy between actual spark-advance calibration and the knock limit for a given engine and fuel is the octane requirement increase associated with the buildup of deposits on the cylinder head and piston over extended mileage (see Sec. 9.6.3).

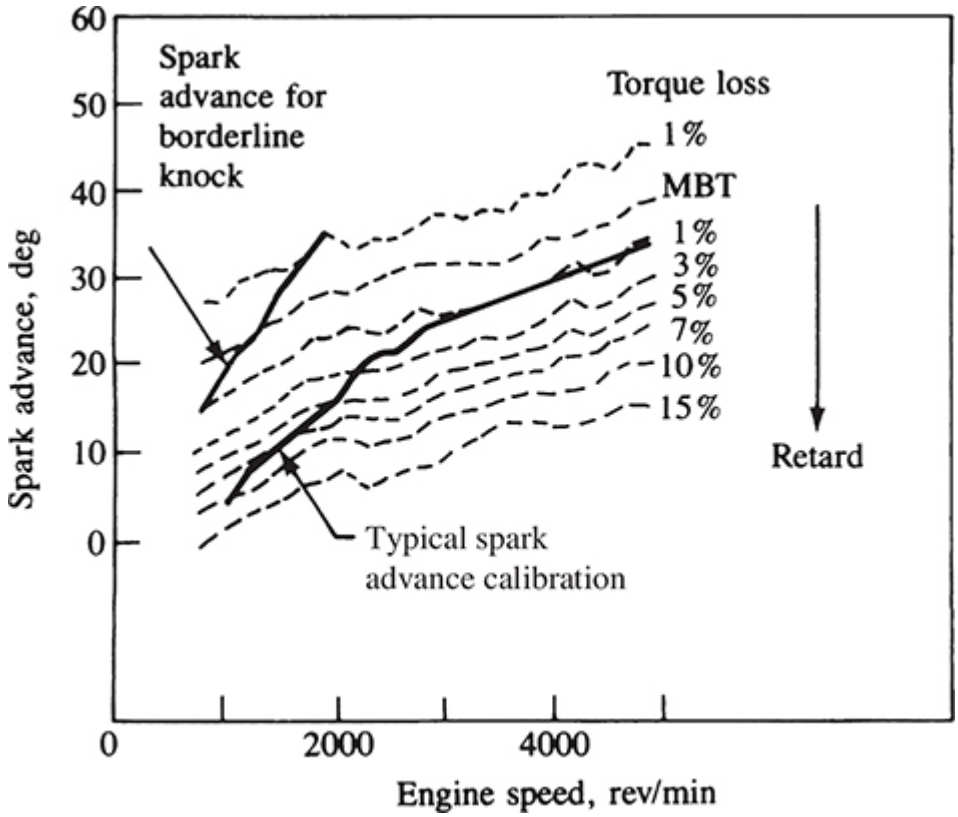


Figure 15.29 Relation between spark advance, speed, and torque loss for spark-ignition engine at wide-open throttle showing knock limit for a specific gasoline and typical spark-advance schedule that avoids knock problems.[38]

In the example shown, it was the problem of knock at lower engine speeds which required the calibrated spark advance to be retarded. Whether low-, medium-, or high-speed knock is the limiting factor in a particular engine depends on the sensitivity of the fuel (see Sec. 9.6.3), on engine design details, and on the engine's spark-advance requirements for MBT. The knock-limited spark advance determined from road octane rating tests can vary with engine speed and fuel sensitivity. While high sensitivity fuels usually show higher in-use octane index, at higher loads and speeds, in some engines, lower sensitivity fuels may resist knock better.

As Fig. 15.29 illustrates, spark retard from MBT spark timing is an effective way to delay the onset of knock, because it can immediately reduce the end-gas peak pressures and temperatures. Spark retard at low to mid engine speeds is built into the engine's calibration to prevent knock with the engine's specified fuel octane requirement. In turbocharged engines, at mid to

high speed, this strategy is even more effective because the torque loss with a given spark retard from MBT is reduced (see Fig. 15.10). However, spark retard also results in an increase in engine exhaust gas temperature, which must not be allowed to exceed the maximum turbine intake gas temperature constraint.

As is well known, the octane requirement increases with increasing compression ratio. There are, however, differences in octane requirement between different types of chamber at the same compression ratio. A critical dimension is the *flame travel distance*: the distance between the spark plug electrodes and the farthest combustion chamber wall. Thus spark plug location in the chamber and the cylinder bore are important factors: center plug locations are better; larger size cylinders are worse. Whether more turbulent modern combustion chambers effect octane requirement much is unclear: increased in-cylinder heat losses, and faster burn, have opposite effects on peak pressures. The latter phenomenon also decreases flame travel time. These effects (for a given size engine) do not appear to be that large.

Operating variables that affect the temperature or pressure time histories of the end-gas during combustion or the basic autoignition characteristics of the unburned fuel, air, residual mixture will also affect the engine's octane requirement. The most important additional variables which increase or decrease octane requirement in a consistent manner are listed in Table 15.5. Relative spark advance has a major impact on knock: since it is also easy to adjust, it is the engine variable most commonly used to control knock. Studies show that typically 0.5 to 1.0 RON reduction is achieved per degree of retard. Atmospheric conditions—pressure, temperature, and humidity—all affect the engine's octane number requirement. The fuel/air equivalence ratio affects the octane requirement of an engine. The highest requirement is for slightly rich mixtures: see Fig. 9.71. Increasing richness and leanness about this point decrease the octane requirement substantially.

TABLE 15.5 Engine conditions affecting octane number requirement

Octane number requirement goes *up* when:	Octane number requirement goes *down* when:
1. Ignition timing is advanced.	1. Ignition timing is retarded.
2. Air density rises due to a larger throttle opening or higher barometric pressure.	2. Engine is operated at higher altitudes or smaller throttle opening, or lower barometric pressure.
3. Humidity or moisture content of the air decreases.	3. Humidity of the air increases.
4. Intake air temperature is increased.	4. Intake air temperature is decreased.
5. Coolant temperature is increased.	5. Air/fuel ratio is richer or leaner than that producing maximum knock ($\lambda \approx 0.95$).
6. Antifreeze (glycol/water mixture) engine coolant is used.	6. Exhaust gas is recycled.
7. Engine load is increased.	7. Engine load is reduced.

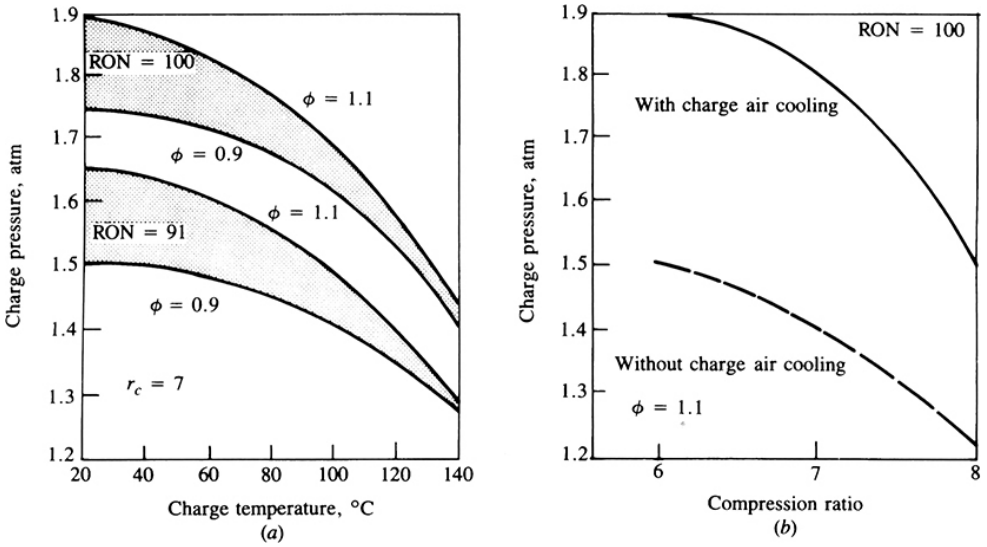


Figure 15.30 Dependence of turbocharged gasoline engine knock limits on: (*a*) charge pressure, temperature, and equivalence ratio *f*, with $r_c = 7$, 2500 rev/min, MBT timing, 91 and 100 research octane number fuel; (*b*) charge pressure and compression ratio, without and with charge air cooling (to 60°C), 2500 rev/min, MBT timing, $f = 1.1$, 100 RON fuel.[40]

In turbocharged engines the knock limit is more constraining, and the compression ratio is usually about one unit lower than an equivalent naturally-aspirated engine. Figure 15.28 illustrates how fuel octane requirement, intake air temperature relative fuel/air equivalence ratio, and compression ratio impact the knock-limited boost pressure in a turbocharged gasoline engine. All the above parameters have a significant impact. The case for an intercooler to cool the air charge prior to entry to the cylinder is clear, as is the importance of higher-octane fuel. As already mentioned, the

evaporative charge cooling effect of direct fuel injection makes this injection technology especially attractive with turbocharged engines. The theoretical in-cylinder evaporative cooling impact of DI with a (hydrocarbon) gasoline is about 20 degrees C. Studies suggest that DI fueling systems relative to a port-fuel-injection system realize about 70% more of that limit in actual engines (i.e., about 14 deg C). Addition of ethanol to the fuel (U.S. gasolines usually contain close to 10% ethanol, E10), with its much higher heat of vaporization than gasoline, increases this evaporative cooling effect significantly. E10 provides about an additional 5 deg C of charge cooling. [39]

As engine load is decreased, the engine's octane requirement significantly decreases. Figure 15.31 shows this behavior: the Octane Number (RON) of the Primary Reference Fuel (PRF)[k] required to avoid knock, as a function of load (bmep), at three engine speeds. There are two distinct regimes: at higher loads (above about 800 kPa bmep) where the slope is modest; at lower loads (below about 600 kPa bmep) where the slope is much steeper. Clearly, much of the time in a given application, gasoline SI engines would be able to operate knock free with octane levels well below (of order 50 RON) what is needed to avoid knock at high loads (90 to 100 RON). Whether or not we will transition to matching more closely the fuel octane supplied to the engine with the actual engine's operating (and varying) octane requirement in the future is unclear. Currently, our approach of using a given gasoline fuel, with a single high-octane rating (high enough to avoid knock at high engine load) over all engine loads and speeds is simpler and cost effective.

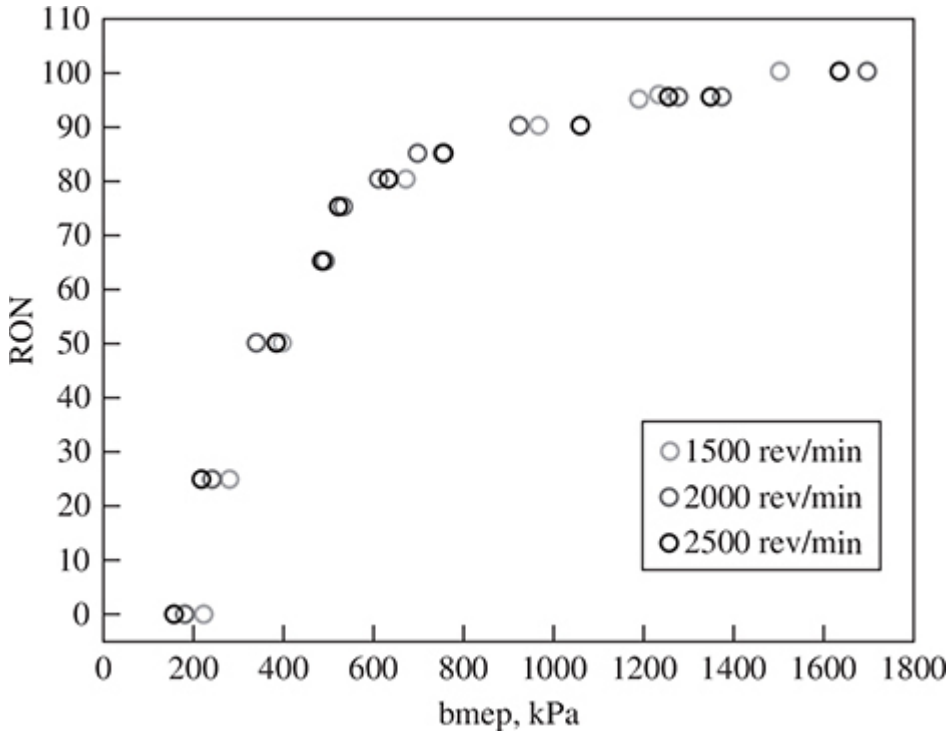


Figure 15.31 Octane requirement (primary reference fuels) as a function of engine load (bmep) for a 2-liter four-cylinder turbocharged gasoline SI engine, at 1500, 2000, and 2500 rev/min. Stoichiometric operation at MBT spark timing. [41]

Use of low-percentage ethanol gasoline blends (e.g., E5 in Europe, E10 in the U.S.) is allowing use of lower-octane base hydrocarbon gasolines which the ethanol upgrades. This ethanol octane blending benefit is illustrated in Fig. 9.70. Depending on the base gasoline composition and octane, 10% ethanol increases the base octane RON by some 3-7 octane numbers.

15.4.5 SI Engine Emissions

Fast-burn combustion systems allow use of substantial EGR (up to about 15%) to control engine-out NO_x emissions, improve bsfc, and hold the EGR-produced increase in HC emissions to modest levels. These benefits are illustrated in the set of graphs in Fig. 15.32 where key combustion and emissions parameters are plotted against EGR rate, at constant engine operating conditions. MBT spark advance, coefficient of variation in imep

(which defines the combustion cyclic variability), bsfc, and bsNO $_x$ and bsHC are plotted with increasing EGR for a moderate and fast-burn combustion system. The difference in burn rate is apparent from the MBT spark advance: the slower-burning combustion system requires 10 to 15 degrees more advance. One consequence of faster burn is less combustion variability, especially as dilution with EGR becomes significant. Acceptable COV of imep is a few percent, depending on load. The fast-burn system maintains such levels out to about 15% EGR. Substantial NO $_x$ reductions are realized as EGR increases, essentially independent of burn rate; but only the fast-burn system provides acceptable combustion stability. Also, the increase in HC emissions up to this EGR level is more modest.

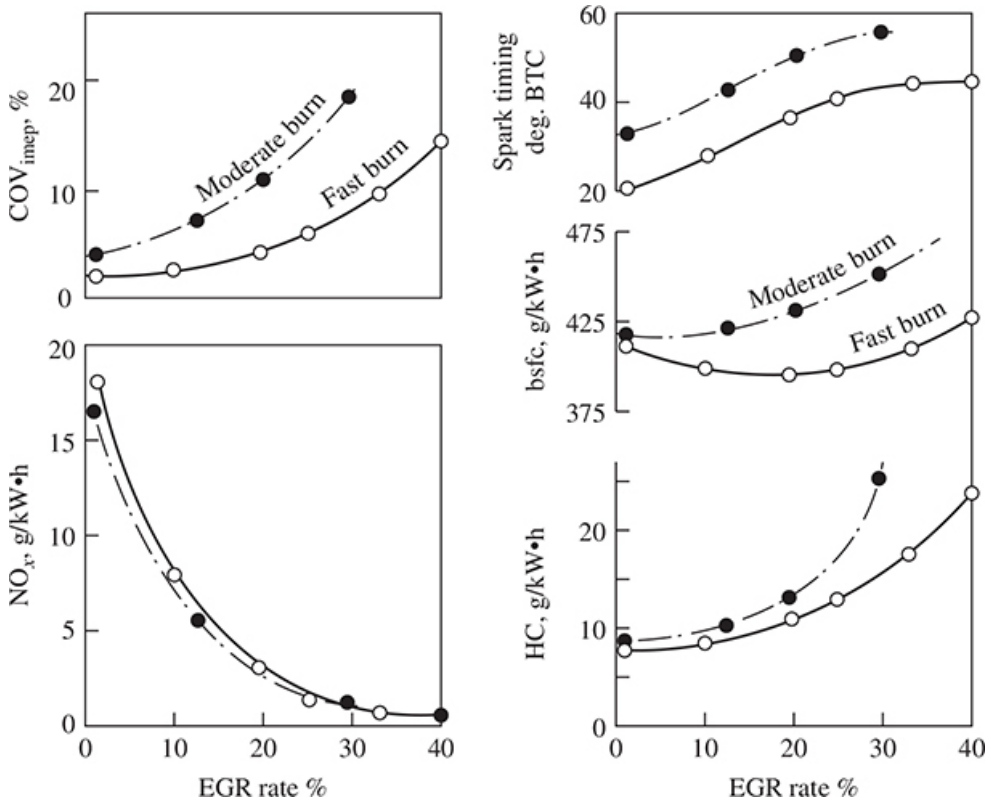


Figure 15.32 Illustration of how a fast-burn combustion system extends the EGR tolerance limit, improves engine-out NO_x and HC emissions, and fuel consumption, with acceptable combustion stability. Data for four-cylinder gasoline engine, stoichiometric operation, MBT spark timing, at 30 N·m torque, 1400 rev/min. COV, coefficient of variation in imep. [33]

Especially important is the impact of fast burn on fuel consumption. Increasing dilution of the fuel-air mixture with EGR has several benefits: (1) Lower burned gas temperatures (less dissociation), so higher expansion stroke γ; (2) lower heat loss to combustion chamber walls; (3) at the same engine part-load torque, intake system pressure is increased and pumping work decreases. All these effects are largely independent of burn rate. But increasing EGR slows down the burn rate which worsens fuel consumption. For a slower burning combustion system this negative burn-rate impact almost immediately offsets the three positive factors listed above, and then overshadows them. For the fast-burn system, this negative impact is much smaller and only becomes significant as EGR levels rise above about 15%. At that point, the fuel consumption benefit with fast burn relative to zero EGR is about 5%. This combined efficiency and emissions benefit with a fast-burn combustion system through use of significant EGR, while maintaining good combustion characteristics has been an important development.

15.4.6 Optimization

Combustion system optimization is a core requirement for improving gasoline engine performance and efficiency, and reducing engine-out emissions. The objectives and challenges involved for a turbocharged SI engine (fueled with gasoline) are summarized on the performance map in Fig. 15.33. Two separate areas on the map (torque or bmep versus speed) are critical for automotive application: the wide-open-throttle curve; and the part-throttle lower load and speed shaded region shown which, for this SI engine application is normally centered at one-quarter maximum torque and 1500 to 2000 rev/min (depending on cylinder size through $\overline{S}_p$, the mean piston speed). [l] Expanding the wide-open throttle "upward and outward" is clearly advantageous. Increasing engine output allows engine downsizing, which significantly improves part-load fuel consumption. For turbo engines, shifting the rising low speed torque boundary to lower speeds is especially important since it improves the response time of the engine to rapid throttle openings (decreasing the *turbo-lag* problem). Knock is a constraint at WOT: first at low speeds, then also at mid and higher speeds.

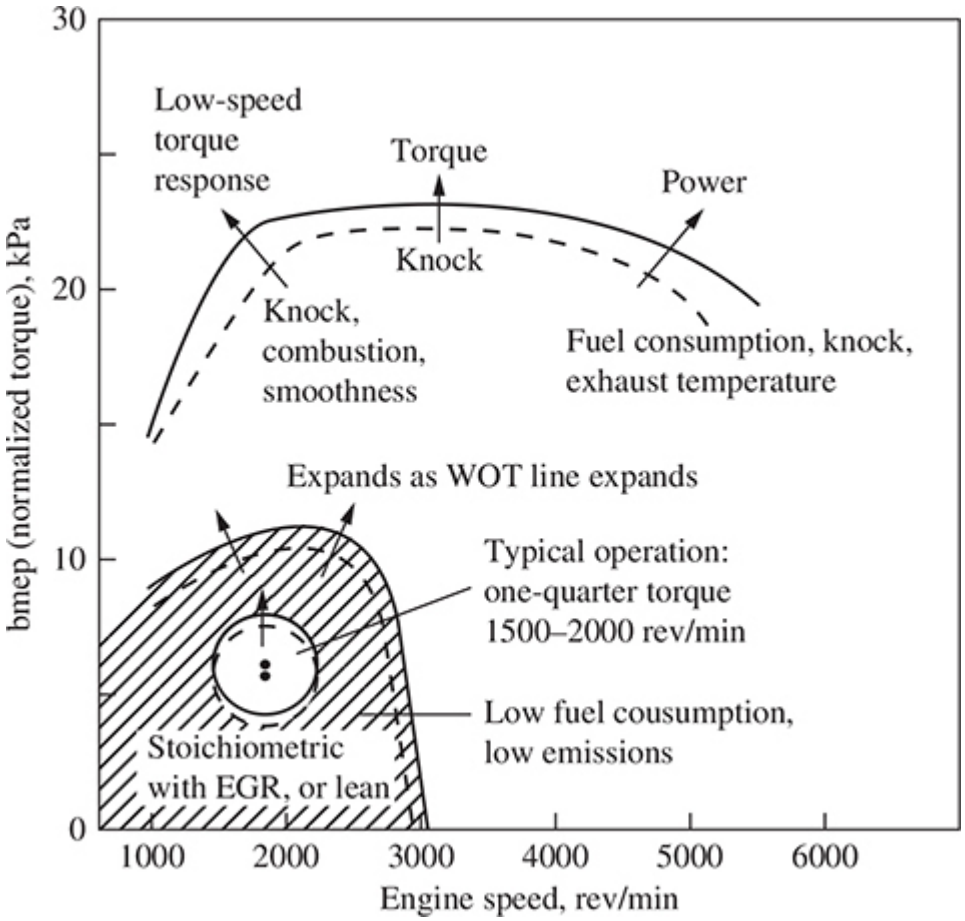


Figure 15.33 Opportunities for improvement, and the constraints, on a turbocharged gasoline-engine performance map in the key wide-open-throttle and part-load low-speed engine operating regimes. (Developed from Ref. 42.)

At a more detailed level, the following sequence of steps in a combustion chamber development process is most logical. First should come the selection of the best chamber geometry. Geometric optimization can result in substantial benefits and carries no significant penalties. Chamber geometry involves cylinder-head and piston-crown shape, and spark plug location. Open chambers such as the pent-roof or hemispherical cylinder head, with near central plug location, give close to the maximum flame front surface area (and hence a faster burn), have the lowest chamber surface area in contact with the burned gases (and therefore the lowest heat transfer), and have inclined valves which give high volumetric efficiency. Spark plug

location close to the center of the chamber is especially important in obtaining a fast burn rate.

Following this first step, two problem areas may remain: the chamber may have a higher octane requirement than is desired, and the burn rate may not be fast enough to absorb the EGR or excess air dilution required at part load to meet the emissions and fuel consumption objectives. Positioning the spark plug as close as possible to the cylinder axis will have reduced the octane requirement for that particular chamber shape. Depending on chamber design details, some squish area could be introduced. However, the octane advantage of chamber designs which contain substantial squish is offset, at least in part, by their higher heat losses. For example, a unit compression ratio increase results in a 3% increase in efficiency at part load (with downsizing to hold torque constant). However, if the measures required to increase the compression ratio resulted in a 10% increase in heat transfer, engine efficiency would not improve.

The next step should be to reduce the cyclic variability in the combustion process to the maximum extent feasible, by improving the uniformity of the in-cylinder fuel, air, and EGR mixture. Delivery of equal amounts of each of these constituents to each cylinder, provision for good mixing between constituents in the intake processes, and accurate control of mixture composition during engine transients are all especially important. So is achieving closely similar flow patterns within each engine cylinder during intake so as to obtain equal burn rates in all cylinders. Attention to these intake process and mixture preparation details will always improve engine operation and carries no performance penalties.

However, the burn rate may still not be fast enough, especially during the early stages of flame development, and cyclic variability may still be too high to meet the engine's operational goals. Then higher turbulence levels during combustion must be achieved. This is usually best done by creating a stronger tumbling flow during the induction process. The appropriate method for doing this will depend on any geometric manufacturing constraints and cost issues. With no geometric constraints, use of a four-valve cylinder head with low intake port axes, possibly with intake port throttling plates do direct the flow appropriately at light load are likely to have the lowest power penalties. It is especially important that only the minimum additional turbulence required to achieve the overall performance objectives be added

at this stage. Higher than necessary gas velocities within the cylinder result in excessive heat losses and lower volumetric efficiency.

In summary, to meet the objectives of a fast, repeatable, and robust combustion process with high volumetric efficiency, low heat transfer, and acceptable octane requirement, combustion chamber development should proceed through the following steps. [35]

- Optimize the chamber geometry within the design constraints for maximum flame front area, minimum burned gas/chamber wall contact area, and largest intake valve size.
- Obtain additional reductions in the cyclic combustion variability by improving mixture distribution and uniformity and by creating flow patterns into each cylinder that are close to identical.
- Achieve any additional improvement in burn rate and cyclic variability required to meet objectives by increasing in-cylinder turbulence to the needed level. This is usually best done by creating tumble during the induction process.

15.5 VARIABLES THAT AFFECT DIESEL ENGINE PERFORMANCE, EFFICIENCY, AND EMISSIONS

The performance characteristics of a turbocharged high-speed DI automotive diesel engine over the engine's load and speed range have already been introduced in Sec. 15.2.3 . Here we examine diesel performance maps in more detail.

15.5.1 Load and Speed

As with SI engines, performance maps where bsfc contours are plotted on a graph of torque or bmep versus engine speed are commonly used to describe the effects of variations in load and speed. Figure 15.34 shows the performance map for a four-stroke cycle medium-swirl turbocharged and aftercooled truck DI diesel engine (similar to that in Fig. 1.31). Maximum

rated power for this 8-liter displacement in-line six-cylinder engine is 188 kW at 2500 rev/min, maximum torque is close to 800 N·m at 1500 rev/min, minimum bsfc (at 1200 rev/min) is just below 200 g/kW·h. These practical parameters correspond to a brake mean effective pressure (bmep) of 1260 kPa and a brake fuel conversion efficiency of 42.5%. The gross indicated fuel conversion efficiency would be over 50%.

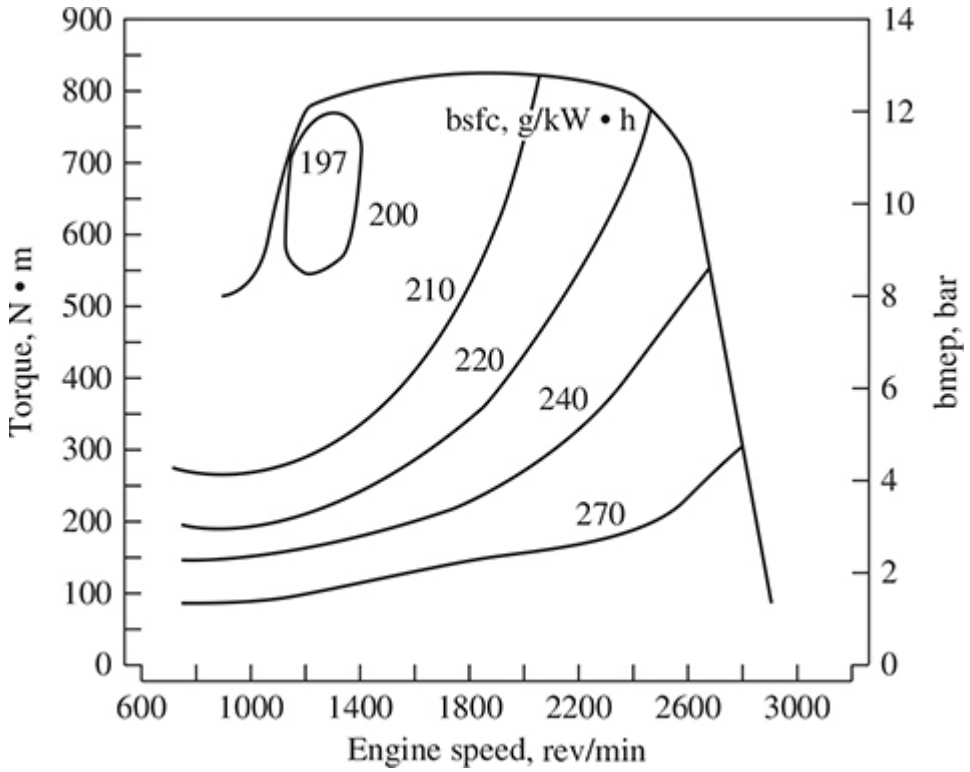


Figure 15.34 Performance map for an 8-liter in-line six-cylinder, turbocharged and aftercooled, DI diesel engine with a medium-swirl, central injection, multi-hole nozzle combustion system. Contours of constant bsfc in grams per kilowatt-hour are shown. Bore = 114 mm, stroke = 130 mm. r_c = 16.5.[43]

Turbocharged-diesel-engine performance maps are usually similar in shape. Torque and bmep rise rapidly as engine speed increases above about 1000 rev/min, then remain relatively flat through the mid-speed range to close to the rated speed at which maximum power is realized. Figure 15.6 shows the performance map of an automotive high-speed light-duty diesel of much smaller size that operates up to 4500 rev/min. The individual cylinder

displaced volume of this 2.46-liter five-cylinder in-line engine (492 cm ^{3}/cylinder) is much less than the cylinder size of the 8-liter medium-duty six-cylinder in-line engine of Fig. 15.34 (which is 1333 cm). For diesels such as the one illustrated in Fig. 15.34 maximum power occurs at mean piston speeds of 10 to 12 m/s: for the HSDI smaller diesel of Fig. 15.6 it occurs at 14 m/s. [10] The increase in bsfc from its minimum value with increasing speed at constant load is due to the increase in friction mep, partly offset by the effect on efficiency of decreasing importance of heat losses per cycle. The increase in bsfc with decreasing load at constant speed is dominated by the decreasing mechanical efficiency as bmep is reduced. The indicated fuel conversion efficiency increase, as the fuel/air equivalence ratio is decreased, partly offsets this. The trends in bsfc when increasing load at constant speed and increasing speed at constant load from the minimum are modest. They are the net results of (1) the increase in mechanical efficiency and decrease in indicated fuel conversion efficiency as the load (and fuel/air equivalence ratio) increases and (2) decreasing indicated efficiency due to increasing importance of heat losses, and increasing mechanical efficiency, as the speed decreases. Note that the drop in efficiency (increase in bsfc) as load decreases is not that large (10%) until load drops below about one-third the maximum load. The significantly better part-load efficiency of the diesel relative to the gasoline engine is one of the diesels major advantages.

Diesel engine performance and efficiency are steadily improving. Use of higher boost levels in turbocharged diesels (4 bar or more) increases bmep levels above about 20 bar. Best brake fuel conversion efficiencies are rising through mid-40% levels.

Figure 15.35 shows the effect of load on engine-out NO_x and HC emissions for a naturally-aspirated DI diesel engine. For this DI engine NO_x concentrations rise steadily as the fuel/air ratio increases with increasing bmep at constant injection timing. The increasing quantity of fuel injected per cycle results in an increasing amount of close-to-stoichiometric diffusion-flame combustion near the peak pressure and temperature (see Sec. 11.2.4). The high HC at idle and light load are thought to result from fuel mixing to too lean an equivalence ratio to burn. If diesel engines are over-fueled at high load, HC emissions then rise rapidly. These HC mechanisms are described in Sec. 11.4.4 . Injection timing affects NO_x and HC emissions significantly, as discussed in Sec. 15.5.3 below.

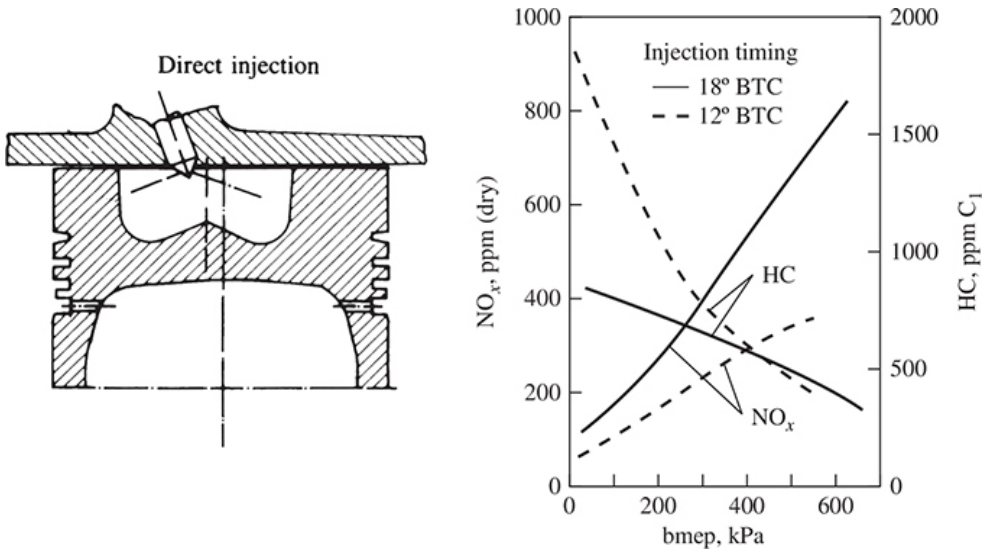


Figure 15.35 Effect of load on naturally-aspirated DI diesel engine-out NO_x and HC emissions at rated speed, with two injection timings. The direct-injection combustion system is shown. Six-cylinder, 5.9-liter displaced volume, engine: $r_c = 17$, rated speed = 2800 rev/min. [44]

With structurally rugged component design, aftercooled turbocharged medium-speed diesel engines with swirl can utilize higher boost and

generate much higher bmep. Usually, wastegate control of boost level is no longer required. Figure 15.36 shows the performance characteristics of a V-8 diesel engine with its compressor map and full-load boost characteristic. This turbocharged intercooled engine achieves a maximum bmep of about 16 bar at the maximum torque (2700 N·m), and bsfc below 200 g/kW·h between the maximum torque speed and rated power. Boost pressure at full load increases continuously over the engine speed range. Maximum mean piston speed for this size TC engine is 10 m/s.

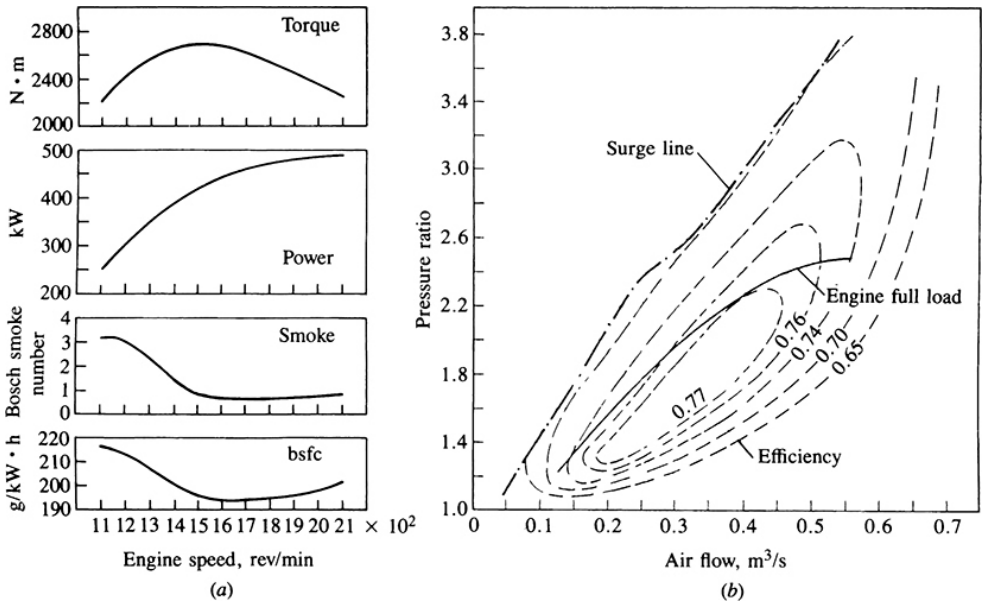


Figure 15.36 Performance characteristics of medium-speed turbocharged aftercooled DI diesel engine: (*a*) torque, power, Bosch smoke number, and bsfc for V-12 cylinder 21.6-liter version; (*b*) compressor characteristics and engine full-load line for V-8 cylinder 14.4-liter version. Bore = 128 mm, stroke = 140 mm, r_c = 15. [45]

Higher outputs can be obtained with two-stage turbocharged aftercooled diesel engines, the arrangement shown in Figs. 6.45*d* and *f*. The performance characteristics of such a high bmep (17.4 bar) six-cylinder engine of 14-liter displacement are shown in Fig. 15.37. The high airflow requires an overall pressure ratio of 3 at sea level ambient conditions (rising to 4 at 3658 m altitude). This was obtained at lower cost with two turbochargers in series than with a multistage single turbocharger. At rated conditions, the maximum

cylinder pressure is 12.7 MPa and the maximum mean piston speed is 10.6 m/s.

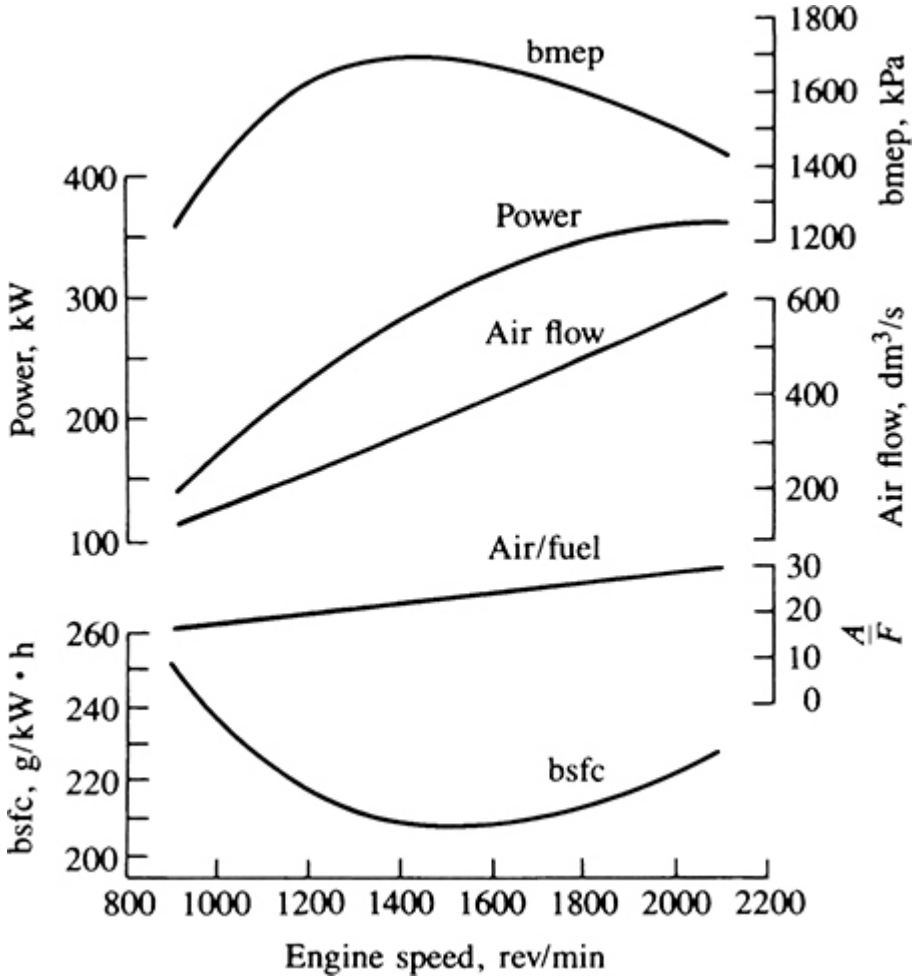


Figure 15.37 Operating characteristics of 14-liter six-cylinder two-stage turbocharged aftercooled quiescent-chamber DI diesel engine. Maximum bmep = 17.4 bar. Boost pressure ratio at rated power = 3. Bore = 140 mm, stroke = 152 mm. [46]

Additional gains in efficiency with these heavy-duty automotive diesel engines can be achieved with turbocompounding: some of the available energy in the exhaust gases is captured in an additional turbine which is geared directly to the engine drive shaft. The above discussion indicates that typical turbocharged DI diesel engines achieve bsfc levels about 200 g/kW·h (brake fuel conversion efficiencies of over 40%). With the increased cylinder pressure capability, higher fuel-injection pressures, and lower-

temperature aftercooling, lower bsfc can be achieved. With turbocompounding, bsfc values can be reduced another 5% to about 180 g/kW · h, or a brake efficiency of about 50%, at rated power.

The largest four-stroke cycle DI diesel engines are used for marine propulsion. An example is the Sulzer 400-mm bore, 480-mm stroke engine which produces 640 kW per cylinder at 580 rev/min ($\overline{S}_p$ = 9.3 m/s). Very high bmep levels (22 bar) are achieved at maximum continuous rated power through progress in turbocharger design and engine improvements which allow higher maximum cylinder pressures. These, combined with optimization of gas exchange and combustion processes, achieve bsfc values of 185 to 190 g/kW · h (45 to 46% brake efficiency). [47]

Many diesel system concepts are being examined which promise even higher output and/or efficiency. Variable-geometry turbocharger-turbine nozzles are extensively used to increase utilization of the available exhaust-gas energy at lower engine speeds. Diesel engine systems can increase power through additional exhaust energy recovery in devices such as compounded turbines and exhaust-heated Rankine cycle and thermo-electric systems.

15.5.2 Combustion-System Design

As discussed in Sec. 10.2 , a variety of combustion systems (combustion chamber shape, air motion, fuel injection and spray geometry) are used to obtain both low fuel consumption and pollutant emissions. Figure 10.1 shows the basic bowl-in-piston geometries, and the types of in-cylinder air motion that are used over the engine size range (bore 75 m to 150 mm). Figure 15.38 illustrates the combustion process in a medium-duty diesel engine, and identifies the critical combustion-system design features.

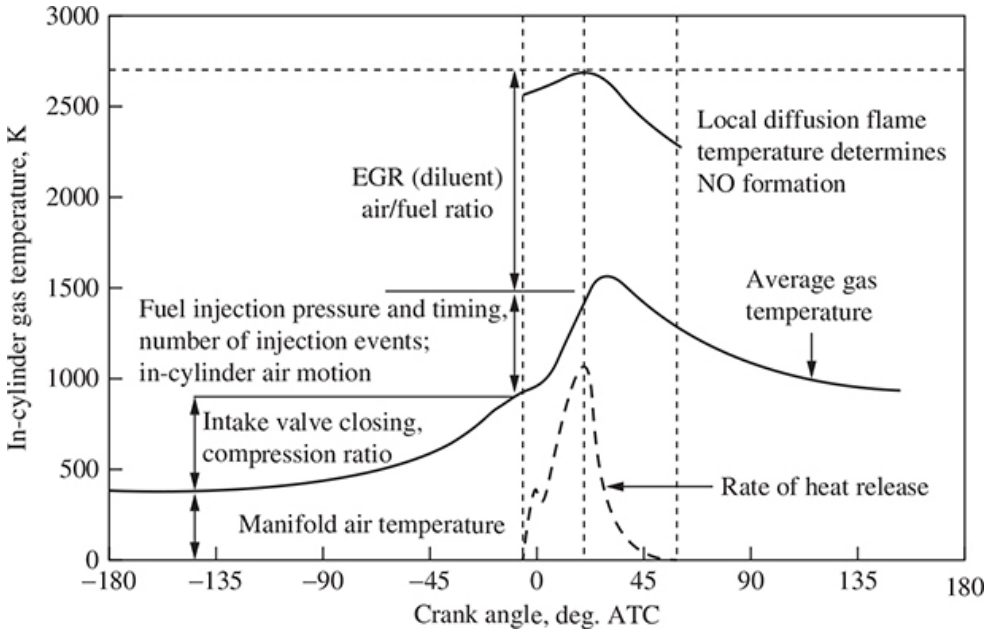


Figure 15.38 Important combustion-system design features: heat release rate and critical in-cylinder gas temperature profiles typical of a medium-duty diesel engine.[48]

The high gas temperatures within the close-to-stoichiometric diffusion flames surrounding each fuel-rich spray (the main diesel combustion phase) is the primary NO formation region. These diffusion flame temperatures are much higher than the average in-cylinder gas temperatures. Recycled exhaust gas, at part load, is used to reduce diesel engine NO_x emissions. Note that since diesel engines operate with the airflow unthrottled, at part load the CO_2 and H_2O concentrations in exhaust gas are low; they are essentially proportional to the overall fuel/air ratio. Because of this, high EGR levels are required for significant reductions in engine-out NO_x emissions. Figure 11.17 *b* shows how NO_x concentrations decrease as a DI diesel engine inlet airflow is diluted at a constant fueling rate. The dilution can be expressed in terms of oxygen concentration in the mixture after dilution. Figure 15.39 shows how engine-out NO_x emissions decrease as the EGR increase decreases the intake oxygen concentration. The impacts of EGR are discussed more fully in Sec. 15.5.3 .

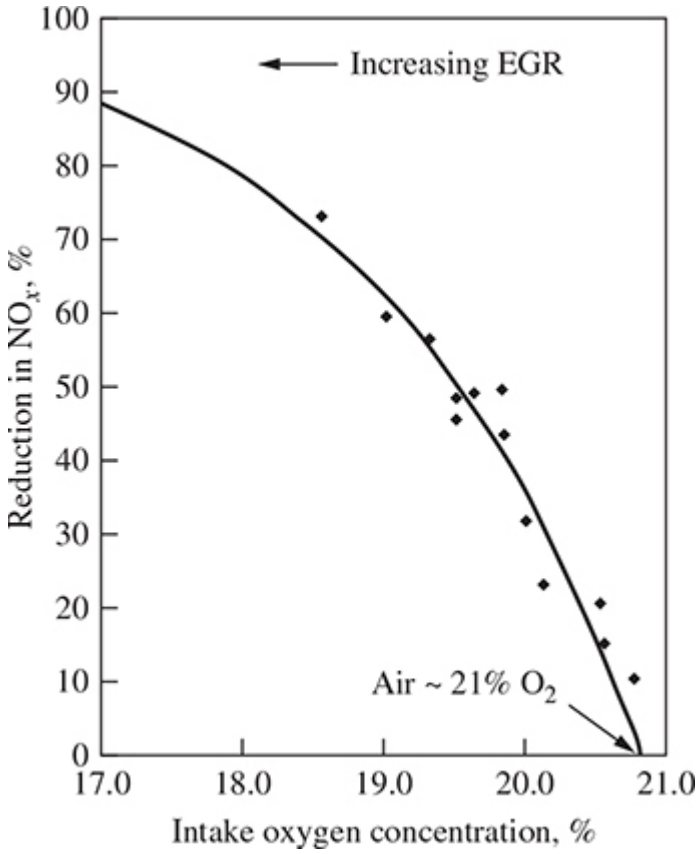


Figure 15.39 Impact of increasing EGR (expressed through the corresponding decrease in oxygen concentration in the intake air and EGR mixture) on the reduction in engine-out NO_x emissions.[48]

As Fig. 15.38 indicates, fuel injection system pressure and fuel injection timing (relative to the timing that gives minimum bsfc) influence the heat release rate, and therefore maximum cylinder pressure. Also, with multiple injections (e.g., pilot injection), the rate-of-fuel-burning profile is affected. Mixing rates can be increased through use of swirl and optimized bowl-in-piston combustion chamber shapes, thus influencing the rate of chemical energy release. Finally, the amount of air compression (from about IVC to top center), and the intake air temperature, affect the mixture temperature just before combustion starts: this affects the mixture burning rate and the peak (diffusion flame) temperatures.

As engine size decreases and maximum engine speed increases, increasing amounts of air swirl within the cylinder are used in direct-injection diesel engines to achieve adequately fast fuel-air mixing rates (see

Sec. 10.2.1). In these smaller-size engines, use of a bowl-in-piston combustion chamber (Figs. 10.1*b* and *c*) results in substantial swirl amplification at the end of the compression process (Sec. 8.3.3). Here, the impacts of varying air swirl on the performance and emissions characteristics of this type of DI engine are reviewed.

Swirl is used to increase the rate of air entrainment into the developing fuel sprays, and thus the fuel-air mixing rate. So one would expect the overall duration of the combustion process to shorten as swirl increases, and the emissions that depend on the local fuel/air equivalence ratio to be dependent on swirl level. The injection timing which gives minimum bsfc shifts toward TC as the swirl ratio increases due to the decreasing total combustion duration. For each combustion chamber design, there is an optimum swirl ratio for minimum bsfc: higher swirl levels continue to increase fuel-air mixing rates, but heat transfer increases also and eventually offsets the increased-mixing-rate gain. Particulate and CO emissions decrease as swirl increases due to more rapid fuel-air mixing and less-rich within-the-spray mixture. NO_x emissions increase with increasing swirl. At constant injection timing, however, part of the increase is due to the effect of injection advance relative to the optimum timing due to the shorter combustion process.

In production engines, the various types of port design shown in Fig. 8.14 can be used to generate swirl during the induction process. Of these, the helical ports are most effective at producing relatively-uniform high swirl with the minimum loss in volumetric efficiency.

The geometry of the bowl-in-piston combustion chamber governs the extent to which induction-generated swirl is amplified during compression. The flow field in the bowl during fuel injection is also dependent on the interaction between this swirling flow and the squish motion which occurs as the top of the piston crown approaches the cylinder head (see Sec. 8.5). Two types of bowl-in-piston design for multihole fuel nozzle DI engines are shown in Fig. 15.40. Designs (e.g., Fig. 15.32*a*) used when really high mixing rates are not required have the bowl sides essentially parallel to the cylinder liner. Note that it may be necessary to offset the bowl axis from the cylinder axis and the injector nozzle hole locations from the bowl axis, due to the geometric constraints imposed by the valves. Smaller-size higher-speed designs use a reentrant bowl (Fig. 15.40*b*) to promote more rapid fuel-air mixing within the bowl. The squish-swirl interaction with highly reentrant bowl designs differs markedly from the interaction in nonreentrant

bowls as shown by the two flow patterns set up in a diametral plane, at the bottom of the figure. With the straight-sided bowl, the swirling air entering the bowl flows down to the base of the bowl, then inward and upward in a toroidal motion. In reentrant bowls the swirling air entering the bowl spreads downward and outward into the undercut region, and then divides into a stream rising up the bowl sides and a stream flowing along the bowl base. Reentrant chambers produce higher swirl at the end of compression, and maintain a high swirl level further into the expansion stroke. [49] Reentrant chambers usually achieve lower HC and smoke emissions and slightly lower bsfc, especially at retarded injection timings.

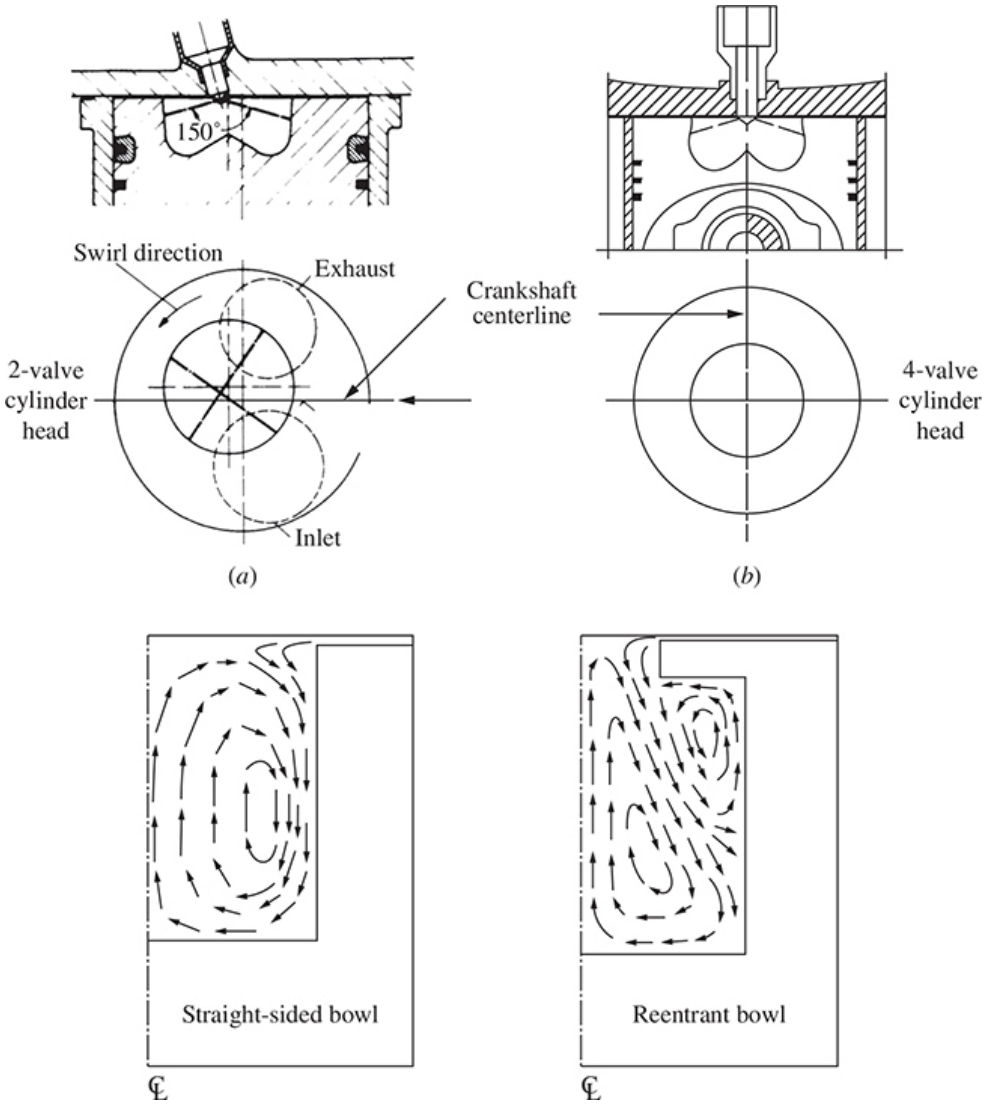


Figure 15.40 Bowl-in-piston chamber designs for DI diesel engines with swirl: (*a*) straight-sided bowl; (*b*) reentrant bowl. Below, flow pattern set up

in diametral plane by squish-swirl interaction in the straight-sided bowl and the reentrant bowl combustion chambers. See discussion in text. [44, 49]

The airflow field within bowl-in-piston combustion chambers when fuel injection occurs is highly complex. However, certain generalizations hold: for example, reducing the bowl diameter at a constant compression ratio increases the swirl levels in the bowl at TC (see Sec. 8.3.3 and the accompanying text) which decreases smoke and increases NO_x, and HC emissions.[44] However, unraveling the squish-swirl interaction is challenging.

15.5.3 Fuel Injection and EGR

Fuel-injection timing essentially controls the crank angle at which combustion starts. While the state of the air into which the fuel is injected changes as injection timing is varied, and thus the ignition delay will vary, these effects are predictable (see Sec. 10.5.4). The fuel-injection rate, fuel nozzle design (including number of holes), the number of injections per cycle, and fuel-injection pressure all affect the characteristics of the diesel fuel spray and its mixing with air in the combustion chamber.

Figure 15.41 shows the effect on performance and emissions of varying injection timing, in a medium-swirl DI diesel engine at fixed speed and constant fuel delivery per cycle. The DI engine shows an optimum bsfc and bmep at a specific start of injection for a given injection duration. This optimum injection timing gives maximum brake torque, though the designation MBT timing is less commonly used with diesels than with SI engines.

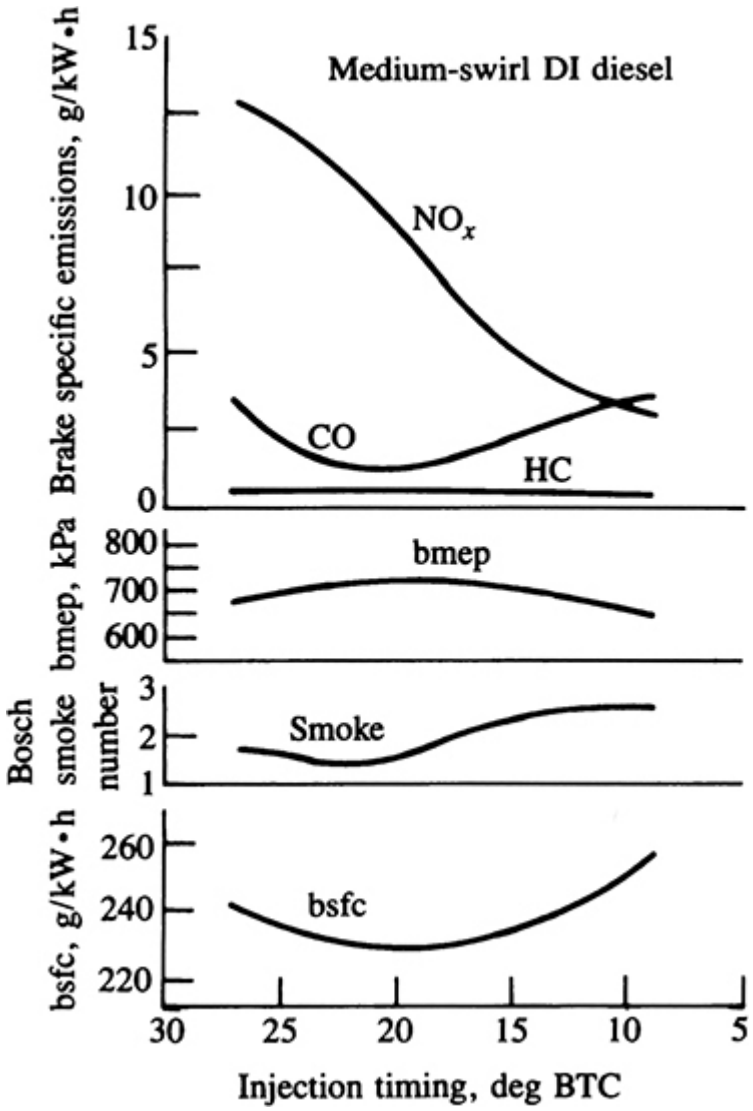


Figure 15.41 Effect of start-of-injection timing on diesel engine performance and emissions: medium-swirl DI diesel engine with deep combustion bowl and four-hole injection nozzle, 2600 rev/min, fuel delivery 75 mm^3/cycle, fuel/air equivalence ratio 0.69.[44]

Injection timing variations have a strong effect on engine-out NO_x emissions for DI diesel engines. Retarded injection is commonly used to help control NO_x. It gives substantial reductions, initially with modest bsfc penalty. For DI engines, at high load, specific HC emissions are low and vary only modestly with injection timing. At lighter loads, HC emissions are higher and increase as injection becomes significantly retarded from

optimum. This trend is especially pronounced at idle. Retarding timing generally increases particulate emissions and smoke, though trends vary between different types and designs of diesel engine. Mass particulate emissions increase as injection is retarded.

The injection rate depends on the fuel-injector nozzle open-area and injection pressure. Higher injection rates result in higher fuel-air mixing rates, and hence higher heat-release rates (see Sec. 10.6.3). For a given amount of fuel injected per cylinder per cycle, as the injection rate is increased the optimum injection timing moves closer to TC. The higher heat-release rates and shorter overall combustion process that result from the increased injection rate decrease the minimum bsfc at optimum injection timing; however, a limit to these benefits is eventually reached.

Increasing the injection rate increases NO_x emissions and decreases smoke and particulate emissions. The controlling physical process is the rate of fuel-air mixing in the combustion chamber so, at constant fuel injected per cylinder per cycle, both increased injection pressure at fixed nozzle orifice area (which reduces injection duration) and reduced nozzle area at fixed injection duration produce these trends.

A major factor is diesel NO_x control is use of exhaust gas cycle, EGR. The fundamentals of EGR's impact on emissions were reviewed in Sec. 11.2.4 . The nitrogen, CO_2 and H_2O in the exhaust recycled into the intake air act as an additional diluent which reduces the burned gas temperatures by an amount in rough proportion to the diluent fraction. With the diesel's always lean operation ($\phi \leq 0.7$), the recycled exhaust gases can be thought of as air, mixed with a certain quantity of stoichiometric burned gases: the leaner the diesel's overall operation (the lighter the engine load), the smaller is the fraction of stoichiometric burned gas mixed with the air. Only the amount of stoichiometric burned gas (N_2, CO_2, and H_2O) acts as a "true" diluent: the "air component" in the EGR acts just like the intake air. Thus, the leaner the engine's operation, the higher must be the fraction of the overall exhaust recycled to achieve a given level of actual dilution of the intake mixture. Figure 11.17 *b* illustrates this effect via EGR's impact on engine-out NO_x emissions.

EGR, at high levels, significantly increases particulate matter emissions (often referred to as smoke) which consists largely of soot particles. Figure 15.42 shows typical NO_x and PM emissions data as a function of EGR. The

negative impact of EGR on PM is largely due to the reduced percentage of oxygen in the fuel-air mixture as exhaust gas is recycled (see Fig. 15.39). This both impacts the mixture within the sprays, where the soot is formed, and the diffusion flame temperature (see Fig. 15.38) where the soot is oxidized.

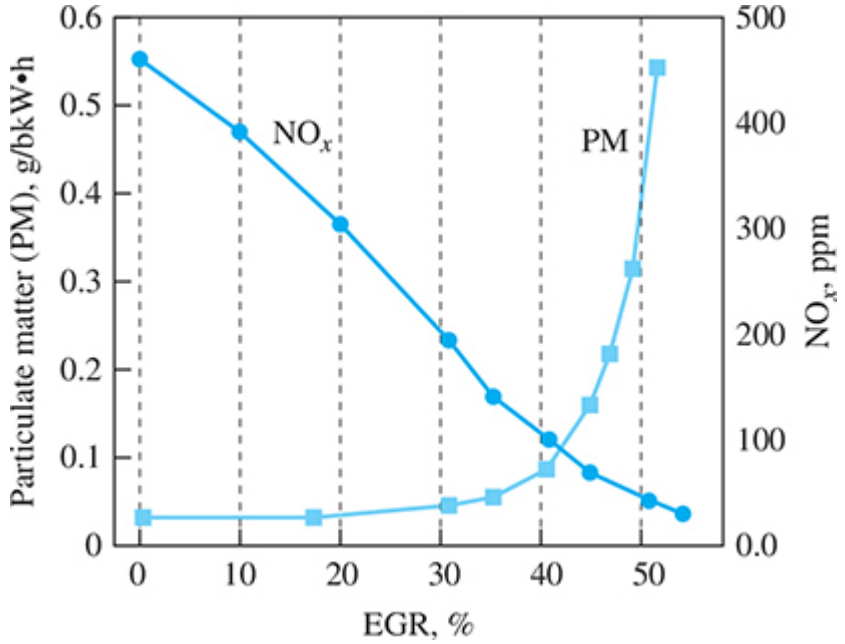


Figure 15.42 Typical response of engine-out brake-specific particulate (PM) emissions, and NO_x emissions (ppm), to EGR rate. Medium-duty DI diesel engine. [48]

This tradeoff between brake-specific particulate and NO_x engine-out emissions is well known, and for a given diesel engine configuration, is almost universal as injection timing and EGR levels are varied.

The engine designer's goal is obviously to achieve the best bsfc possible with emission levels low enough to satisfy the constraints imposed by emission standards. The variations in engine bsfc, NO_x, and particulate emissions described above involve tradeoffs that make achieving this goal especially difficult. As noted above, one tradeoff is between bsfc and engine-out $bsNO_x$. Injection retard from optimum injection timing decreases $bsNO_x$ at the expense of an increase in bsfc. A second important tradeoff is

that between engine-out NO_x and particulate emissions, illustrated for a DI diesel engine in Fig. 15.42.

15.5.4 Overall System Behavior

The factors that limit turbocharged diesel engine performance are different to those that limit turbocharged spark-ignition engines. The power output of naturally-aspirated diesel engines is limited by the maximum tolerable smoke emission levels, which occur at overall equivalence ratio values of about 0.7. Turbocharged diesel engine output is usually constrained by stress levels in critical mechanical components. These maximum stress levels limit the maximum cylinder pressure which can be tolerated under continuous operation, though the thermal loading of critical components can become limiting too. As boost pressure is raised, unless engine design and operating conditions are changed, the maximum pressures and thermal loadings will increase almost in proportion. In practice, the compression ratio utilized may need to be is reduced and the maximum fuel/air equivalence ratio must be reduced in turbocharged engines (relative to naturally-aspirated engines) to maintain peak pressures and thermal loadings at acceptable levels.

The different methods of supercharging internal combustion engines were reviewed in Sec. 6.8 . Turbocharging and mechanical supercharging with a compressor are alternative methods of boosting the performance of diesel engines. The density of the charge air is the parameter that determines the amount of charge, and hence the torque. Aftercooling in a heat exchanger of the air coming out of the compressor is now standard: it increases the density of the air entering the engine and a lower air temperature reduces all temperatures within the cylinder, an important factor in determining engine-out NO_x emissions.

Figure 15.43 shows the layout of a typical current large-truck diesel engine: engine, turbocharger, controls, plus emissions aftertreatment system. The engine-out emissions, fuel consumption, tradeoff is handled through a charge air cooler (aftercooler) downstream of the turbocharger compressor, a cooled EGR system (high pressure EGR is extracted upstream of the variable nozzle geometry (VG) turbine, cooled, and fed into the intake air downstream of the air throttle which modulates the post-compressor air

pressure so it is below the engine exhaust pressure), [m] and a multi-catalyst emissions-reduction aftertreatment system.

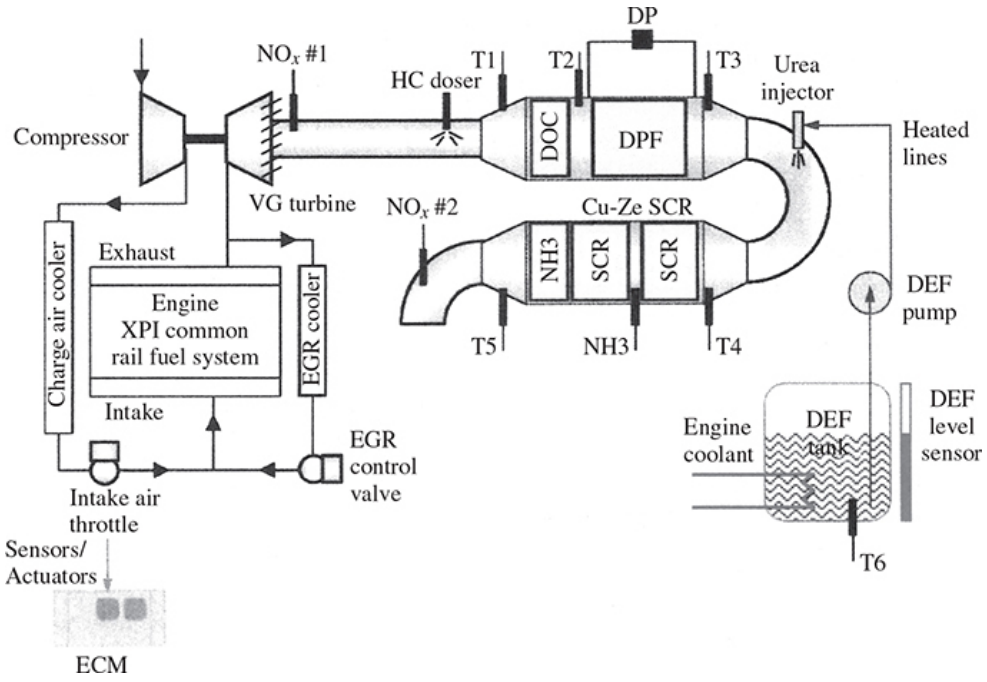


Figure 15.43 Turbocharged diesel, engine emissions controls, and aftertreatment system layout for heavy-duty truck application.[48]

This aftertreatment system consists of a diesel oxidation catalyst (DOC), a diesel particulate filter (DPF), urea to ammonia conversion catalyst (NH_3), and a selective catalytic reduction (SCR) catalyst (with two catalyst bricks). These operate as follows. The DOC oxidizes most of the NO in the exhaust to NO_2, which then burns up (oxidizes) the soot collected in the DPF at relatively low temperatures much of the time, through "passive regeneration." When higher exhaust gas temperatures are required, hydrocarbons are added upstream of the DOC as shown. Downstream of the DPF, urea ($CO(NH_2)_2$), the diesel emissions fluid (DEF), is injected, and in the hydrolysis catalyst produces ammonia, the reducing agent for the SCR system where NO and NO_2 are reduced to N_2 by reaction with NH_3. NO_x and NH_3 sensors and HC injection/dosing allow tight control of this multi-component aftertreatment system. The operation of these aftertreatment components is discussed in more detail in Sec. 11.6 .

15.6 TWO-STROKE CYCLE ENGINES

This section reviews the operating characteristics of the different types of two-stroke spark-ignition and compression-ignition engines. The effects of changes in the major design and operating variables on their operating characteristics are then related to the more fundamental engine processes such as scavenging, fluid flow, thermodynamics, combustion, heat transfer, and friction. The intent is to provide data on, and an explanation of, actual engine operation—performance, efficiency, and emissions.

15.6.1 Performance Parameters

The practical engine performance parameters of interest are power, torque, brake-specific fuel consumption, and emissions. Following the definitions in Chap. 2 , the brake power and the torque of a reciprocating engine can be expressed in terms of the brake mean effective pressure (bmep) as follows:

$$P_b = \begin{cases} \text{bmep} \times A_p \bar{S}_p/2 & \text{(for two-stroke cycle)} \\ \text{bmep} \times A_p \bar{S}_p/4 & \text{(for four-stroke cycle)} \end{cases} \tag{15.8}$$

$$T = \begin{cases} \text{bmep} \times V_d/(2\pi) & \text{(for two-stroke cycle)} \\ \text{bmep} \times V_d/(4\pi) & \text{(for four-stroke cycle)} \end{cases} \tag{15.9}$$

Thus, for a well-designed engine, where the maximum values of bmep and mean piston speed are airflow (or stress) limited, power is proportional to piston area, and torque to displaced volume. Introducing the definitions of trapping efficiency and delivery ratio from Sec. 6.6.2 , and of mechanical and fuel conversion efficiencies from Eqs. (2.24) and (2.29), bmep can be expressed in terms of other basic parameters as follows:

$$\text{bmep} = \eta_m(\eta_{tr} \Lambda)\eta_{f,i}\rho_{a,i} \frac{Q_{HV}}{\lambda(A/F)_s} \tag{15.10}$$

Equations (15.8) and (15.10) indicate that two-stroke engine power and torque depend on trapping efficiency and delivery ratio: for four-stroke

engines, the product of the delivery ratio and trapping efficiency is replaced by the volumetric efficiency.

Brake-specific fuel consumption and normalized indicators of emissions levels have the same definitions and defining equations for both types of engines. Emission rates can also be normalized by the engine's power output (specific power), or by the fuel flow rate.

It is instructive to use typical numbers for full-load and part-load operation to compare the performance of equivalent two-stroke and four-stroke engines. For typical spark-ignition engines of the same displaced volume, at the same air/fuel ratio, the ratio of their maximum torques is given by

$$\frac{T_{2\text{-stroke}}}{T_{4\text{-stroke}}} = \frac{\left[\eta_m(\eta_{tr}\Lambda)\eta_{f,i}/2\right]_{2\text{-stroke}}}{\left[\eta_m(\eta_v)\eta_{f,i}/4\right]_{4\text{-stroke}}} \tag{15.11}$$

This is consistent with typical maximum bmep values of about 850 kPa for this type of two-stroke engine compared with the 1200 kPa value for a four-valve-per-cylinder modern four-stroke SI engine. This gives a torque ratio [Eq. (15.9)] of

$$\frac{T_{2\text{-stroke}}}{T_{4\text{-stroke}}} = \frac{2\times \text{bmep}_{2\text{-stroke}}}{\text{bmep}_{4\text{-stroke}}} \approx \frac{2\times 850}{1200} = 1.4 \tag{15.12}$$

Note that if the bmep of the two engines could be the same (not likely to occur in practice), then the torque ratio would be 2.0.

A comparison of maximum powers from these two different-cycle engines is less straightforward because the maximum power in the two-stroke depends strongly on the scavenging performance of the engine. This depends on the scavenging system, the design of the ports, and especially on the sophistication of the exhaust system and its tuning. For two-stroke SI engines the specific power can vary between about 30 and 80 kW/liter. With the upper value typical of high-performance motorcycle engines. High-performance naturally-aspirated four-stroke passenger car SI engines average about 50 to 60 kW/liter.

In a similar way, for equivalent engines at the same operating conditions, we can estimate the ratio of the brake-specific fuel consumptions to be

$$\frac{\text{bsfc}_{\text{2-stroke}}}{\text{bsfc}_{\text{4-stroke}}} = \frac{(\eta_m \eta_{f,i})_{\text{4-stroke}}}{(\eta_m \eta_{f,i})_{\text{2-stroke}}} \approx \frac{(0.6 \times 0.38)}{(0.75 \times 0.36)} = 0.85 \tag{15.13}$$

Because the pumping losses under part-load conditions in two-stroke engines are smaller than in naturally-aspirated four-stroke engines, and the friction of the four-stroke's gas-exchange crank revolution is absent, the mechanical efficiency is higher. So, although the indicated fuel conversion efficiency of the two-stroke is somewhat lower, for the same fuel/air ratio and engine displaced volume, the brake-specific consumption at typical part-load operating conditions of a direct-injection two-stroke SI engine is lower than that of a standard modern four-stroke SI engine by about 15%. Note that if the two-stroke SI engine is carbureted or intake port-fuel-injected, this benefit essentially disappears due to fuel short-circuiting the cylinder.

15.6.2 Two-Stroke Gasoline SI Engines

The two-stroke cycle spark-ignition engine in its standard form employs sealed crankcase induction and compression of the fresh charge prior to charge transfer, with compression and then spark ignition in the engine cylinder after charge transfer. The fresh mixture must be compressed to above exhaust system pressure, prior to entry to the cylinder, to achieve effective scavenging of the burned gases. Two-stroke cycle scavenging processes were discussed in Sec. 6.6 . The two-stroke gasoline engine is an especially simple and light engine concept and finds its greatest use as a portable power source or on motorcycles where these advantages are important. Its inherent weakness in its standard form is that the fresh fuel-air mixture which short-circuits the cylinder directly to the exhaust system during the scavenging process constitutes a significant fuel consumption penalty and results in excessive unburned hydrocarbon emissions.

This section reviews the performance characteristics of small crankcase compression two-stroke cycle SI engines. The performance characteristics (power and torque) of these engines depend on the extent to which the displaced volume is filled with fresh mixture, that is, the charging efficiency [Eq. (6.27)]. The fuel consumption will depend on both the trapping efficiency [Eq. (6.24)] and the charging efficiency. Figure 15.44 *a* shows how the trapping efficiency η_{tr} varies with increasing delivery ratio Λ at

several engine speeds for a two-cylinder 347-cm^3 displacement motorcycle crankcase compression engine. The delivery ratio increases from about 0.1 at idle conditions to 0.7 to 0.8 at wide-open throttle. Lines of constant charging efficiency η_{ch} [which equals $\Lambda\eta_{tr}$; see Eq. (6.28)]. Figure 15.44 *b* shows bmep plotted against these charging efficiency values and the linear dependence on fresh charge mass retained is clear.

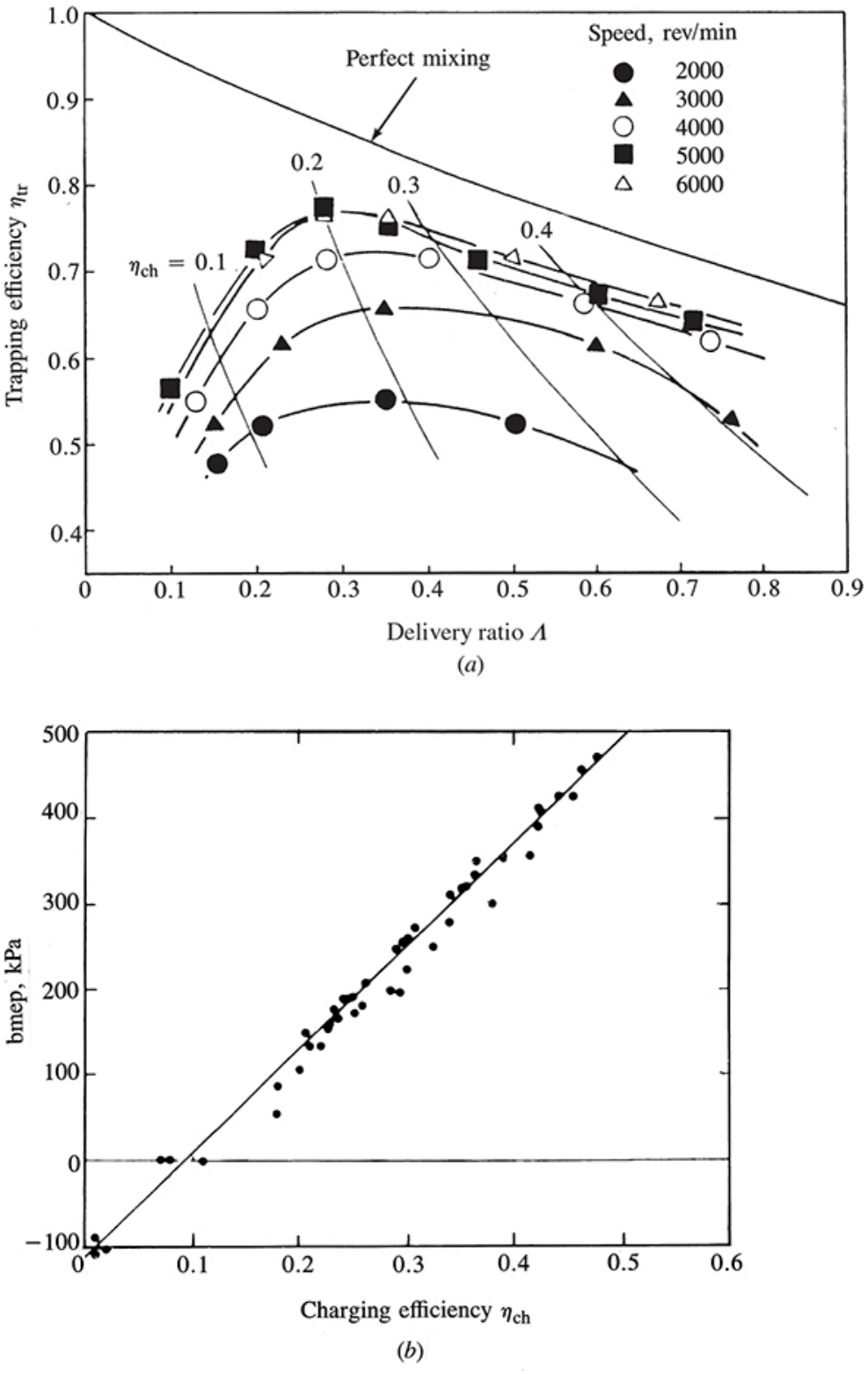


Figure 15.44 (*a*) Trapping and charging efficiencies as a function of the delivery ratio and (*b*) dependence of brake mean effective pressure on fresh-

charge mass, defined by the charging efficiency. Two-cylinder 347-cm ^{3} displacement two-stroke cycle spark-ignition engine. [50]

Performance curves for a three-cylinder 450-cm ^{3} two-stroke cycle minicar engine are shown in Fig. 15.45. Maximum bmep is 640 kPa at about 4000 rev/min. Smaller motorcycle engines can achieve slightly higher maximum bmep at higher speeds (7000 rev/min). Fuel consumption at the maximum bmep point is about 400 g/kW·h. Average fuel consumption is usually one-and-a-half to two times that of an equivalent four-stroke cycle engine.

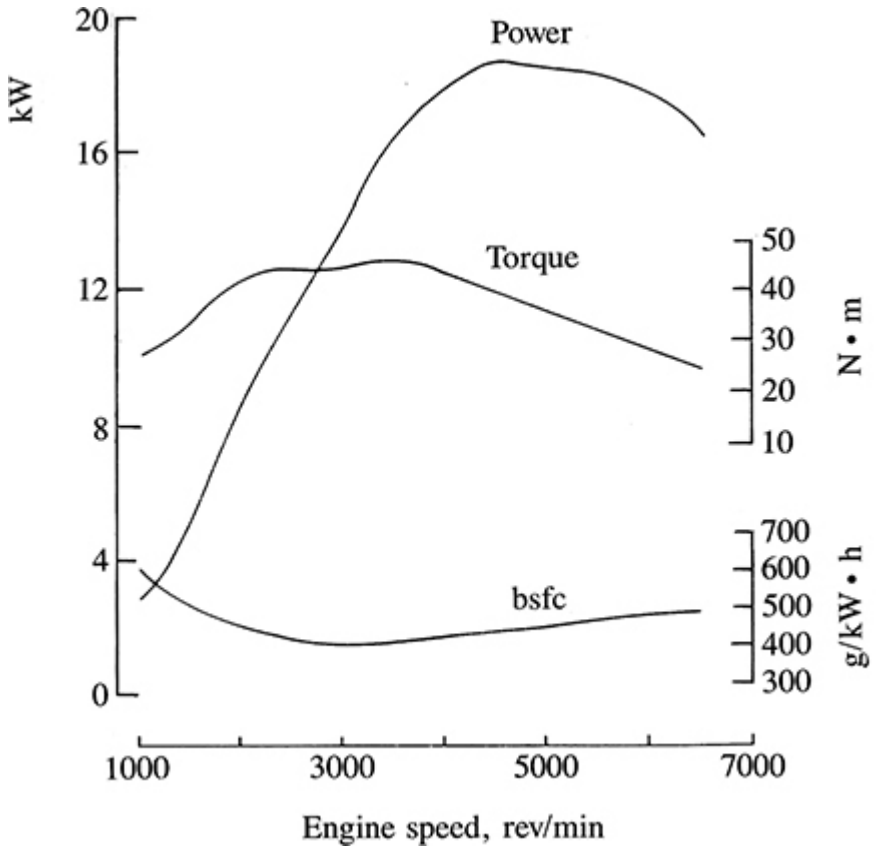


Figure 15.45 Performance characteristics of a three-cylinder 450-cm^3 two-stroke cycle spark-ignition engine. Maximum bmep = 640 kPa. Bore = 58 mm, stroke = 56 mm. [51]

CO emissions from two-stroke cycle engines vary primarily with the fuel/air equivalence ratio in a manner similar to that of four-stroke cycle

engines (see Fig. 11.19). NO_x emissions are significantly lower than from four-stroke engines due to the high residual-gas fraction resulting from the low charging efficiency especially at light to mid loads. Unburned hydrocarbon emissions from carbureted or manifold fuel injected two-stroke engines are about five times as high as those of equivalent four-stroke engines due to fresh mixture short-circuiting the cylinder during scavenging. Exhaust mass hydrocarbon emissions vary approximately as $A(1 - \eta_{tr})\,\phi$, where ϕ is the fuel/air equivalence ratio. [50]

Table 15.6 compares the fuel consumption and emissions from a two-stroke and a four-stroke marine engine designed for similar applications. The fuel consumption and emissions data are weighted averages over different load and speed points. The two-stroke engine is inferior to the four-stroke engine in engine-out CO and HC levels, while superior in NO_x emissions. The first two are in part attributable to the engine lubrication approach used in this type of two-stroke engine: the piston ring and skirt in-cylinder lubricant is mixed with the fuel and introduced through the fuel system. Short-circuiting of the fuel through the exhaust ports during the scavenging process cannot be avoided with such fuel-air premixing, so high HC emissions and fuel consumption are unavoidable.

TABLE 15.6 Comparison between fuel consumption and exhaust gas composition of two- and four-stroke two-cylinder carbureted marine engines [52, 53]

Engine	Two-stroke OMC J10RCSE		Four-stroke Honda BF9.9A	
Rated power (kW)	7.4 at 5000 rev/min		7.4 at 5000 rev/min	
Displacement (cm^3)	216		280	
Fuel/oil ratio	50:1		NA	
Fuel consumption (g/kW·h)	910		550	
CO (g/kW·h)	520	EI = 570	370	EI = 670
CO_2 (g/kW·h)	1300	EI = 1430	1050	EI = 1920
HC (g/kW·h)	240	EI = 260	32	EI = 58
NO_x (g/kW·h)	0.73	EI = 0.80	3.4	EI = 6.2

Because of its simple construction, compactness, high power-to-weight ratio, and low cost, small carbureted two-stroke cycle engines have been widely used for hand held equipment. As illustrated above, its most serious

drawback is its high HC emission and low thermal efficiency due to short-circuiting of fuel to the exhaust port. This type of two-stroke engine, where the fuel and air are premixed really cannot meet the strict emissions limits now imposed.

Alternative two-stroke cycle spark-ignition engine approaches have been explored that largely avoid this major challenge. Several of these have been developed to the prototype stage. Table 15.7 summarizes three different concepts which we will briefly describe. Further details can be found in Ref. 52, along with engine layout drawings.

TABLE 15.7 Three different two-stroke cycle gasoline engine prototypes

	Orbital	Toyota	Honda (AR)
Engine type	DI, two-stroke	DI, two-stroke	DI, two-stroke
Scavenging flow	Cross-scavenged, ports	U-scavenged valves	Schnürle (loop)
Air compression	Crankcase	Supercharged, Roots blower	Crankcase
Fuel system	Air-assist, DI	High press. (10 MPa) liquid fuel, DI	Pneumatic (250 kPa), DI
Combustion	SI premixed flame	SI premixed flame	Active radical combustion
Displacement	1.2 dm^3 three-cylinder	804 cm^3 two-cylinder	402 cm^3 one-cylinder
Bore/stroke	81/78 mm	80/80 mm	80/80 mm
Max. power/speed	66 kW/5500 rev/min	47 kW/4000 rev/min	33 kW/6900 rev/min
Spec. P_b: bmep	55 kW/ℓ: 600 kPa	58 kW/ℓ: 877 kPa	82 kW/ℓ: 713 kPa
Compression ratio		8.5 (geometric)	6.1 (trapped)
$\bar{S}_p$ at P_b max.	14.3 m/s	10.7 m/s	18.4 m/s
Lubrication	Oil spray, dry sump	Wet oil sump	
Exhaust port/valve open/close	(port, w. throttle)	70° BBC/50° ABC (valve)	94° BBC/94° ABC (port, w. throttle)
Intake port/valve open/close		50° BBC/70° ABC (valve)	57° BBC/57° ABC
Fuel consumption	Reduced	Reduced	Much reduced
Emissions: HC	Factor ~ 5 less	Factor ~ 5 less	Much reduced
NO_x	Less	Factor ~ 2 less	Less

Source: Heywood and Sher[52]: 1990s prototype engines

The three engine's listed in Table 15.7 utilize direct fuel injection into the cylinder. This obviously prevents loss of fuel through short circuiting during the scavenging process, an important improvement for both fuel consumption and HC emissions. Each uses a different form of direct fuel injection: air-assist (6 bar air pressure) to realize fine-spray atomization; high-pressure

liquid fuel injection; lower pressure air-driven injection. While the Orbital two-stroke engine has a specific combustion-chamber design, its combustion process is similar to a fast-burn four-stroke SI engine combustion system. The Toyota, uniflow-scavenged valved (two intake and three exhaust valves, per cylinder) engine, is more highly boosted (and achieves the highest bmep at the quoted maximum power condition).

The Honda engine uses a different combustion process than normal spark-ignition flame initiation and propagation. By controlling the amount of hot residual gas trapped within the cylinder at the end of the scavenging process, the temperature of the fuel vapor, air, residual gas can be controlled at a high enough level for the charge to autoignite just after TC, releasing its chemical energy (rapidly, but not too rapidly) through a volume burning rather than flame propagation process. This novel two-stroke cycle combustion mode, first developed in Japan, is termed active-radical combustion (1979). [52, 54] The pressure rise this spontaneous combustion process produced in two-stroke cycle engines can be readily controlled and is highly repeatable, cycle-by-cycle. A valve that controls the effective open area of the exhaust port is a key component in achieving the desired degree of control. This mode of operation works best up to about mid-load. It is a forerunner of the HCCI (homogeneous charge compression ignition) process now being explored and developed for four-stroke cycle engines (see Sec. 10.7.2). Note that these three examples of direct-injection two-stroke cycle gasoline engines are prototype concepts that have not yet been used in mass production.

15.6.3 Two-Stroke Cycle CI Engines

Large marine diesel engines (0.4 to 1 m bore) utilize the two-stroke cycle. These low-speed engines with relatively few cylinders are well suited to marine propulsion since they are able to match the power/speed requirements of ships with simple direct-drive propeller arrangements. These engines are turbocharged to achieve high brake mean effective pressures and specific output. The largest of these engines can achieve brake fuel conversion efficiencies of up to 55%. An example of a large marine two-stroke engine is shown in Fig. 1.36. Over the past few decades, the output per cylinder of such engines has increased by a factor of more than two, and fuel consumption has decreased by 25%. These changes have been achieved by

increasing the maximum firing pressure to about 15 MPa, and by refining critical engine processes such as fuel injection, combustion, supercharging, and scavenging. The uniflow-scavenging process is preferred to loop scavenging since it achieves higher scavenging efficiency at high stroke/bore ratios and allows increases in the expansion ratio.

The performance characteristics of a 580-mm bore Sulzer two-stroke marine diesel engine with a stroke/bore ratio of 2.9 are shown in Fig. 15.46. The solid lines show the standard turbocharged engine characteristics. The rated speed for the engine is 125 rev/min, corresponding to a maximum mean piston speed of 7.2 m/s. The rated bmep is 1.66 MPa. The minimum bsfc is 175 g/kW·h which equals a brake fuel conversion efficiency of 48%. For larger lower-speed engines, the efficiency is higher, above 50%. The dashed lines show how the performance of this engine can be improved by turbo-compounding. A proportion of the engine's exhaust flow, at loads higher than 50%, is diverted from the turbocharger inlet to a separate turbine coupled to the engine power takeoff gear via an epicyclic speed-reduction gear and hydraulic coupling. The additional power recovered in this manner from the engine exhaust flow improves bsfc by 5 or so g/kW·h. At part load, when the full exhaust flow passes through the turbocharger, an efficiency gain is also obtained, due to the higher scavenging pressure (and therefore increased cylinder pressure) obtained with the full exhaust flow.

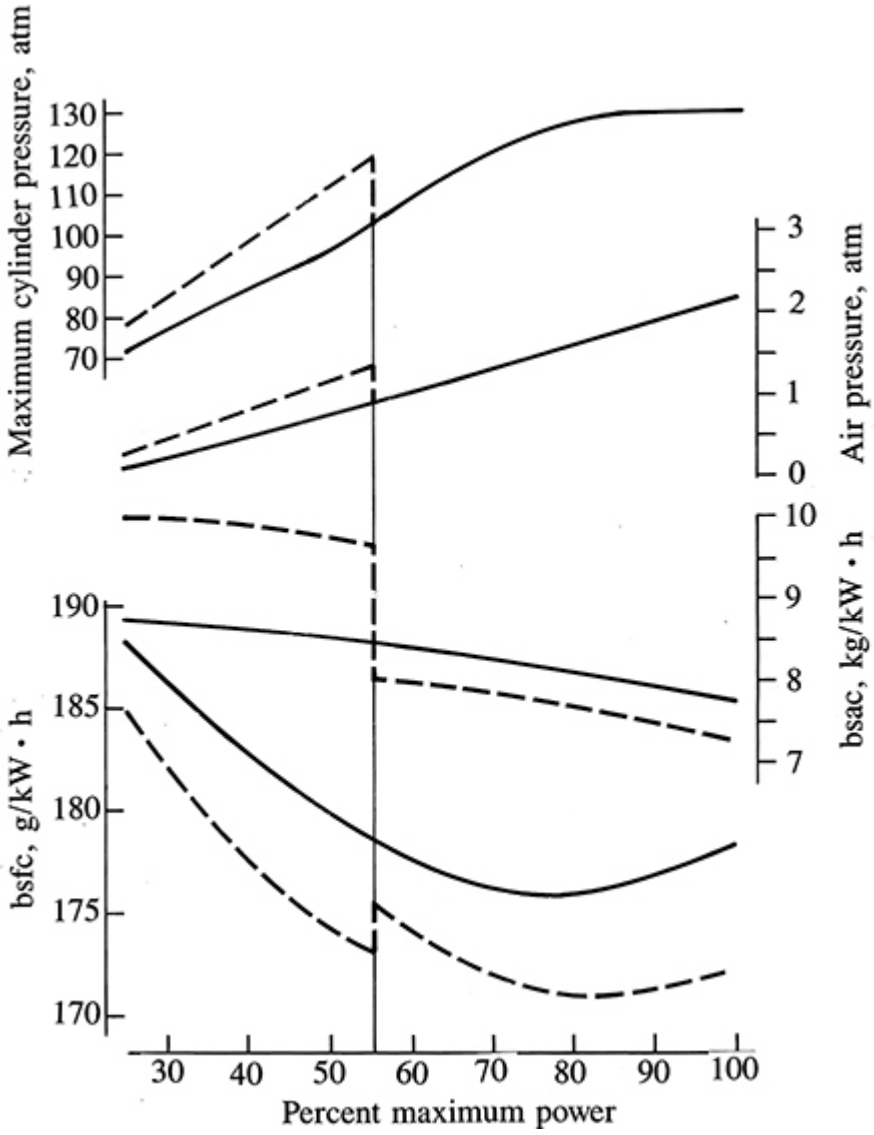


Figure 15.46 Performance characteristics of large marine two-stroke cycle uniflow-scavenged DI diesel engine. Bore = 580 mm, stroke/bore = 2.9, maximum rated speed = 125 rev/min (mean piston speed = 7.2 m/s), bmep (at rated power) = 1.66 MPa. Solid line: standard turbocharged configuration. Dashed lines: parallel turbocompounded configuration at greater than 50% load, bsac: brake-specific air consumption. [47]

The combustion characteristics of large-bore marine two-stroke diesel engines are shown in Fig. 15.47, at several different loads. The heat release rate profiles (bottom graph) lengthen as load increases: at the higher loads, the mixing-controlled diffusion flame phase of combustion becomes longer in

duration and more dominant. Peak cylinder pressure and (average) in-cylinder gas pressure rise as load and overall fuel-air equivalence increases. Because diesels always run lean, *average* peak gas temperatures are substantially lower than burned-gas temperatures in SI engines. The integral of each heat-release-rate profile indicates the amount of fuel injected per cycle, at these different loads.

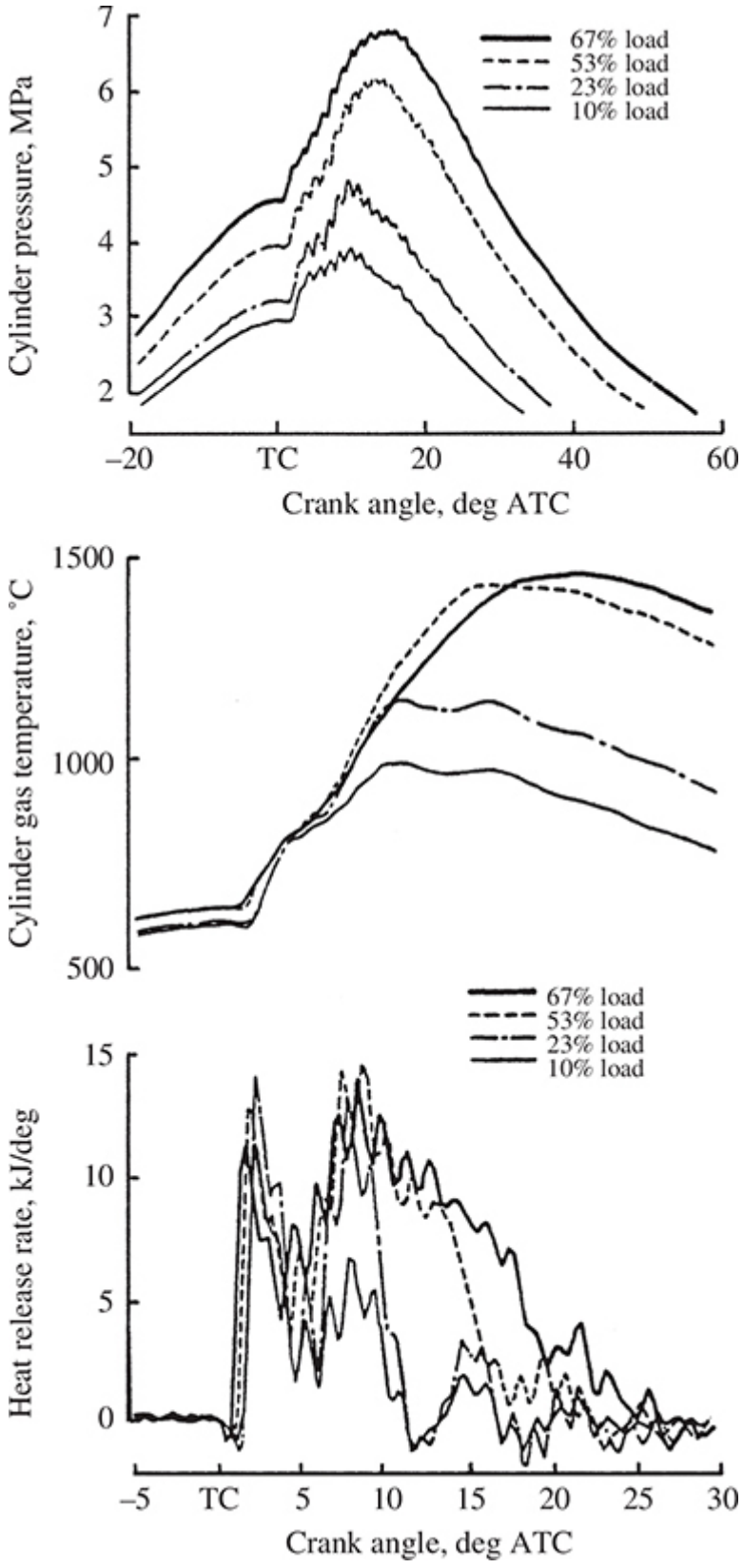

Cylinder pressure, MPa
7
6
5
4
3
2
67% load
53% load
23% load
10% load
–20
TC
20
40
60
Crank angle, deg ATC
Cylinder gas temperature, °C
1500
1000
500
Heat release rate, kJ/deg
15
10
5
0
–5
TC
5
10
15
20
25
30
Crank angle, deg ATC

Figure 15.47 Combustion, characteristics of large-bore turbocharged two-stroke cycle uniflow-scavenged marine diesel engine. Bore = 450 mm, stroke = 750 mm. Maximum rated power is 2.8 MW, maximum bmep = 1.0. [55]

The NO_x emissions from large marine diesels (and when these engines are used for electricity generation) must be controlled. Engine-out NO_x emissions are of special concern. For the same large-bore diesel whose combustion details are shown in Fig. 15.47, Fig. 15.48 shows engine-out NO_x emissions as a function of load at different speeds. The engine here was used to drive the propeller in a large vessel (e.g., oil tanker) application. Here NO_x emission rate has been normalized by fuel flow rate (NO_x emission index is used). In absolute terms, the NO_x emission rate (gNO_x/h) goes up as load increases (as would be expected, ϕ is increasing). Relative to the fuel flow rate as EI_{NO_x}, the NO_x goes down. Relative emissions are more challenging to interpret since, as load increases, the fuel/air equivalence ratio increases, airflow increases, and combustion duration lengthens.

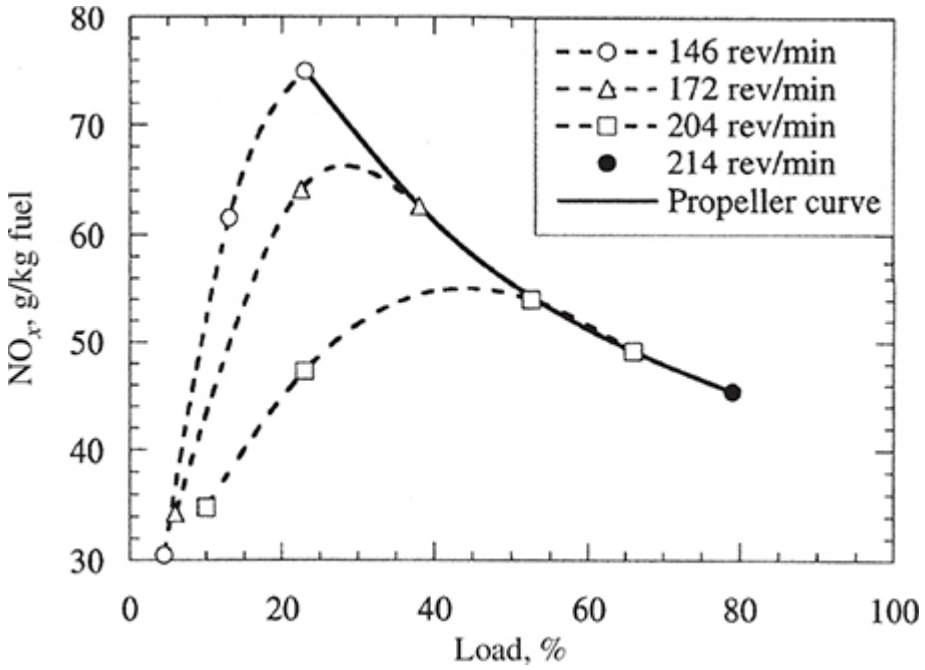
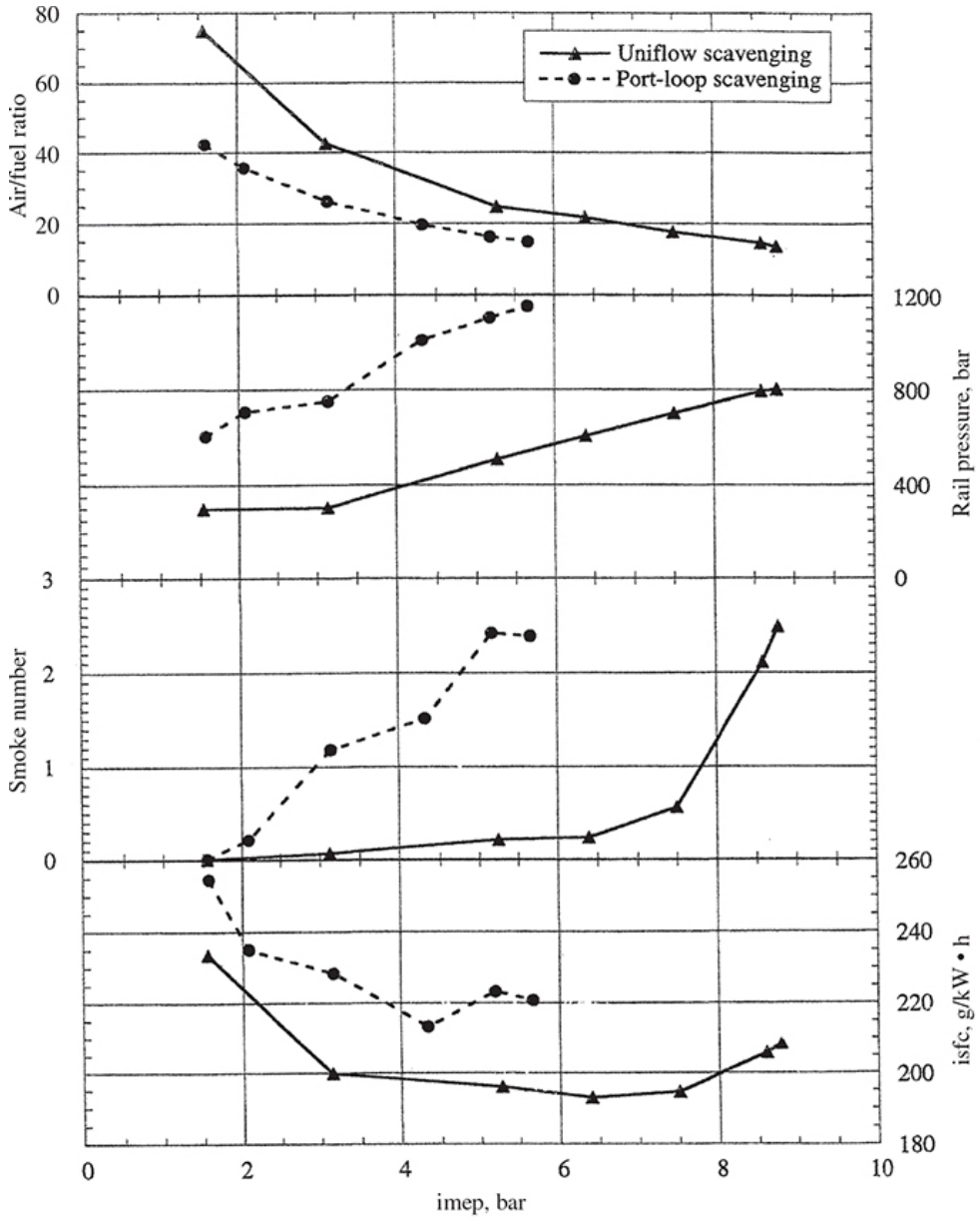


Figure 15.48 Effect of engine load and speed on engine-out NO_x emission index (with ppm in exhaust corrected to 13% oxygen): large-bore two-stroke

marine diesel engine.[55]

Two-stroke diesel engines of smaller size than these large-bore engines are used in several applications: rail, industrial, marine, and oil drilling. The performance and emissions characteristics of two turbocharged two-stroke DI diesel engines, one with a uniflow-scavenged, the other with a port loop-scavenged gas exchange process, with an 80-mm bore, at part load, are shown in Fig. 15.49. Such smaller two-stroke cycle DI diesel engines also compete with four-stroke cycle engines in the marine, industrial, and construction markets. Engine details are given in Table 15.7. The more effective scavenging process inherent in the uniflow scavenging flow design is apparent in the data shown in Fig. 15.49. For both engines, the air/fuel ratio decreases as load increases. The isHC emissions decrease, and the smoke number (SN) and isNO $_x$ emissions both increase. The specific fuel consumption reaches a minimum at about 80% of the maximum load. The increase in isCO emissions when load decreases below 50% of its maximum value indicates that poor mixing is occurring within parts of the cylinder. With the port loop-scavenged engine, the SN exceeds a value of 1.0 at 300 kPA imep, and reaches the full load limit of 2.5 at about 500 kPa. With the uniflow-scavenged engine, the values of smoke number are significantly lower. As a consequence, lower fuel injection pressures can be utilized, and still achieve adequate fuel-air mixing rates. The fuel consumption for the ported loop-scavenged engine is 10% higher (in part due to its less favorable surface-to-volume ratio). The isNO $_x$ emission levels are closely comparable.


Uniflow scavenging
Port-loop scavenging
Air/fuel ratio
80
60
40
20
0
Rail pressure, bar
1200
800
400
0
Smoke number
3
2
1
0
isfc, g/kW • h
260
240
220
200
180
0
2
4
6
8
10
imep, bar

(*a*)

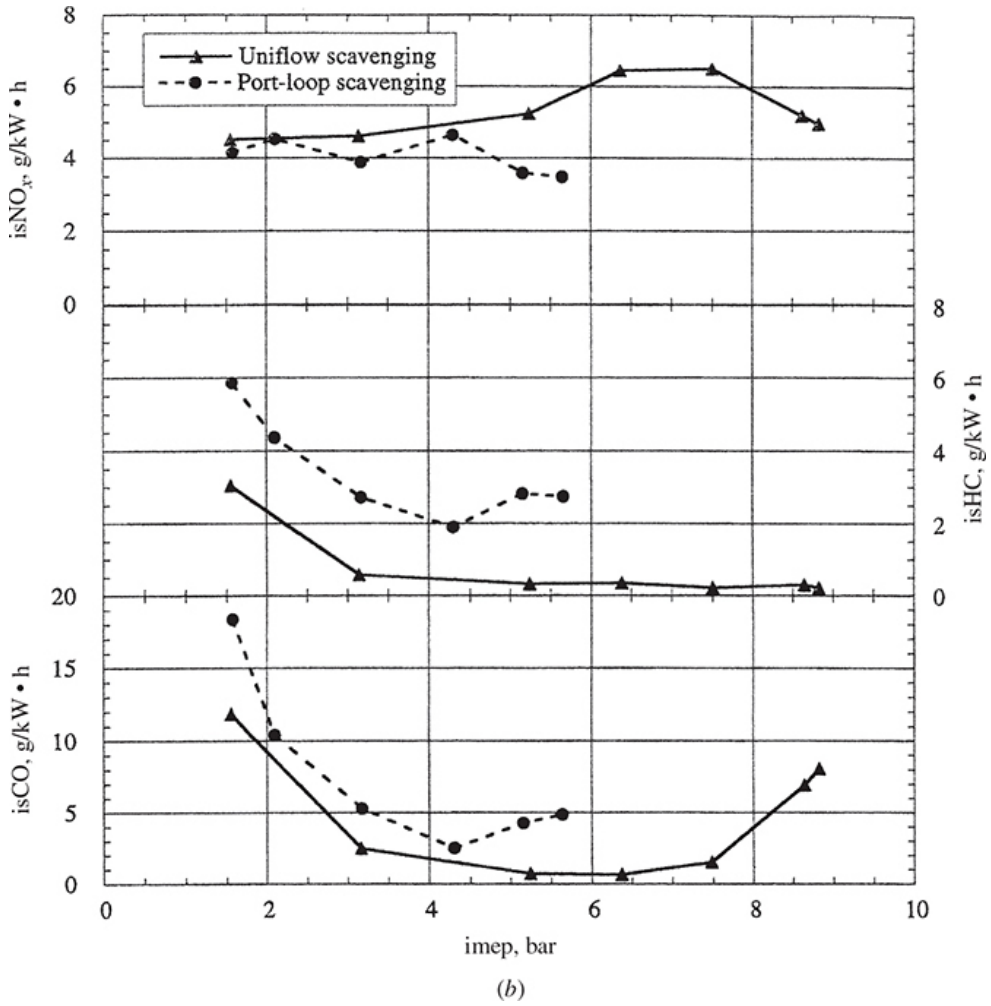


Figure 15.49 (*a*) Part-load operating characteristics and (*b*) engine-out emissions characteristics, for a small-bore loop-scavenged and a uniflow-scavenged two-stroke diesel engine.[n] Engine details in Table 15.8.[52, 56]

TABLE 15.8 Daimler-Benz direct-injected two-stroke diesel engine comparison [52, 56]

Engine type	Ported loop scavenging	Uniflow scavenging
Bore/stroke	80/70 mm	80/98 mm
Displacement, one cylinder	350 cm^3	493 cm^3
Combustion chamber	Bowl in cylinder head	Bowl in piston
Scavenging air	Crankcase compressed	External compressor

15.7 NOISE, VIBRATION, AND HARSHNESS

Noise, vibration, and harshness (NVH) is the phrase used to describe the audible and tactile disturbances that engines generate which are experienced by the operator, user, or bystander. In probably the most important application area, engines in vehicles, the engine, the source of the vibrational energy that creates these disturbances, couples with the vehicle structure leading to additional disturbances that vehicle occupants experience negatively. The mechanisms that generate these disturbances, and approaches that reduce their impacts, are the subject of this section. *Noise* is discussed in Sec. 15.7.1 . The *dynamics* of the reciprocating engine's power transmitting components—pistons, connecting rods, and crankshaft, and the forces involved, are covered next (in Sec. 15.7.2), followed by a review of *balancing* issues in multi-cylinder engines (Sec. 15.7.3). Section 1.5 (Multi-cylinder Engines), Sec. 2.2 (Geometrical Relationships for Reciprocating Engines), and Sec. 2.3 (Forces in the Reciprocating Mechanism) provide important background material for these topics.

Harshness refers primarily to the audible impact of an overly fast engine combustion process. The rapid cylinder-pressure rise rate that this generates is "noisy" and unattractive to engine operators. It, therefore, needs to be controlled. Factors that control burn rates in spark-ignition engines (Chap. 9), and diesel engines (Chap. 10) have already been discussed. In gasoline engines, combustion system design (Sec. 15.4.6) embodies achieving "fast-enough" combustion at light to mid-load to achieve the desired efficiency and emissions benefits. However, the burn rate increases as engine load is increased (as the throttle is opened), so combustion harshness as wide-open-throttle operation is approached, may be a problem. Diesel engines inherently have a different burning profile (see Fig. 15.1), with a rapid initial burn rate and much slower later burn rate, with corresponding rates of pressure rise. This challenge becomes worse when the engine is cold, and the longer ignition delays result in more rapid initial burn rates still. Pre- or pilot injection helps here by reducing the duration of the delay, and thus the initial rate of pressure rise.

The overly rapid rise in pressure excites strong vibrations of the cylinder liner and head walls, thus radiating audible sound waves to the engine's surroundings.

15.7.1 Engine Noise

Noise is often defined as "unwanted sound." It is a significant problem with internal combustion engines, especially with diesel engines and small two-stroke gasoline engines. It is a difficult problem to resolve because effective noise suppression usually reduces engine power, increases engine bulk size and weight, and increases cost, all of which run counter to the competitive pressures in most engine markets. Noise is an extensive and well-defined field, with its own vocabulary and literature, which can separate it from more obvious aspects of engine operation. Increasingly, however, as noise limits on engines become more stringent, the reduction of engine noise must be integrated into a systems approach to engine design. While an extensive treatment of engine noise is beyond the scope of this book, in this section we develop enough of the fundamentals of the subject for the reader to be able to understand the sources of noise in internal combustion engines and the means available to reduce them. Additional details and more extensive reviews can be found in Annand and Roe, [57] Blair, [58] and the Bosch Automotive Handbook. [59]

Fundamentals and Definitions

Sound propagates through a medium as fluctuations in pressure. The frequency of these fluctuations determines the *pitch* of the sound. The human ear can detect frequencies from 20 Hz up to 15,000 to 20,000 Hz. A simple sinusoidal pressure oscillation produces a sound wave that is called a *pure tone*. Usually sound is composed of a fundamental frequency with smaller amplitude harmonics—simple multiples of this fundamental frequency. This mixture of frequencies gives the sound its "tone"; significant amplitude high-frequency components make sounds especially objectionable.

The pressure fluctuations generated by the noise source transfer energy outward through molecular collisions. The more energy transferred, the higher the sound level or volume. The *intensity of sound*, I (W/m^2), is defined as the energy transfer per unit time by the sound wave through unit area perpendicular to the wave propagation direction. It quantifies the actual energy flux associated with the sound wave. Usually, the effective sound pressure—the root-mean-square (rms) fluctuating pressure δp—is what is measured. For plane or spherical waves,

$$I = (\delta p)^2/(\rho a) \tag{15.14}$$

where ρ is the medium density [for ambient air, ρ = 1.184 kg/m^3 (0.0739 lbm/ft^3)] and a is the sound speed [given by $(\gamma RT)^{1/2}$, 346 m/s for air at 25°C]. With a localized noise source, the sound intensity I and the square of the rms fluctuating pressure $(\delta p)^2$ both fall off as the square of the distance from the source.

The range of sound intensities that humans can hear and tolerate is very large, from 10^{-12} W/m^2 to 1 W/m^2, so a logarithmic scale is used. The *sound intensity level* L_I is therefore defined as

$$L_I = 10 \log_{10}(I/I_0) \tag{15.15}$$

where I_0 is a chosen reference level. The unit of this relative intensity is the decibel (dB). Equation (15.15) can be expressed in terms of the sound pressure level L_p,

$$L_p = 10 \log_{10}[(\delta p)^2/(\delta p_o)^2] = 20 \log_{10}(\delta p/\delta p_o) \text{ dB} \tag{15.16}$$

where δp_0 is the reference sound pressure level. The standard reference level corresponds to the threshold of hearing, where $I_0 = 10^{-12}$ W/m^2 and $\delta p_0 = 5 \times 10^{-5}$ Pa. Using Eqs. (15.14) to (15.16), a sound intensity of 1 W/m^2, which is the sound level at which the ear begins to feel pain, corresponds to a mean effective sound pressure of 20 Pa, and a sound pressure level of 120 dB.[52]

In the frequency range 500 to 5000 Hz, the ear responds almost logarithmically to sound intensity, so this logarithmic decibel scale is appropriate. However, sounds of different pitch or frequency perceived as equally loud do not have the same intensity. To provide a quantitative measure of subjective *loudness*, sound is said to have a loudness in *phons* equal to the intensity level of a pure tone of frequency 1 kHz that appears equally loud. Figure 15.50 shows how the actual intensity required to produce a certain sensation of loudness for an average listener varies with frequency. The weighting shown is known as the *A-weighting*, and corresponds to the ear's response at 40 phon. An A-weighting filter is

normally applied automatically to noise-meter (a microphone that converts variations in pressure to variations in voltage) measurements and is designated dB(A). Note that the most irritating frequencies are between about 1 and 5 kHz, with increasingly higher sound pressure levels sounding equally loud as frequencies decrease and increase below and above this band, respectively. [52]

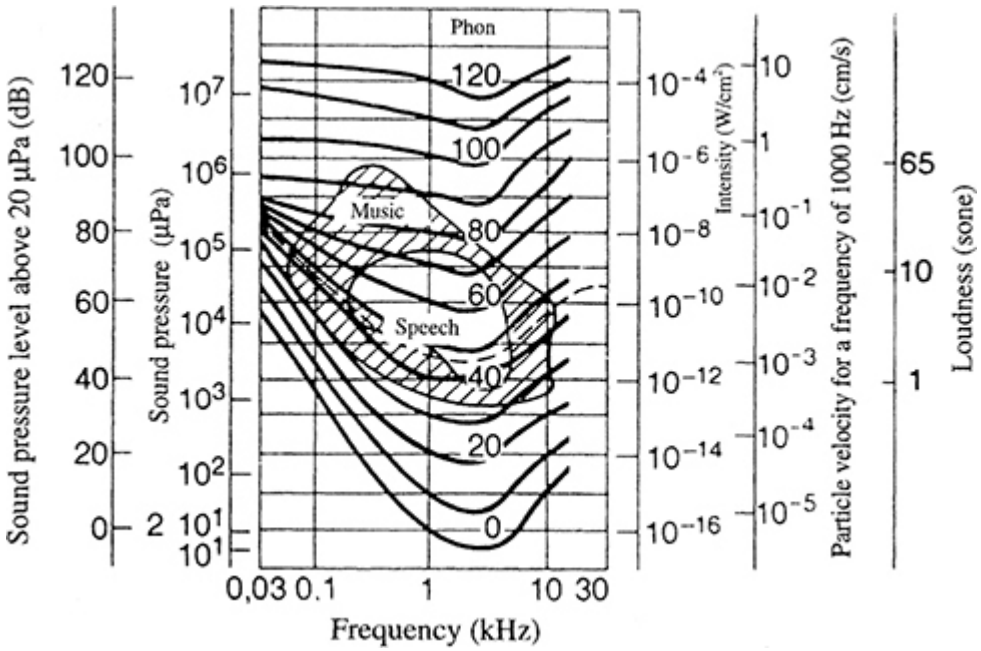


Figure 15.50 Objective (sound pressure level, in decibels) and subjective (A-weighted sound-meter curve, in phons) scales for noise curves of equal loudness level, as a function of frequency.[52, 59] (*Courtesy Robert Bosch GmbH and SAE.*)

With more than one source of sound, in nonreflecting surroundings and sufficiently distant from the sources, the combined effect is obtained by adding the absolute source intensities (the energy fluxes per unit area), as they are each planar sound waves. For example, with two sources of intensity I_1 and I_2, the combined intensity I_{12} is

$$I_{12} = I_1 + I_2 \tag{15.17}$$

where, from Eq. (15.15) ,

$$I_1 = I_0 \text{ antilog}_{10}(L_{I1}/10) \text{ and } I_2 = I_0 \text{ antilog}_{10}(L_{I2}/10)$$

The resultant sound intensity level is given by

$$\begin{aligned} L_{I12} &= 10 \log_{10}(I_{12}/I_0) \\ &= 10 \log_{10}[\text{antilog}_{10}(L_{I1}/10) + \text{antilog}_{1}(L_{I2}/10)] \end{aligned} \tag{15.18}$$

If the two sources (1 and 2) are equal, Eq. (15.18) gives

$$\begin{aligned} L_{12} &= 10 \log_{10}[2 \text{ antilog}_{10}(L_{I1}/10)] \\ &= 10[\log_{10} 2 + (L_{I1}/10)] = L_{I1} + 3.01 \text{ dB} \end{aligned}$$

that is, doubling the intensity of sound increases the sound intensity level by 3 dB. If one source is substantially larger than the other, for example, the two sources are 100 dB and 90 dB, then the two sources together give

$$L_{I12} = 10 \log_{10}[\text{antilog}_{10} 9] = 100.4 \text{ dB}$$

Here, the smaller source contributes little to the combined sound level. [52]

Two important consequences for reducing noise levels in engines follow from this review of noise fundamentals: [58]

- The loudest of the several sources of noise in an engine will be much the most important in determining the total sound pressure level. To reduce the noise emissions, that source must first be dealt with effectively.
- The higher frequency range (1 to 5 kHz) is much more objectionable to the human ear, at the same sound intensity level, than are lower frequencies. So treatment should focus first on this higher frequency range.
- When a major noise source varies with time and produces short intense pulses (e.g., a poorly muffled engine exhaust system at low speed), these are highly disagreeable and must be heavily damped by the noise treatment system.[57]

Sources of Engine Noise

There are three major sources of engine noise: the exhaust, the intake, and mechanical impacts and vibration. Exhaust noise is easiest to understand; its origin is the pressure waves set up in the exhaust system by each cylinder's exhaust blowdown process, and audible noise is generated as these waves enter the atmosphere. Intake noise is generated by the pressure waves set up by the periodic induction processes into each cylinder. Mechanically generated noise is generally lower than exhaust- and intake-generated noise. There are several sources. One is the rapid combustion-produced pressure rise that is transmitted through the block and cylinder-head walls to the atmosphere (often referred to as *harshness*). Another source is when the piston *slaps* against the cylinder liner, causing the block to vibrate. At close to TC piston position, the forces acting on the piston cause it to rock from one side of the liner to the other causing the block to vibrate. Mechanical noise also originates from the oscillating forces on the crankshaft bearings. The roller bearings sometimes used in the crankcase and valvetrain are noisier than standard lubricated plain bearings where the oil film damps out the mechanical vibration. In air-cooled engines, the cylinder and head cooling fins vibrate and produce noise. Liquid cooling of the engine damps out some of the vibrations the block and head structure would otherwise transmit. Gears and chain drives are also significant sources of mechanical noise. [57, 58]

In boosted engines, an additional noise source is the turbocharger. Constant-pressure turbocharger systems, where a gas reservoir upstream of the turbine is used to dampen the exhaust blow-down pressure pulses to improve overall turbine performance, are significantly quieter than the alternative impulse systems where the turbine is designed to take advantage of the exhaust pressure pulsations. In diesel engines, the high-pressure fuel-injection system is also a noise source. [52]

We now show some examples of engine noise levels and spectra. Noise level meters provide such information in several ways. One is to weight the sound pressure levels at different frequencies and then integrate to produce an overall sound intensity reading. The A-weighting mentioned earlier, which corresponds to the average ear's response at the 40-phon level, is usually used in engine and vehicle noise tests. However, knowledge of the noise spectral distribution is helpful when designing noise suppression systems.

Figure 15.51 compares the noise spectrums of equivalent four-stroke cycle and two-stroke cycle diesel engines. The sound pressure levels are

highest in the 1 to 3 kHz range and decline significantly at higher and lower frequencies. The maximum noise levels are about 8 dB(A) lower for the two-stroke engine due to its shorter ignition delay (and hence slower initial combustion-generated rate of pressure rise), and the lower two-stroke engine peak pressures. The two-times-higher firing frequency of the two-stroke, at the same engine speed, also reduces mechanical vibration substantially, and the mechanical noise it generates. [60]

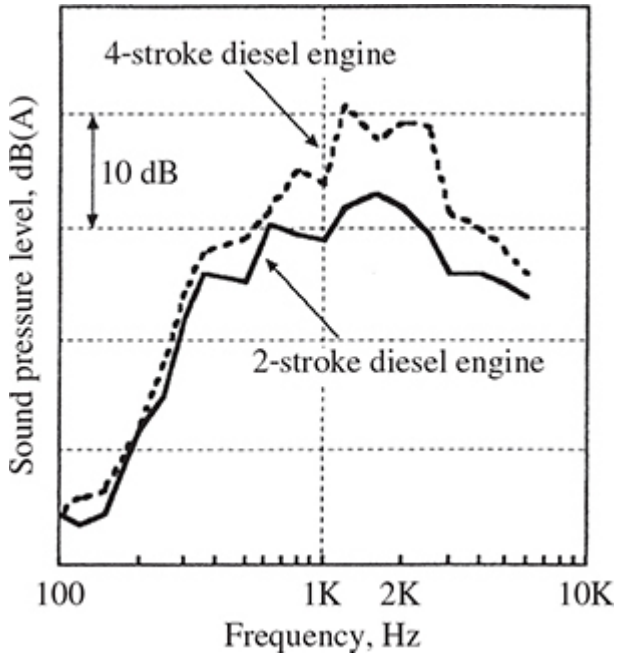


Figure 15.51 Engine noise spectra in dB(A) (the A-weighting corresponds to the ear's response at 40 phon) from two automotive diesel engines. Lower curve: 2.5-liter four-cylinder two-stroke supercharged CI engine producing 320 N·m of torque at 1800 rev/min and 100 kW at 3400 rev/min; bore/stroke 96/86 mm, r_c = 18. Upper curve: equivalent four-stroke cycle diesel engine. [60]

Figure 15.52 shows the noise level in dB(A) for a 4.5-liter V-8 naturally-aspirated gasoline engine, incorporating several noise reducing features, as a function of engine speed. Noise levels rise by some 30 dB(A) from low speed to high speed, which is typical for gasoline engines. [61]

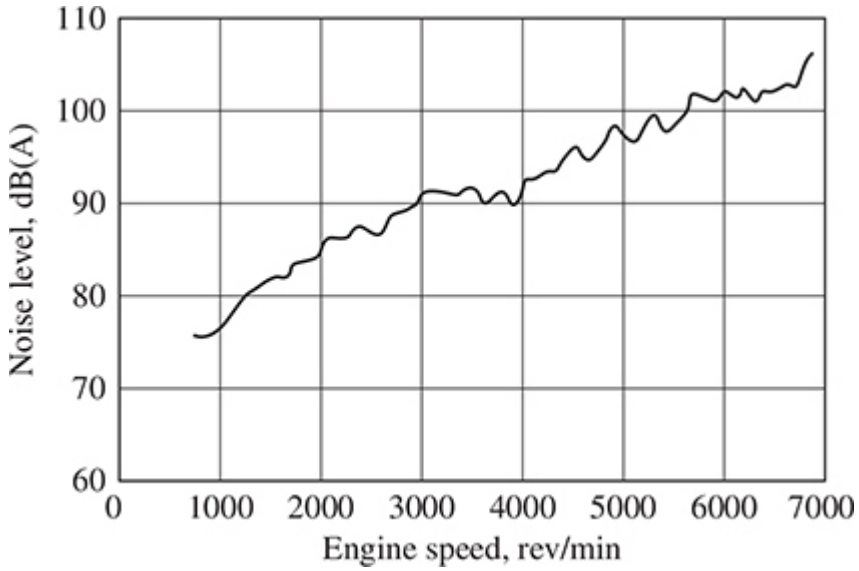


Figure 15.52 Noise level in dB(A) at wide-open-throttle as a function of engine speed, for a 4.5-liter V-8 naturally-aspirated gasoline engine.[61]

Engine Noise Reduction

Ever increasing specific power, and changing requirements such as reduced vehicle weight, are demanding reductions in engine size and weight, and in fuel consumption. Increasing in-cylinder pressure levels and reducing engine weight, make achieving reductions in noise and vibration more important. The following are examples of engine enhancements that improve NVH characteristics.

Stiffer block structures reduce vibration and sound radiation. A more rigid crankcase structure, larger crankshaft diameter with more rigid end bearings with a thinner oil film, reduce and damp crankshaft vibration. Lower noise-generating lighter smaller-pitch chain drives have been developed with smoother link and sprocket contract, with chain-driven accessories more rigidly mounted on the block, reduce noise generation. Soft mounted and stiffer oil pan and chain drive covers more effectively absorb the noise produced.

Simple two-stroke cycle gasoline engines are designed to be light and compact, so the engine surfaces radiate noise more easily, and the space available for noise suppression devices is limited. These small two-stroke engines usually employ a tuned exhaust, with a choked final outlet to maximize the power output. This exhaust system feature eases the task of

designing a silencer-muffler with minimum impact on power. Also, due to its less effective filling of each cylinder with air, the two-stroke SI engine has lower peak cylinder pressures than the four-stroke. The resulting oscillating forces in major engine components are therefore lower and so is the mechanical noise they generate.

With diesel engines, these same opportunities are available. In addition, since the diesel's combustion process follows from the fuel injection strategy and rate (and in-cylinder air temperature and pressure at time of injection), opportunities exist to reduce the rapid rate of pressure rise that results in combustion harshness. Also, pre (or pilot) injection moderates the initial rapid-burning phase of diesel combustion and reduces its initial pressure rise rate. See Secs. 10.5.4 and 10.7.1. The attendant noise reductions are illustrated in Fig. 15.53 for a 19-liter V-8 turbocharged and intercooled diesel. The impact of pilot injection is greatest at below 50% load: also, in the frequency range 600 to 3000 Hz. Greatest benefits were realized with a small quantity of pilot-fuel (2 to 6 mm ^{3}/injection) injected some 15 to 20 degrees before the main injection was initiated (main injection, 120 to 220 mm ^{3}/injection). [62]

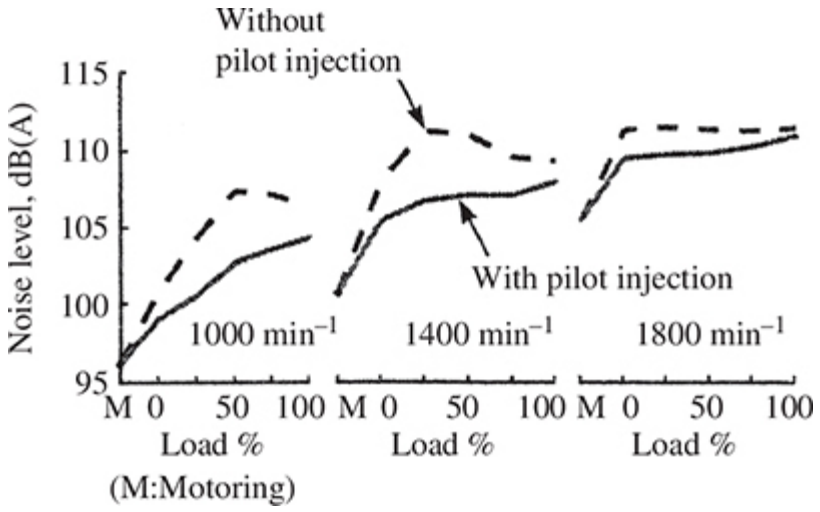


Figure 15.53 Noise reduction, dB(A), resulting from use of pilot injection in a 19-liter V-8 turbocharged DI diesel engine. Pilot fuel injection amount 2 to 6 mm ^{3}/injection; main injection amount 120 to 220 mm ^{3}/injection. [62]

In the large marine diesel application area where two-stroke cycle engines are often used, the major noise challenge is overall engine-room

noise, which must be kept below 100 dB(A), measured at several locations at 2 to 3 m heights about 1 m from the engine's outer surface. The noise is primarily generated by individual engine components and their surfaces. Exhaust gas noise can be brought down by about 25 dB(A) to acceptable levels with large absorption silencers, which are especially effective at damping the higher frequencies.

Exhaust Noise

Exhaust noise is the result of pressure waves in the exhaust system exiting through its open end into the ambient air, and from vibration of the exhaust system surfaces caused by these pressure waves. Various muffler (silencer) technologies and exhaust system configurations have been developed to reduce exhaust noise. The selection of the appropriate approach, with minimum impact on engine power levels, and its integration into the total exhaust system depend on the engine application (and space and weight constraints).

Mufflers take many forms, and more than one type may be used in a given exhaust system. Two basic principles are involved, absorption and reflection, and mufflers are divided into two types according to which principle is involved (although both absorption and reflection usually occur). Mufflers, together with the exhaust system, form an oscillator with natural resonance frequencies, so the location of the mufflers in the exhaust is critical. To avoid structure-borne noise (and provide thermal insulation) mufflers often have double walls and an insulating outer layer. [59]

Absorption mufflers (see Fig. 15.54) are constructed with one chamber filled with sound-deadening material through which the perforated exhaust pipe is passed. The absorbing material is usually long-fiber mineral wool with a bulk density of 120 to 150 g/liter. The level of muffling depends on the bulk density and sound-absorbing characteristics of the absorbing material, the muffler length, and material thickness. Damping takes place across a wide frequency band but only starts at higher frequencies (above about 1 kHz). Absorption mufflers are normally used as rear mufflers: they are the most frequently used design in large two-stroke diesels. [63]

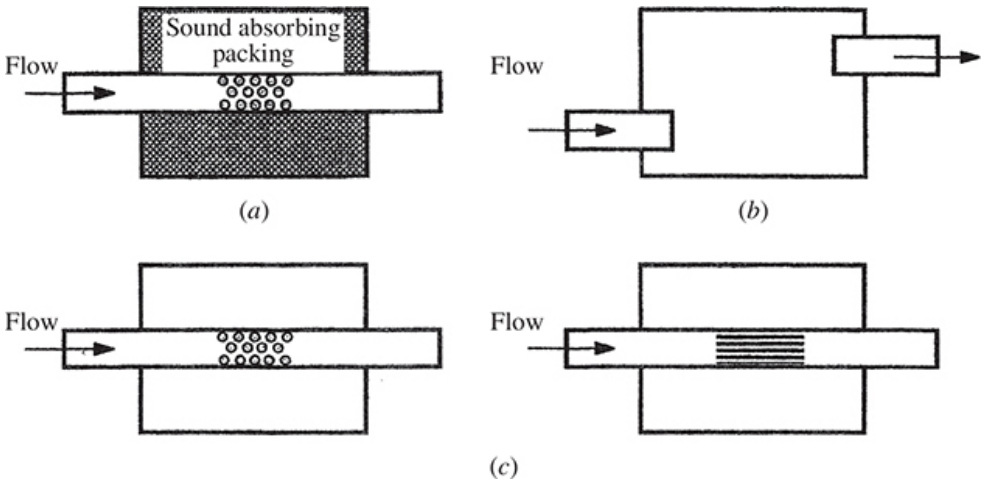


Figure 15.54 (*a*) Absorption muffler element; (*b*) diffusing muffler element; (*c*) side-resonator muffler element.[57]

Reflection mufflers consist of chambers of varying dimensions connected together by pipes. The differences in cross sections of the pipes and the chambers, the dispersion of the exhaust gas flow, and the resonators formed by connecting pipes to chambers, all reduce the fraction of the incident pressure fluctuations transmitted. Diffusing mufflers or silencers (see Fig. 15.54*b*) absorb noise at frequencies other than those at which the expansion box will resonate. Side resonators (Fig.15.54*c*), where the flow connects with a separate closed chamber through holes or slits in the pipe wall, absorb noise at a specific frequency, such as the fundamental exhaust pulse frequency. The more chambers used, the more effective the muffler. [52]

Such reflection mufflers remove annoying components of the noise emitted by the exhaust through several different principles. The operation of the expansion box muffler in Fig. 15.54*b* can be explained as follows. The performance of an isolated muffler or silencer is often expressed as a *transmission loss*. Provided the relative amplitude of the pressure fluctuations is not too large, and the sound source is not affected by reflections from the muffler, the sound intensity at a given frequency is reduced by a fixed fraction. Thus, the intensity level is reduced by a fixed number of decibels. This reduction in decibels is the transmission loss. The transmission loss of a single pass through expansion box is a function only of the expansion area ratio $m = A_2/A_1$, and the ratio of the box length to the wavelength λ of the sound wave. The transmission loss T_{loss} is given by [57]

$$T_{\text{loss}} = 10 \log_{10}\left[1+\frac{1}{4}\left(m-\frac{1}{m}\right)^2 \sin^2\left(\frac{2\pi L}{\lambda}\right)\right] \text{dB} \tag{15.19}$$

The box resonates whenever an integral number of half wavelengths just fits the length L [i.e., where $\lambda = 2L, L, 2L/3, L/2$, etc., at frequencies of $f = a(2L), a/L, 3a/(2L), 2a/L$, etc.]: the transmission loss then goes to zero. Between, it rises to a maximum value of

$$T_{\text{loss,max}} = 10 \log_{10}\left[\frac{1}{4}\left(m+\frac{1}{m}\right)^2\right] \text{dB} \tag{15.20}$$

Experimental measurements show good agreement with theory until frequencies are reached where the box can resonate in a transverse or radial mode (see Fig. 15.55). For a box with circular cross-section, the first radial resonance occurs, as shown, at a frequency of $1.22\,a/D$. [57]

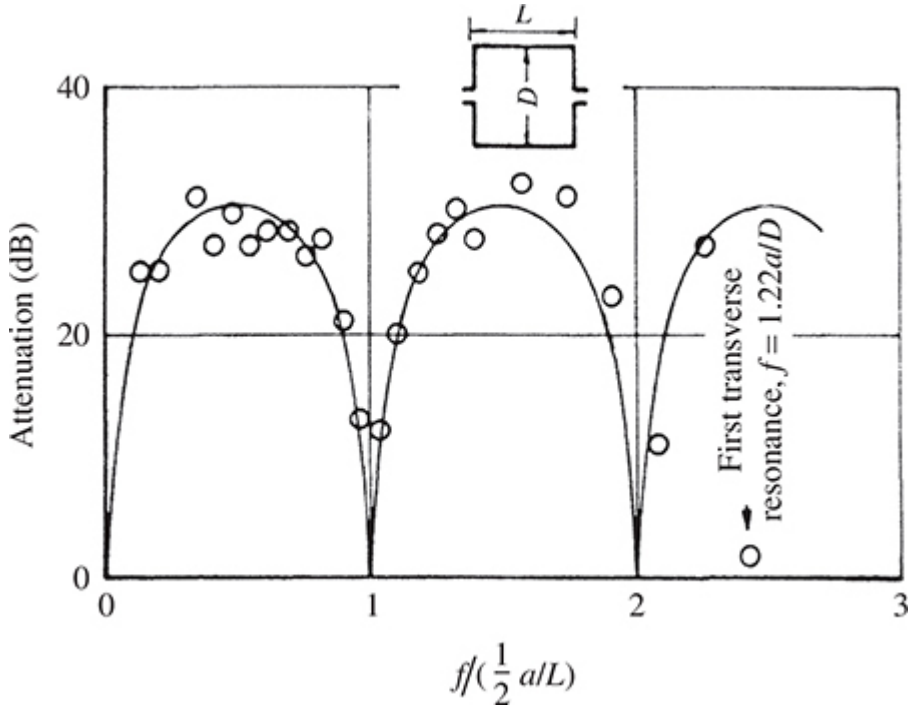


Figure 15.55 Transmission loss (attenuation) of an expansion box of circular cross-section showing pass bands at the first two longitudinal resonance frequencies, and the first transverse resonance.[57]

Pipes perforated with holes or slits connected to boxes, as used in side-resonator mufflers (see Fig. 15.54*c*), absorb strongly at the natural frequency of the box or side cavity. The transmission loss of a side-resonator element of volume V connected to a pipe of cross-sectional area A_p through an opening of conductivity K is given by

$$T_{\text{loss}} = 10 \log_{10}\left[1 + KV \Big/ \left\{2A_p\left(\frac{f}{f_n} - \frac{f_n}{f}\right)\right\}^2\right] \text{dB} \tag{15.21}$$

The conductivity of the opening (which has the dimensions of length) is given approximately by

$$K = \pi r^2/[l + (\pi/2)r] \tag{15.22}$$

where r is the radius of the opening and l is the length of the pipe leading to the cavity (zero for a simple hole, when $K = 2\,r$). The resonant frequency f_n is

$$f_n = \frac{a}{2\pi}\sqrt{\frac{K}{V}} \tag{15.23}$$

The transmission loss is very high at and near the resonance frequency of the cavity. Away from this frequency, the loss depends on $\sqrt{KV}/A_p$, as shown in Fig.15.56. With high values of $\sqrt{KV}/A_p$, substantial transmission loss is achieved over a wide frequency range. [57]

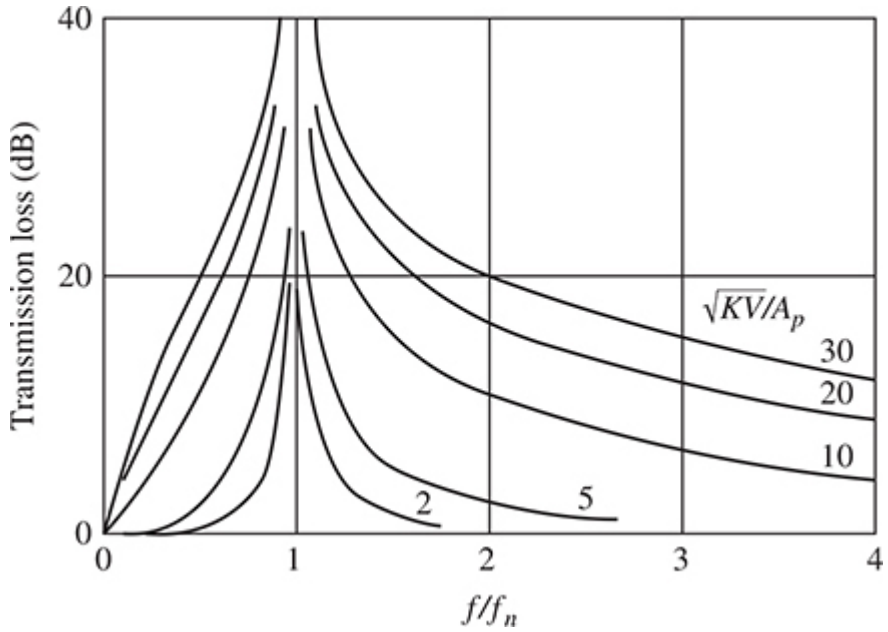


Figure 15.56 Transmission loss of a simple side-resonator muffler element.[57]

Where there are several holes, the conductivity is the sum of the individual values. For a rectangular slit of length l and width b, in a sheet of thickness t, K is

$$K = bl/(t + 0.92k\sqrt{bl}) \tag{15.24}$$

where k decreases from 1.0 for $l = b$ to 0.6 at $l = 20\ b$. The attraction of side-cavity resonators is that the flow through the main pipe is virtually unimpeded. [57]

Practical mufflers are often combinations of more than one expansion box and/or side resonators so their operation is more complex than these simpler illustrations. The preceding transmission loss expressions are for a reflection-free muffler outlet. The open end of the tailpipe is not reflection free; an exiting pressure wave is reflected at an open-ended pipe as an equally strong rarefaction wave. Muffler location in the exhaust system is also important. The exhaust pipe upstream of the muffler can be thought of as an organ pipe with one end (the engine end) closed and one end open; the exhaust pipe downstream of the muffler has both ends open. Their resonant behavior will be different as a result. The design of exhaust systems,

including their noise reducing features, is now done with unsteady one-dimensional gas-dynamic computer codes (see Sec. 14.3.4) as well as based on extensive empirical data.[52]

Intake Noise

Relative to exhaust noise, intake noise is surprisingly high, especially near wide-open throttle. Intake noise originates from the pulsating intake flow corresponding to individual cylinder filling events. Intake noise reduction is challenging because any flow restriction will reduce engine output. The basic induction pulse frequency is

$$f = n_c N(\text{rev/min})/60 \tag{15.25}$$

where n_C is the number of cylinders. Intake noise reduction on small engines is usually accomplished with an expansion box silencer, which incorporates the air filter in its downstream end. The expansion box or cavity has a resonant frequency given by Eq. (15.23) where, because the entry duct is long in relation to its radius, the conductivity K is A_p/l [see Eq. (15.22)]. Intake flow losses limit the minimum value of A_p. The desired deep-toned sound requires the effective attenuation of the higher frequencies. This requires a low natural frequency for the resonator (where no attenuation will occur), requiring a high value for $V/K = lV;$ it also requires a high value for $\sqrt{KV}/A_p$ [see Eq. (15.21)] or a large expansion-box cavity volume. In practice, the natural frequency is set at or below the frequency corresponding to the lowest engine speed at which high loads are produced, determined using Eq. (15.25) ; and using a box volume V between 10 and 20 $V_d\sqrt{n_c}$, where V_d is the displaced volume of each cylinder. The intake noise damping that such a system produces is zero at the resonant frequency of the cavity, and rises steadily to about 15 to 20 dB as frequency increases, before additional wave phenomena effectively limit further improvements. [59]

15.7.2 Reciprocating Mechanism Dynamics

Aspects of the motion of the internal combustion engine's reciprocating mechanism impact NVH and especially *vibration*. This linkage transmits the net pressure force acting on the piston, as it moves up and down the cylinder,

through the oscillating motion of the connecting rod, to the rotating crankshaft (see Fig. 2.1). As explained in Sec. 2.3 , the gas pressure forces (along with the other forces) acting on the piston generate a turning moment about the crankshaft axis (the torque), as indicated in Fig. 2.3. This time-varying torque (reduced by what is needed to overcome the engine's friction, Fig. 2.7), as a function of crank angle for an individual cylinder, is shown in Fig. 2.4. Thus each combustion event in each cylinder creates a torque pulse that is used to drive the engine's load.

Most engines are multi-cylinder engines for the reasons explained in Sec. 1.5 . Then, the motions of the individual piston, connecting rod, and crank throw mechanism, with the appropriate crank angle phasing, are summed together. The major advantage, of course, is that more cylinders (and thus smaller displacement cylinders) results in more frequent and smaller torque pulses and thus smoother engine operation. An important downside is that the time-varying forces and the dynamics of the motions of the several piston/con rod combinations do not necessarily fully *balance*, resulting in oscillating net forces on engine components that can cause vibrations of the engine structure. In this section, we examine the dynamics of piston, connecting rod, and crankshaft behavior in multi-cylinder engines. In the following section (15.7.3), we apply the results of this analysis to the task of engine balancing.

At high engine speeds, the forces required to accelerate and decelerate the reciprocating mass (the piston assembly plus upper part of the connecting rod) become significant. These are termed *inertia forces* (mass × acceleration), and in multi-cylinder engines we want these forces to balance so that the net engine inertia force is zero. Unbalanced vertical or lateral inertia forces, if large enough, would cause the engine to vibrate, potentially exciting vibrations in the vehicle structure.

Following the analysis of Den Hartog, [64] the piston's displacement, given in exact form by Eq. (2.6) , can be simplified for this purpose by expanding the square-root term as a power series in $1/R$ (which is close to 0.3 for most engines), and retaining only the first term in that expansion. This yields the approximation for c, the piston clearance from its TC position,

$$\frac{c}{a} = \left(1 + \frac{1}{4R}\right) - \left(\cos \omega t + \frac{1}{4R} \cos 2\omega t\right) \tag{15.26}$$

where $R = l/a$ (l is the con rod length, a the crank radius) and is between 3.0 and 3.5, ω is the crankshaft rotation rate in radians/s, and ωt equals the crank angle through which the crankshaft has turned at time t.

Differentiating once with respect to time gives the piston velocity (S_p); and differentiating twice, the acceleration (a_p), to obtain

$$S_p = a\omega\left(\sin\omega t + \frac{1}{2R}\sin 2\omega t\right) \tag{15.27}$$

$$a_p = a\omega^2\left(\cos\omega t + \frac{1}{R}\cos 2\omega t\right) \tag{15.28}$$

After multiplying by the piston mass, these expressions become the piston's vertical momentum and the inertia force, respectively. They consist of two terms, one varying with the same frequency as the crankshaft rotation and known as the *primary* or first-order term and the other varying at twice this frequency, known as the second-order term. [o] If the connecting rod is infinitely long, $R \to \infty$, and Eq. (15.28) indicates that the secondary term disappears and the piston executes harmonic motion. With normal connecting-rod lengths ($l \approx 3.5\ a$) the secondary term is significant.

The piston inertia force is given by the product $m_p a_p$. The piston mass is the mass of the piston, rings, piston pin, and upper portion of the connecting rod. Usually this con rod (reciprocating) mass is determined by replacing the actual connecting rod by two concentrated masses (one at each end of the rod at the bearing axes) so that the total rod mass remains unchanged, as does the location of its center of gravity. [p] The motion of the connecting rod is complex, with its top bearing axis moving up and down in a straight line and the bottom bearing axis moves in a circle. However, if the path of the center of gravity in this simple mass distribution model remains unchanged by this substitution, then the total inertia force on the rod is the same. In summary, the inertia force of the piston is

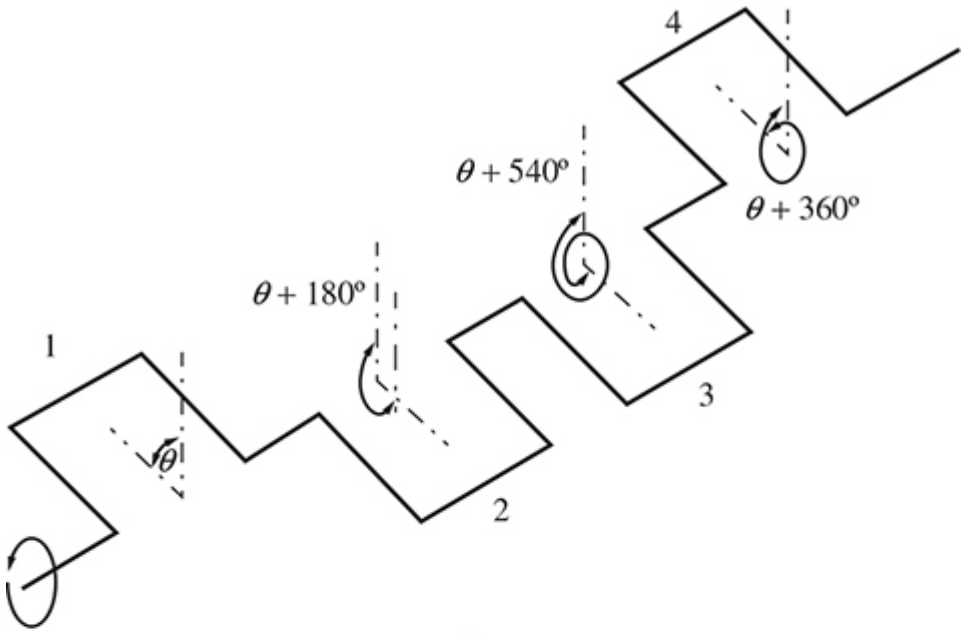
$$F_{p,i} = (m_p + m_{r,\text{rec}})a_p \tag{15.29}$$

Typically, the reciprocating component of the connecting rod mass is close to one-third the total mass. Note, importantly, that the magnitude of the inertia

force in Eq. (15.29) , through the piston acceleration Eq. (15.28) , increases as the square of engine speed. Thus, at 6000 rev/min it is 16 times its value at 1500 rev/min and is a dominant force. At low engine speeds, it is not nearly as important and is usually neglected.

We illustrate the use of the above analysis by examining the primary and secondary components of the piston inertia force for an in-line four-cylinder engine. This multi-cylinder configuration is the easiest to understand because the reciprocating motions of each piston (in its own cylinder) are 180° apart.

Figure 15.57 *a* shows, schematically, the positions of the four crank throws as the crankshaft rotates. If the crank angle position of the throw for cylinder number 1 is θ [which equals ωt in Eq. (15.28)], then for cylinders 2, 4, and 3, the throw angles at equivalent points in the operating cycle are (θ + 180), (θ + 360), (θ + 540) degrees, respectively, as shown.[q] Thus the cos(ωt) term in Eq. (15.28) for each of the pistons alternates in sign L: if θ is, say 45° CA, they are positive, negative, positive, negative. However, the cos(2 ωt) term for each of the four pistons, with the angles now doubled, all have the same sign (again if θ is 45°, they are all positive). The combination of the individual piston first-order, and then second-order inertia forces for a given multi-cylinder engine geometry, when displayed in a star diagram, provides a useful visual representation of whether the engine is balanced or not. Figure 15.57 *b* shows that the first-order (primary) inertia forces in the four-cylinder in-line engine balance. The second-order forces do not: with the same sign they all add to each other, as shown. This second-order inertia force is *out of balance*. To ensure smooth operation with four-cylinder in-line engines, balance shafts are frequently used to offset these (rotating) second-order out-of-balance inertia forces: see Sec. 15.7.3 .



(*a*)



(*b*)

Figure 15.57 (*a*) Schematic showing phasing of crank throws and their individual cylinders and pistons in four-cylinder in-line engines. (*b*) Star diagram of first-order (primary) and second-order individual-piston reciprocating inertia forces. The first-order inertia forces balance; the second-order do not.

An additional dynamic feature of piston behavior that can generate noise and vibration is *piston slap*. As the piston moves through its top center position, due to the necessary clearance between the piston and liner the

piston rotates slightly as it transitions from one side of the liner to the other, as shown in Fig. 15.58. During the compression stroke, the connecting rod reaction force pushes the piston to the right-hand side in Fig. 15.58 (the minor-thrust side); during the expansion power stroke, the connecting rod pushes the piston to the left, the major-thrust side. As this transition at TC occurs, the piston rotates as shown and then straightens up. The forces on the piston are high and increasing at this point in the cycle as combustion proceeds. If the piston "slaps" the major-thrust side hard, liner vibration and noise can result. In Sec. 2.2 we explained that the piston pin axis is usually offset by 1.5% of the bore toward the major-thrust side of the liner to reduce this impact by initiating this piston transition from the minor-thrust side of the liner earlier when the forces on the piston are lower. As indicated in the figure, this transition then starts some 15 degrees before TC and ends at 5 degrees after TC.

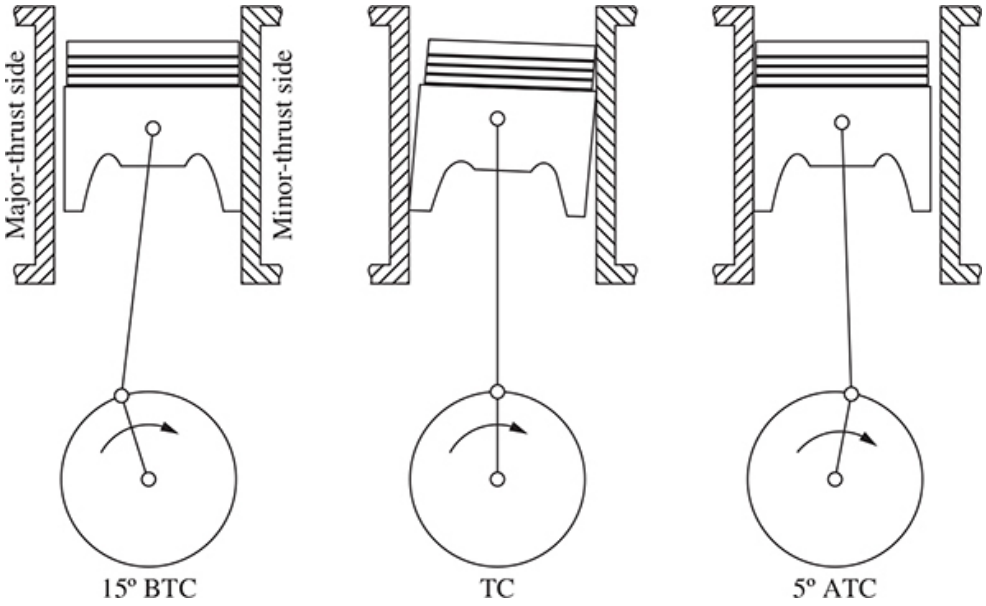


Figure 15.58 Piston motion from one side of the liner to the other side, just before TC, during the early part of combustion. With piston-pin offset, transition occurs earlier than TC so piston slap is reduced. [65]

15.7.3 Engine Balancing

The time-varying forces involved in the reciprocating engine mechanism result from:

- Oscillating masses (pistons, top portion of the connecting rods)
- Rotating masses (crankshaft, connecting rod big ends)
- Periodic forces (cylinder pressure)

As discussed above, at higher engine speeds the forces associated with accelerating and decelerating these masses (called inertia forces—mass times acceleration) are significant. In multi-cylinder engines, we want to "balance" these inertia forces so that the net force on the crankshaft is zero. Unbalanced forces cause the engine to vibrate, exciting vibrations in the vehicle.

Figure 15.59 illustrates these phenomena: vertical (axial) and horizontal (transverse) oscillating inertia forces acting on each piston; moments (in multi-cylinder engines) tending to rock the crankshaft; twisting crankshaft moments due to unequal forces on the crank throws.

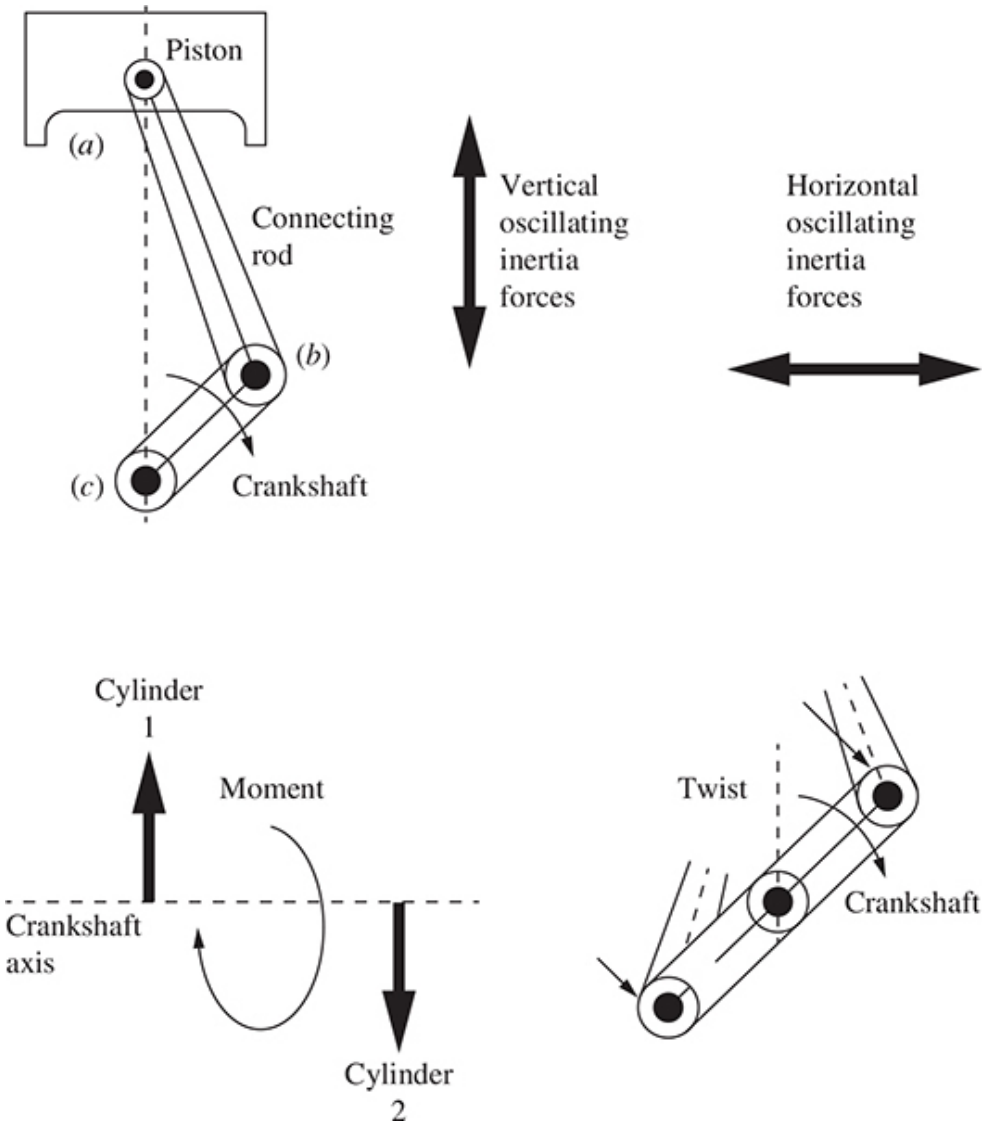


Figure 15.59 Schematics illustrating the oscillating inertia forces along (vertical) and transverse to (horizontal) the cylinder axis for each piston assembly. Also shown, are the moments tending to rock or twist the crankshaft which result from the different magnitude and the separation of the forces imposed by the motion of the pistons and connecting rods, in multi-cylinder engines.

Rotating masses are the easiest to balance: counterweights are used to oppose the centrifugal force: for example, appropriate counterweights can readily offset the rotating inertia of the crank throws and connecting rod big end. Oscillating masses generate forces that vary responsively: to first order at the crankshaft angular speed, and at double that speed at the second order.

Balancing can be achieved by using compensating unbalanced systems rotating at the appropriate speed and direction.

In the four-cylinder in-line engine discussed in the previous section, the first-order inertial forces balance. The out-of-balance second-order inertial forces can be offset using two identical auxiliary shafts rotating at double the rate of the crankshaft, in opposite directions. Balancing weights of appropriate mass are attached to these shafts as shown in Fig. 15.60. The two shafts are driven by the crankshaft at double the crankshaft rotation rate, with an idler gear to reverse the rotation direction of one of the shafts. The vertical inertia forces that these eccentric balance weights generate exactly offset the vertical components of the second-order inertia forces of the four pistons. The transverse component of the inertia force that each of the rotating balance weight shafts create cancel out since the shafts rotate in opposite directions. Thus the inertia forces in a four-cylinder in-line engine, *with* counter-rotating balance shafts become fully balanced.

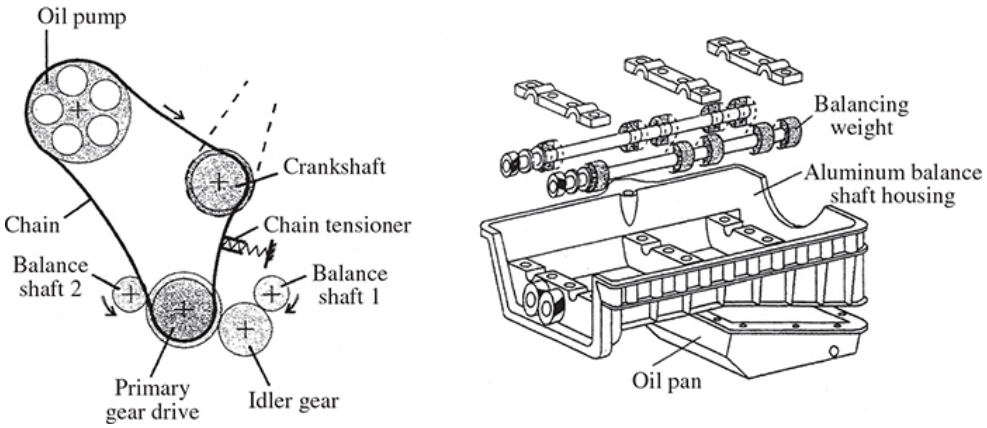


Figure 15.60 Example of balance shaft system for a four-cylinder in-line gasoline engine showing balancing weight and shaft positions, and gear drive mechanism.[66]

The inertial forces of a multi-cylinder engine result from the superposition of each individual piston-in-cylinder system, according to the crankshaft throw arrangement. Also, free moments are created as a result of the cylinder-to-cylinder spacing. Table 15.9 diagrams and summarizes the possible transverse and longitudinal titling moments, and axial free inertia forces and the vibration modes, that can result in multi-cylinder engines.

The mutual balancing of the inertial forces is one of the factors that determine the choice of an engine's number of cylinders and layout. The inertial forces are balanced if the common center of gravity for all moving crankshaft assembly components lies at the crankshaft's midpoint: that is, if the crankshaft is symmetrical when viewed from the front. [59] Figure 15.61 shows the crank arrangements and the star diagrams for the first- and second-order inertia forces for in-line engines, three through six-cylinders. All are balanced to first order (the diagrams are symmetric). All except the four-cylinder in-line engine (as discussed above) are balanced at second order.

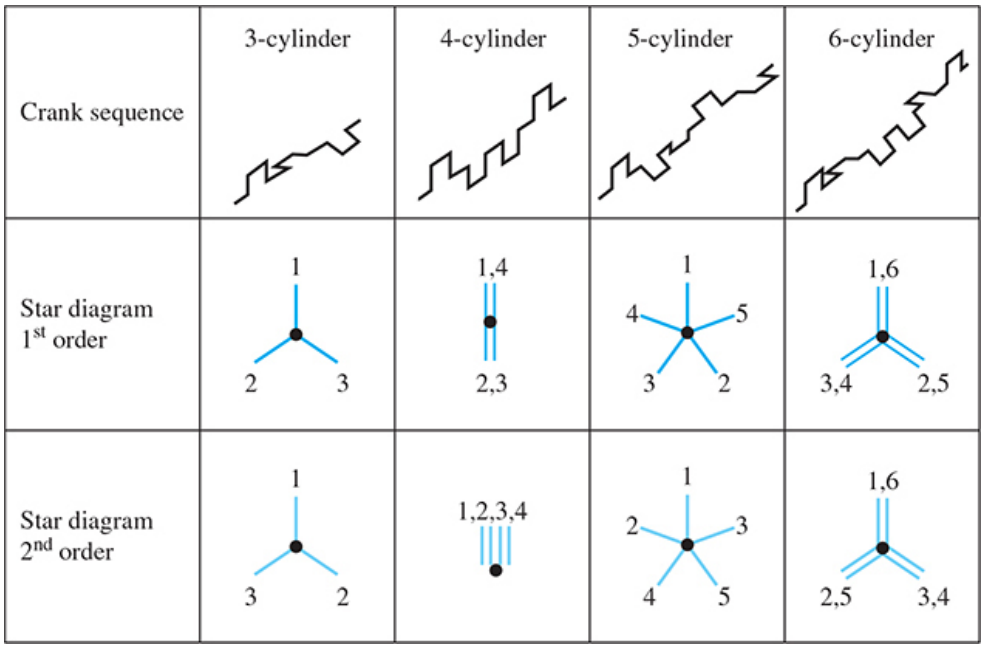
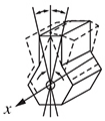
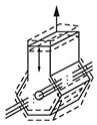
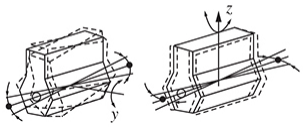
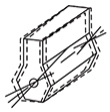


Figure 15.61 Crank arrangements and star diagrams for first- and second-order components of the reciprocating inertia (free) forces, for three- through six-cylinder in-line engines.[59]

TABLE 15.9 Transverse and longitudinal titling moments, and axial free-inertia forces, and modes of vibrations, in multi-cylinder engines

Forces and moments at the engine				
Modes of vibration	Oscillating torque about cylinder axis: Engine transverse titling moment	Free reciprocating axial inertia force	Free inertial moment: Longitudinal tilting moment about the *y* (transverse) axis; pitching moment about the *z* (vertical) axis, also rolling moment	Internal flexing/ twisting moment

Developed from Bosch Automotive Handbook.[59]

Thus, three-cylinder engine (inertia) forces are balanced to first and second order; first- and second-order moments are not. Four-cylinder in-line engines are first-order balanced, but unbalanced in their second-order free forces. Horizontally-opposed four-cylinder engines are balanced, except for second-order moments. Five cylinder engines are balanced in terms of free forces, but not in free moments. Six-cylinder in-line engines are fully balanced (and are often chosen for their smooth operation). V-6 engines (with 60° and 90° vee angles) are balanced in terms of inertia forces, but not free moments. Opposed six-cylinder engines (with 6 crank throws) are fully balanced. Eight-cylinder 90° V engines inertia forces are balanced to first and second order; free moments are not. Vee-12 engines are fully balanced. Additional details can be found in the Bosch Automotive Handbook. [59]

15.8 ENGINE PERFORMANCE AND FUELS SUMMARY

Engines

This chapter has described the major performance characteristics of the spark-ignition and the compression-ignition (diesel) engine, and has highlighted overall trends. Note that engine performance and efficiency are steadily improving over time, and emissions levels continue to be reduced. There are many different engine configurations and uses, and for each of these there are differences in design and operating characteristics. Nonetheless, our discussion of representative values of the various performance parameters for the more important types of internal combustion engines can be summarized as follows:

1. Within a given category of engines (e.g., naturally-aspirated four-stroke SI engines) the performance characteristics such as values of maximum bmep (at the maximum torque), specific power and $\overline{S}_p$ (at maximum rated power), and best bsfc (at lower engine speeds and close to wide-open throttle) are closely comparable.
2. Within an engine category where the range in engine size is substantial, there is an increase in maximum bmep and a decrease in minimum bsfc as size increases due to the decreasing relative importance of friction and heat loss per cycle.
3. The higher maximum bmep of gasoline-fueled SI engines compared to equivalent diesels results from the close-to-stoichiometric operation of the former at wide-open throttle.
4. The effect of supercharging or turbocharging, by boosting inlet air density by increasing intake air pressure, increases maximum bmep and specific power values substantially. Air compression *with after-cooling* gives further increases in bmep relative to turbocharging without after-cooling, at the same manifold air pressure level. The maximum bmep of turbocharged SI engines is limited by allowable maximum pressures, and the onset of knock. The maximum bmep of turbocharged compression-ignition engines is stress-limited. The larger CI engines are designed to accept higher maximum cylinder pressures, and hence can utilize higher boost levels.
5. Two-stroke cycle gasoline spark-ignition engines have significantly lower bmep and higher bsfc than equivalent four-stroke cycle SI engines.
6. The best efficiency values of modern automotive high-speed (turbocharged) DI diesel engines are about 15 to 20% higher than for naturally-aspirated gasoline engines. However, the diesel has a greater advantage at lower loads due to its low pumping work and much leaner air/fuel ratios. Brake fuel conversion efficiencies at about one-quarter maximum load, and lower engine speeds, are 30 to 40% higher.
7. The maximum bmep and best brake fuel conversion efficiency of DI diesel engines used over the size range, 80 cm to 1 m bore, steadily improve with increasing cylinder size. This is due to the reduced

impacts of friction and heat loss per cycle, the latter enabling higher maximum cylinder pressure (thus achieving higher air boost levels): additionally, the output and efficiency of these larger engines can be usefully increased through turbo-compounding.

Fuels

This text has been largely focused on internal combustion engines operating with petroleum-based fuels, gasoline (or petrol) and diesel. These fuels currently dominate in engine use due to their being liquids with high inherent energy density, each extracted separately from multicomponent crude oil so their average characteristics can be controlled to match the fuel requirement of each type of engine (gasoline for SI, or diesel for CI). The current costs of these fuels (the combined extraction, processing, and delivery costs) make them in an overall sense the most attractive available options for engine users. However, they contain some 85% carbon (by mass) and are thus a major source of CO_2 (the key greenhouse gas emission).

Several alternatives are in limited use or being explored for use in transportation: natural gas (NG), liquid petroleum gas (LPG), ethanol, methanol, biodiesel; so is electricity. On a global scale, use of these alternatives is, as yet, modest. [4]

Since diesel fuel demand is growing faster than is gasoline demand worldwide, the use of petrol/gasoline in diesel-type engines, with a compression-ignition combustion process (achieved spontaneously, or with pilot diesel-fuel injection), is being explored as a way to better balance the demand for these two fuels. This represents one of the more promising opportunities for new Reaction-Controlled Compression-Ignition combustion modes (see Sec. 10.7.2).

Natural gas is used in spark-ignition engines, at moderate levels in a few countries where major tax incentives create an attractive market. It is obviously a gaseous and not a liquid fuel. Its combustion characteristics are not that different from gasoline. It is also used in large spark-ignition engines as an alternative to diesel. It has a higher knock resistance than gasoline but displaces significant air (some 10% in a stoichiometric mixture). On-board fuel storage in-vehicles is either as high-pressure gas, or liquefied in cryogenic tanks; both exact a significant cost, space and energy consumption penalty. Liquid petroleum gas, which is extracted along with natural gas, is used in limited amounts: at moderate pressures it can be stored as a liquid.

The cost of gasoline/petrol SI engine conversions (or dual fuel systems with petrol) is high.

The alcohols, ethanol and methanol, have been introduced as supplements to gasoline. For example, ethanol is used in E10 (10%) blends with gasoline in the United States. It is largely produced from corn (in the U.S.) and sugar cane (Brazil). It has a significantly higher octane rating than gasolines (greater resistance to knock), and blending at these levels upgrades the base gasoline. Methanol offers similar benefits to those of ethanol, and is being used at modest levels in China. Both these alcohols have attractive characteristics, but also negatives: lower energy density relative to hydrocarbon fuels (ethanol some 70%, methanol about 50%); separation from gasoline-alcohol mixtures when water is present, toxicity. Currently, ethanol and methanol costs are higher than gasoline (per unit of energy). Future prospects and biomass supply constraints are uncertain. The full life-cycle greenhouse gas emissions impacts of ethanol and methanol are still being debated. Biodiesel (see Sec. 10.5.5) is also being used at the few percent level. Blended at this level with petroleum-based diesel fuel, its benefits and drawbacks are modest.

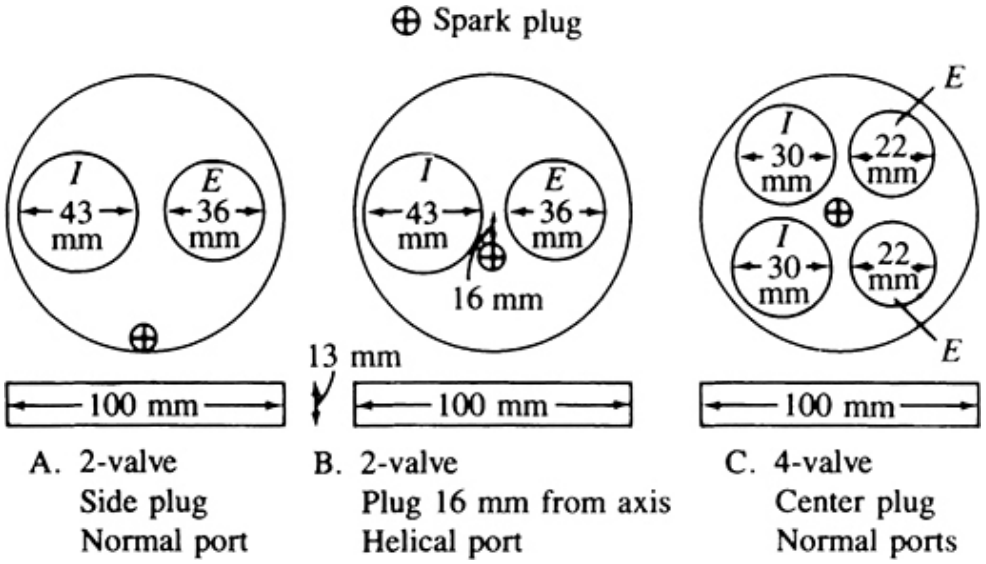
Use of electricity in plug-in-hybrid and battery electric propulsion systems is occurring. The potential for much larger-scale use of electricity is unclear: issues of relative costs, driving range (for EVs), supply-system recharging capabilities and recharging time, emissions from electricity generation, are as yet unresolved.

Overall, alternatives to petroleum-based gasoline and diesel face a challenging set of tasks. These mainstream fuels dominate a vast fuel production and supply system. If the alternative fuel is not readily compatible with this system, which long experience has honed to a high level of effectiveness and low cost, then establishing and growing a convenient-to-use and widely based alternative is a really challenging task. [4]

PROBLEMS

15. 1 The schematics show three different four-stroke cycle spark-ignition engine combustion chambers. A and B are two-valve engines, C is a four-valve engine (two inlet valves which open simultaneously, two exhaust

valves). Dimensions in millimeters are indicated. A and C have normal inlet ports and do not generate any swirl, B has a helical inlet port and generates substantial swirl. Spark plug locations are indicated. All three engines operate at the same speed (3000 rev/min), with the same inlet mixture composition, temperature, and pressure, and have the same displaced volume.



(a) Rank the chambers 1, 2, 3 in the order of their volumetric efficiency (1 = highest η_v).

(b) Rank the chambers in order (1, 2, 3) of their flame frontal area (1 = highest) when the mass fraction burned is about 0.2 and the piston is at TC.

(c) Given this relative flame front area ranking, discuss whether the ranking by mass burning rate dm_b/dt will be different from the flame area ranking.

(d) Briefly discuss the knock implications of these three chamber designs. Which is likely to have the worst knock problem?

15.2 Figures 15.5 and 15.6 show the variation in brake-specific fuel consumption (bsfc) for a port-fuel-injected automobile spark-ignition (SI) engine and a high-speed automobile DI diesel as a function of load and speed, respectively. From these graphs determine, and then plot, brake fuel conversion efficiency: (1) as a function of speed at *full load* and (2) as a function of load at a mid-speed of 2500 rev/min. Both engines are naturally aspirated. Assume the engine details are:

	Compression ratio	Equivalence ratio range	Displacement, dm^3
Diesel	22	0.3–0.75	2.5
SI engine	9	1.0–1.2	3.0

(a) List the major engine design and operating variables that determine brake fuel conversion efficiency.

(b) Explain briefly the reasons for the shapes of the curves you have plotted and the relative relationship of the diesel and SI engine curves.

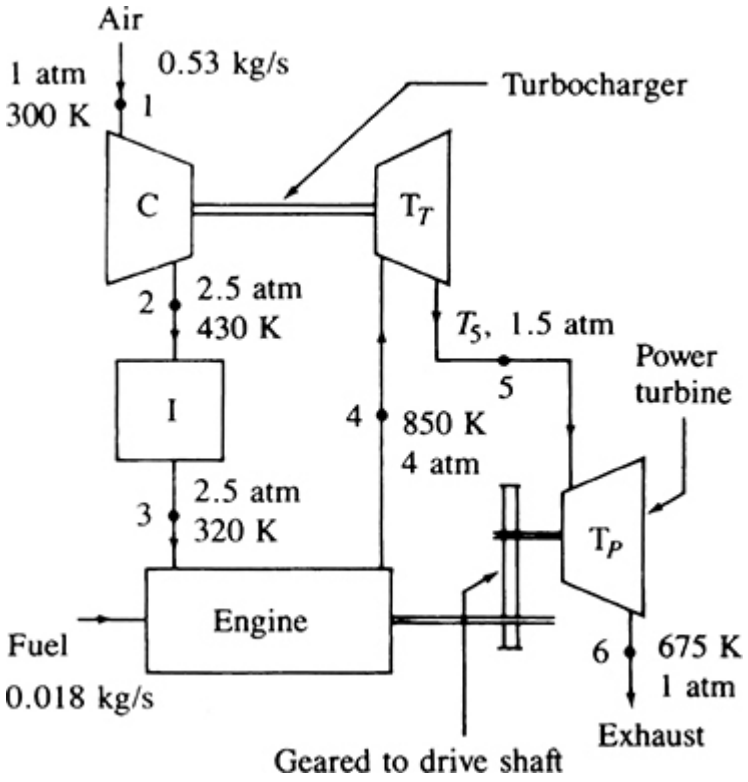
(c) At 2500 rev/min, estimate which engine will give the higher maximum brake power.

15.3 The diesel system shown in the figure consists of a multi-cylinder reciprocating DI diesel engine, a turbocharger (with a compressor C and turbine T_T mechanically connected to each other), an intercooler (I), and a power turbine (T_P) which is geared to the engine drive shaft. The gas and fuel flow paths and the gas states at the numbered points are shown. You can assume that the specific heat at constant pressure c_p of the gas throughout the entire system is 1.2 kJ/kg·K and $\gamma = c_p/c_v = 1.333$. The engine operates at 1900 rev/min. The fuel has a lower heating value of 42 MJ/kg of fuel.

(a) What is the power (in kilowatts) which the turbocharger turbine (T_T) must produce? What is the gas temperature at the exit to the turbocharger turbine?

(b) What is the power turbine power output?

(c) The heat losses in the engine are 15% of the fuel's chemical energy ($\dot{m}_f Q_{LHV}$). Find the engine power output, the total system power output, and the total system brake fuel conversion efficiency (friction effects in the engine and power turbine are internal to these devices and do not need to be explicitly evaluated).



15.4 The graph shows how the brake power and specific fuel consumption of a four-stroke cycle single-cylinder spark-ignition engine vary with the fuel/air equivalence ratio at wide-open throttle. It also shows how the following efficiencies vary with equivalence ratio:

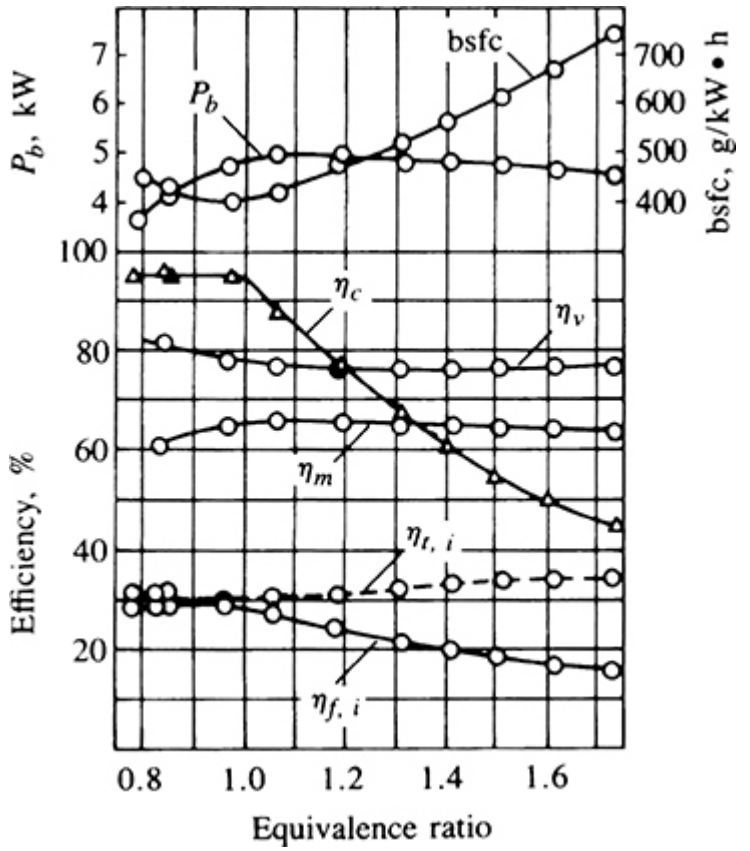
The volumetric efficiency: η_v

The mechanical efficiency: η_m

The combustion efficiency: η_c

The indicated fuel conversion efficiency: $\eta_{f,i}$

The indicated thermal conversion efficiency: $\eta_{t,i}$



(a) Derive a relation between the variables $\eta_{f,i}$, η_c, and $\eta_{t,i}$.

(b) Derive an equation which relates the brake power p_b to η_v, η_m, η_c, $\eta_{t,i}$ and any other engine and fuel parameters required.

(c) Explain briefly why the variations of η_v, η_m, η_c, $\eta_{f,i}$, $\eta_{t,i}$ with equivalence ratio in the figure have the form shown (e.g., why the parameter is approximately constant, or has a maximum/minimum, or decreases/increases with increasing richness or leanness, etc.).

15.5 Automobile spark-ignition engines employ "fast-burn technology" to achieve an improvement in fuel consumption and reductions in hydrocarbon (HC) and oxides of nitrogen (NOJ emissions. This question asks you to explain the experimental data which shows that faster-burning combustion chambers do provide these benefits relative to more moderate burn-rate chambers.

(a) Figure 9.36*b* shows the effect of increasing the percent of the exhaust gas recycled to the intake (for NO_x control) in a moderate burn-rate engine at constant speed and load, stoichiometric air/fuel ratio, with timing adjusted

for MBT at each condition. COV_{imep} is the standard deviation in imep divided by the average imep, in percent. The different types of combustion are: *misfire*, *partial burn*, *slow burn*, *normal burn*, defined in Sec. 9.4.3 . Frequency is percent of cycles in each of these categories. Use your knowledge of the spark-ignition engine flame-propagation process and HC emission mechanism to explain these trends in COV_{imep}, HC, and frequency as EGR is increased.

(b) This fast-burn combustion chamber uses two spark plugs and generates swirl inside the chamber by placing a vane in the inlet port to direct the air to enter the chamber tangentially. The swirl angular velocity in the cylinder at the end of intake is six times the crankshaft angular velocity. There is no swirl in the moderate burn-rate chamber which has a single spark plug and a relatively quiescent in-cylinder flow. The table shows spark timing, average time of peak pressure, average flame-development angle (0 to 10% mass burned) and rapid burning period (10 to 90% mass burned) for these two engines. Figure 15.32 shows how the operating and emission characteristics of the fast burn and moderate burn-rate engines change as percent EGR is increased. Explain the reasons for the differences in these trends in COV_{imep}, bsfc (brake-specific fuel consumption), and HC, and similarity in NO_x. The operating conditions are held constant at the same values as before.

	Fast burn	Moderate burn	
Spark timing	18°	35°	BTC
Crank angle for average p_{max}	15°	16°	ATC
0–10% burned	24°	35°	
10–90% burned	20°	35°	

15.6 Two alternative fuels, methanol and hydrogen, are being studied as potential future spark-ignition engine fuels which might replace gasoline (modeled by isooctane C_8H_{18}). The table gives some of the relevant properties of these fuels.

(a) For each fuel calculate the energy content per unit volume (in joules per cubic meter) of a stoichiometric mixture of fuel vapor and air at 1 atm and

350 K. The universal gas constant is 8314 J/kmol·K. What implications can you draw from these numbers regarding the maximum power output of an engine of fixed geometry operating with these fuels with stoichiometric mixtures?

(b) The octane rating of each fuel, and hence the knock-limited compression ratio of an engine optimized for each fuel, is different. Estimate the ratio of the maximum indicated mean effective pressure for methanol- and hydrogen-fueled engines to that of the gasoline-fueled engine, allowing for energy density effects at intake (at 1 atm and 350 K), at the knock-limited compression ratio for each fuel, for stoichiometric mixtures. You can assume that the fuel-air cycle results (scaled by 0.85) for isooctane apply also for methanol and hydrogen cycles to a good approximation, when the energy density is the same.

(c) The lean operating limit for the three fuels is different as indicated. Estimate the ratio of indicated fuel conversion efficiency for methanol and hydrogen at their lean limit and knock-limited compression ratio, relative to gasoline at its lean limit and knock-limited compression ratio, at the same inlet pressure (0.5 atm). Under these conditions, rank the fuel-engine combinations in order of decreasing power output.

	Gasoline (isooctane), C_8H_{18}	Methanol, CH_3OH	Hydrogen, H_2
Stoichiometric *F/A*	0.066	0.155	0.0292
Lower heating value, MJ/kg	44.4	20.0	120.1
Molecular weight of fuel	114	32	2
Molecular weight of stoichiometric mixture	30.3	29.4	21
Research octane number	95	106	~90
Knock-limited compression ratio	9	12	8
Equivalence ratio at lean misfire limit	0.9	0.8	0.6

15.7 A four-stroke cycle naturally-aspirated direct-injection diesel is being developed to provide 200 kW of power at the engine's maximum rated speed. Using information available in Chaps. 2, 5, and 15, on typical values of critical engine operating parameters at maximum power and speed for good engine designs, estimate the following:

(a) The displaced volume, the compression ratio, the number of cylinders, the cylinder bore and stroke, and the maximum rated speed of an appropriate engine design that would provide this maximum power.

(b) The brake-specific fuel consumption of this engine design at the maximum power operating point.

(c) The approximate increase in brake power that would result if the engine was turbocharged to an air boost level of 3 bar.

15.8. Natural gas (which is close to 100% methane, CH_4) is being considered as a spark-ignition engine fuel. The properties of methane and gasoline (assume the same properties as isooctane) and the engine details for each fuel are summarized below (ϕ is the fuel/air equivalence ratio).

	Natural gas	Gasoline
Composition	CH_4	C_8H_{18}
Heating value, MJ/kg	50.0	44.3
Research octane number	120	94
Compression ratio	14	8
Displaced volume, dm^3	2	2
Lean misfire limit	$\phi = 0.5$	$\phi = 0.8$
Part-load equivalence ratio	$\phi = 0.6$	$\phi = 0.9$
Full-load equivalence ratio	$\phi = 1.1$	$\phi = 1.2$

As indicated in the table, the displaced volume of the engine is unchanged when the conversion for natural gas is made; however, the clearance height is reduced to increase the compression ratio.

(a) Estimate the ratio of the volumetric efficiency of the engine operating on natural gas to the volumetric efficiency with gasoline, at wide-open throttle and 2000 rev/min, with both fuels in the gaseous state.

(b) Estimate the ratio of the maximum indicated power of the engine operating with natural gas to the maximum power of the gasoline engine.

(c) Estimate the ratio of the gross indicated fuel conversion efficiency of the natural gas engine to that of the gasoline engine, at the part-load conditions given.

(d) Explain whether the NO, CO, and hydrocarbon specific emissions (grams of pollutant per hour, per unit indicated power) at part-load conditions of the

natural gas engine will be higher, about the same, or lower than the NO, CO, and HC emissions from the gasoline engine. Explain briefly why.

You can assume that the fuel-air cycle results derived for isooctane-air mixtures are also appropriate for methane-air mixtures.

15.9. For four-stroke cycle direct-injection diesel engines:

(a) Show from the definition of mean effective pressure that

$$\text{bmep} \propto \eta_m \eta_{f,i} \eta_v \rho_{a,i} (F/A)$$

where bmep = brake mean effective pressure

η_m = mechanical efficiency

$\eta_{f,i}$ = indicated fuel conversion efficiency

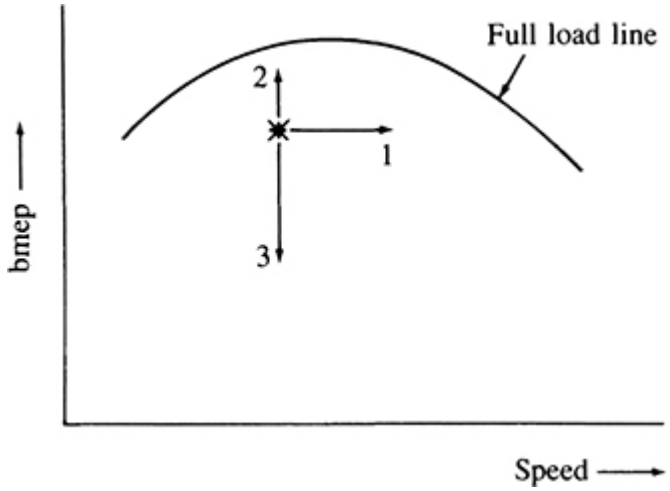
η_v = volumetric efficiency

$\rho_{a,i}$ = density of air in the intake manifold

F/A = fuel/air ratio

(b) Sketch carefully proportioned qualitative graphs of η_m, $\eta_{f,i}$, η_v, $\rho_{a,i}$ and $(F/A)/(F/A)_{\text{stoich}}$ versus speed N at full load, and explain the reasons for the shapes of the curves. Then explain why the maximum bmep versus speed curve has the shape shown in the figure below.

(c) The minimum brake-specific fuel consumption point is indicated by the asterisk (*) in the figure (see Fig. 15.34). Explain why brake-specific fuel consumption *increases* with (1) increasing speed, (2) increasing bmep, (3) decreasing bmep, from the best efficiency/bsfc point.



15.10. Figure 15.7 shows three performance maps for a 2-liter naturally-aspirated gasoline-fueled spark-ignition engine, with two valves per cylinder and standard fixed valve timing. Each performance map has bmep (w_e) as the vertical axis (1 kJ/liter = 1000 kPa), and engine speed as the horizontal axis. The upper curve is the wide-open-throttle bmep. Graph (*a*) shows lines of constant (gross) indicated fuel conversion efficiency. Graph (*b*) shows lines of constant mechanical efficiency. Graph (*c*) shows lines of constant brake fuel conversion efficiency.

(i) Explain what processes and what variables affect (a) the gross indicated fuel conversation efficiency and gross imep; (b) the mechanical efficiency; and (c) the brake fuel conversion efficiency and bmep.

Note, as an example, that engine heat transfer is a process, and engine speed is a variable. Also, only list processes and variables where engine fundamentals show they have a significant direct role. For example, engine heat transfer affects indicated performance directly, and affects brake performance only indirectly through its impact on indicated performance. A complete and well-explained list will be required to get good marks.

(ii) This is a standard production gasoline engine, as described above. Suggest specific improvements and changes in engine design and how the engine is operated, that you judge to be feasible, for a new production engine that will improve the maximum bmep, and the brake fuel conversion efficiency at 0.26 kJ/dm ^{3} (260 kPa) and 1500 rev/min, significantly. The engine will be naturally-aspirated. Provide approximate percentage improvements in each of the areas where you indicate improvements can be made, and combine these to indicate the percentage improvements in

maximum bmep and part-load brake fuel conversion efficiency overall that you judge could be achieved. Explain your logic clearly.

15.11. You are in the Environmental Affairs Department of the City of Cambridge, MA, and your task is to compare the CO_2 emissions per mile of travel from a natural-gas-fueled spark-ignition engine bus and an equivalent diesel engine bus. Both buses must achieve low air pollutant emissions as well. Both engines are turbocharged and intercooled and have comparable outputs.

The diesel bus engine has a compression ratio of 16, operates on average with an overall fuel-air equivalence ratio of 0.4, and an intake manifold pressure of 1.5 bar (150 kPa). The exhaust pressure is essentially the same as the intake pressure, 1.5 bar. The engine's mechanical efficiency at these conditions is 0.7. A particulate trap and selective catalytic reduction NO_x catalyst with urea as a reducing agent reduce air pollutant emissions to acceptable levels. These exhaust emission treatment systems increase the average engine bsfc by 6%. The diesel fuel is a low sulfur fuel with a formula $(CH_{1.8})_n$.

The natural-gas-fueled engine has a compression ratio of 12. The natural gas is injected into the engine's intake so natural gas displaces air (see Fig. 6.5). The engine operates stoichiometric, and at the average driving conditions the intake pressure is 0.8 bar (80 kPa) and exhaust pressure is 1.05 bar (105 kPa). The mechanical and accessory friction mean effective pressure is the same as in the diesel engine. This engine uses a three-way catalyst to reduce emissions with no fuel consumption penalty. Natural gas is 90% CH_4 and 10% C_2H_6 in molar proportions, so its H:C ratio is less than methane's value of 4.

The lower heating value for diesel fuel is 43.2 MJ/kg fuel, and for natural gas is 45 MJ/kg fuel.

(a) Estimate the bsfc of these two engines at their stated operating conditions.

(a) Determine the ratio of CO_2 emissions from the natural gas engine to CO_2 emissions from the diesel, each at its average operating conditions (which give closely comparable engine and hence vehicle performance). Useful units would be brake-specific emissions, g CO_2/MJ or g CO_2/kW·h.

15.12. One promising approach to improving the part-load efficiency of a spark-ignition engine is to deactivate half of the cylinders. For example, an

eight-cylinder engine at part load operates with the intake and exhaust valves on four of the cylinders closed through the complete four-stroke cycle, while the other four cylinders operate with normal valve timing. The throttle must be more open in this cylinder-deactivated mode to produce the same torque that the eight cylinders operating normally at light load would produce.

(a) Using the fuel-air constant-volume-combustion cycle as your model, draw three side-by-side carefully proportioned (qualitative) cylinder-pressure p versus cylinder volume V diagrams for one cylinder of: (i) the eight-cylinder engine operating normally with an intake pressure of 45 kPa (0.45 bar); (ii) the four firing cylinders in the deactivated engine operating mode; (iii) the four deactivated cylinders. Identify the maximum and minimum volumes on the V axis, the exhaust and intake pressures relative to atmospheric pressure, and the relative (qualitative) magnitudes of the end of compression pressure and maximum pressure in each of the three diagrams. Note that the intake and exhaust valves are deactivated immediately following the intake stroke. Briefly explain your diagrams.

(b) A 4-liter displacement, V-8 engine, must generate a brake torque of 80 N·m at this light-load condition. The mechanical rubbing plus accessory friction means effective pressure is 120 kPa and is the same for all cylinders (both those operating normally and those that are deactivated). Determine the imep in each cylinder when all 8-cylinders are operating, and the imep and intake manifold pressure of the four firing cylinders in the deactivated mode. The engine operates at stoichiometric and the compression ratio is 10.

(c) The intent of cylinder deactivation is to increase the brake fuel conversion efficiency at part-load conditions like those in (a) and (b) above. Calculate the ratio of the brake fuel conversion efficiency of the engine in (b) in its cylinder deactivated mode to the brake fuel conversion efficiency in its normal all-eight-cylinders firing mode.

Note: Standard definitions of mean effective pressures assume all cylinders are firing. Fuel-air cycle results when multiplied by 0.85 also apply to normally firing real engine cylinders. When half the cylinders are firing and half are deactivated, appropriate adjustments must be made in how these standard definitions and results are used.

15.13. For a 1.6-liter four-cylinder engine, with compression ratio of 9.1, we would like to get a sense of the forces the piston experiences. Assume that bore is equal to the stroke, the connecting rod length is 132 mm, the ratio of

connecting rod length to crank radius is 3.5, and the effective piston reciprocating mass is 0.57 kg (this includes the piston mass and a portion of the connecting rod mass). The crankcase pressure is 100 kPa. A typical fired pressure trace at high load is shown below.

Hint: A good approximation to the piston acceleration is

$$\ddot{s} = 2\pi^2 N^2 L \cdot \left(\cos\theta + \frac{1}{R}\cos 2\theta \right).$$

In this equation N = engine speed, L = stroke, θ = crank angle, measured from top center, R = ratio of connecting rod length to crank radius.

(a) Draw a free-body diagram of the piston. (A free-body diagram shows a body "free" from its surroundings with all of the forces and moments that act on it.)

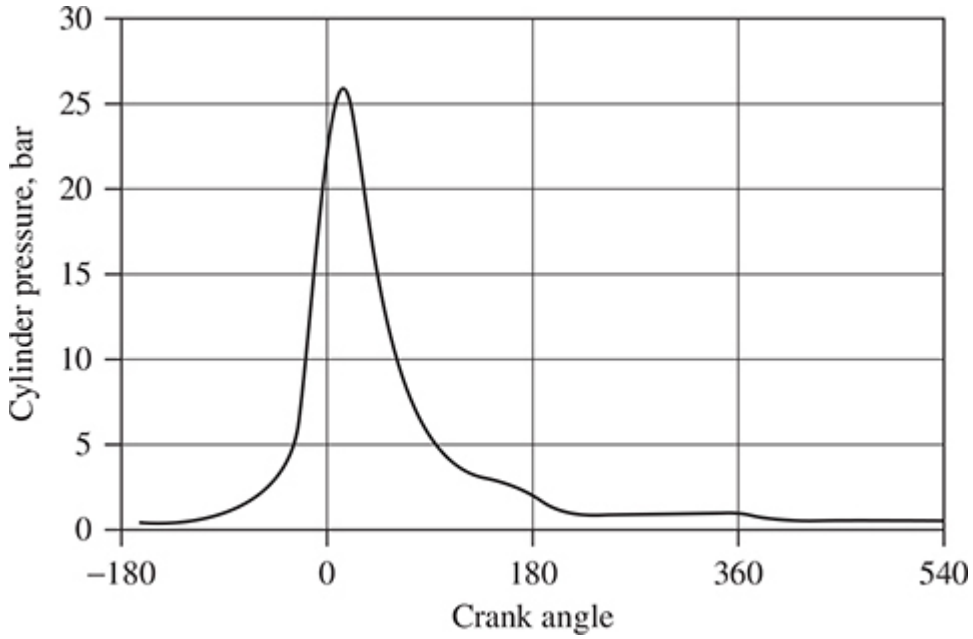
(b) For engine speeds of 1000 rev/min and 6000 rev/min, estimate the maximum inertial force (mass × acceleration) and pressure force on the piston.

We also want to examine the effect of the number of engine cylinders on the output of the engine. Instead of four cylinders with a total displaced volume of 1.6 liters, consider a single larger cylinder with the same displaced volume.

(c) Determine the ratio of maximum power outputs produced by these engines.

(d) Determine the ratio of maximum torques produced by these engines.

(e) Briefly describe the other factors that make the four-cylinder design much more attractive than the single-cylinder design.



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[a] *Load* is used here in the sense of relative torque: high load thus means close to wide-open-throttle or maximum torque.

[b] The basic dependence is on the air’s density in the intake system. Since the air temperature (in kelvins) does not vary much, density and absolute pressure scale together: thus the manifold air pressure (MAP) is often used to characterize the boost level.

[c] The shape of the imep curve for this naturally-aspirated gasoline engine indicates that the intake manifold has not been tuned to significantly improve the engine’s breathing capability in the mid- to high-speed range. See Fig. 6.12 and accompanying text.

[d] In this text we use *absolute* pressure levels (where zero is full vacuum). MAP values are sometimes given as *gauge* pressures: that is, relative to one atmosphere (≈ 1 bar).

[e] Note: this universal behavior between net indicated mep and combustion retard defined by $\phi_{50\%}$ mass fraction burned is expected. It is the indicated work transfer to the piston that combustion phasing controls and the midpoint of the burning process does effectively define the phasing of combustion relative to the piston position.

[f] This aspect of mixture nonuniformity is least well defined. Mixing of the fresh mixture (fuel, air, and EGR) with residual gas is likely to be incomplete (see Fig. 14.31), especially at light load when the residual gas fraction is highest. With intake-port fuel-injection systems, there is evidence of incomplete fuel-air mixing due to the fact that the airflow and fuel flow processes are not in phase. When the engine is cold, fuel distribution within the cylinder is known to be more nonuniform.

[g] EGR is normally zero at WOT, since maximum available torque is usually desired.

[h] Except during start-up and cold engine operation, when a significant part of the fuel within the cylinder can be in the liquid phase.

[i] With a given stoichiometry mixture of ethanol and air, the lower heating value of ethanol is offset by the higher *F/A* ratio required relative to gasoline, resulting in little difference in heating value per unit mass or volume of fuel vapor/air mixture.

[j] There is no assurance that the same cylinder will be the principal offender in all engines of the same model, nor in a given engine at all operating conditions.

[k] PRFs are isooctane/*n*-heptane mixtures: the number is the percent isooctane in the mixture. In the normal RON range (85 to 95 RON), the knock resistance of PRFs and gasolines with the same RON is closely comparable.

[l] Such a performance map for a naturally-aspirated gasoline engine would be similar. However, the wide-open-throttle torque curve would have a different shape and about half the bmep level (see Fig.

15.2). The part-load region where fuel consumption and emissions are critical would be comparable, again centered at about one-quarter the NA engine maximum torque level.

[m] An alternative to the high-pressure EGR system is a low-pressure system where EGR from downstream of the turbine is fed into the (atmospheric pressure) air entering the compressor.

[n] These engines were prototype engines used to explore the potential for two-stroke cycle engines in automotive applications.

[o] Note that in the expansion of the square root term in Eq. (2.6) , in a series in $1/R$, the third and higher order terms were omitted because they are small.

[p] This division of connecting rod mass into two distinct masses is an appropriate procedure for determining the inertia *forces*. However, it is not exact for determining the moments of these forces.

[q] Note that the firing order for a four-cylinder in-line engine is cylinder 1, 2, 4, 3 or 1, 3, 4, 2.

APPENDIX A

Unit Conversion Factors

This table provides conversion factors for common units of measure for physical quantities to the International System (SI) units. The conversion factors are presented in two ways: columns 2 and 3 give the conversion to the base or derived SI unit with the conversion factor as a number between one and ten with six or fewer decimal places, followed by the power of ten that the number must be multiplied by to obtain the correct value; columns 4 and 5 provide conversion to a recommended multiple or submultiple of the SI unit with the conversion factor given as a four-digit number between 0.1 and 1000.

1 To convert from	2 To	3 Multiply by	4 To	5 Multiply by
Area				
foot^2	m^2	$9.290\,304 \times 10^{-2}$	cm^2	929.0
inch^2	m^2	$6.451\,600 \times 10^{-4}$	cm^2	6.452
Energy, heat, and work				
Btu (International Table)	J	$1.055\,056 \times 10^{3}$	kJ	1.055
calorie (thermochemical)	J	$4.184\,000 \times 10^{0}$	J	4.184
erg	J	$1.000\,000 \times 10^{-7}$	μJ	0.1000
foot pound-force (ft · lbf)	J	$1.355\,818 \times 10^{0}$	J	1.356
horsepower-hour (hp · h)	J	$2.684\,520 \times 10^{6}$	MJ	2.685
kilowatt-hour (kW · h)	J	$3.600\,000 \times 10^{6}$	MJ	3.600
meter kilogram-force (m · kgf)	J	$9.806\,650 \times 10^{0}$	J	9.807
Energy (specific, specific heat)				
Btu (IT)/lb	J/kg	$2.326\,000 \times 10^{3}$	kJ/kg	2.326
Btu (IT)/lb · °F	J/kg · K	$4.186\,800 \times 10^{3}$	kJ/kg · K	4.187
calorie (thermo.)/g	J/kg	$4.184\,000 \times 10^{3}$	kJ/kg	4.184
calorie (thermo.)/g · °C	J/kg · K	$4.184\,000 \times 10^{3}$	kJ/kg · K	4.184
Force				
dyne	N	$1.000\,000 \times 10^{-5}$	μN	10.00
kilogram-force	N	$9.806\,650 \times 10^{0}$	N	9.807
pound-force	N	$4.448\,222 \times 10^{0}$	N	4.448
Force per unit length (includes surface tension)				
dyne/centimeter	N/m	$1.000\,000 \times 10^{-3}$	mN/m	1.000
pound-force/inch	N/m	$1.751\,268 \times 10^{2}$	N/m	175.1
pound-force/foot	N/m	$1.459\,390 \times 10^{1}$	N/m	14.59

Fuel consumption (economy)				
pound/horsepower-hour	kg/J	$1.689\,660 \times 10^{-7}$	g/kW · h	608.3
gram/kilowatt-hour	kg/J	$2.777\,778 \times 10^{-10}$	µg/J	0.2778
mile/gallon (U.S.)	m/m^3	$4.251\,437 \times 10^{5}$	km/dm^3	0.4251
mile/gallon (Imp.)	m/m^3	$3.540\,060 \times 10^{5}$	km/dm^3	0.3540
Heat flux (includes thermal conductivity)				
Btu (IT) · in/h · ft^2 · °F	W/m · K	$1.442\,279 \times 10^{-1}$	W/m · K	0.1442
Btu (IT)/ft^2	J/m^2	$1.135\,653 \times 10^{4}$	kJ/m^2	11.36
Btu (IT)/h · ft^2 · °F	$W/m^2 \cdot K$	$5.678\,263 \times 10^{0}$	$W/m^2 \cdot K$	5.678
calorie (thermo.)/cm^2	J/m^2	$4.184\,000 \times 10^{4}$	kJ/m^2	41.84
Length				
foot	m	$3.048\,000 \times 10^{-1}$	m	0.3048
inch	m	$2.540\,000 \times 10^{-2}$	mm	25.40
micron	m	$1.000\,000 \times 10^{-6}$	µm	1.000
mile	m	$1.609\,344 \times 10^{3}$	km	1.609
Mass				
ounce	kg	$2.834\,952 \times 10^{-2}$	g	28.35
pound	kg	$4.535\,924 \times 10^{-1}$	kg	0.4536
ton (long or Imp., 2240 lb)	kg	$1.016\,047 \times 10^{3}$	Mg	1.016
ton (short, 2000 lb)	kg	$9.071\,847 \times 10^{2}$	Mg	0.9072
tonne (metric)	kg	$1.000\,000 \times 10^{3}$	Mg	1.000
Mass per unit time (flow)				
pound/second	kg/s	$4.535\,924 \times 10^{-1}$	kg/s	0.4536
pound/minute	kg/s	$7.559\,873 \times 10^{-3}$	g/s	7.560
pound/hour	kg/s	$1.259\,979 \times 10^{-4}$	g/s	0.1260
Mass per unit volume				
gram/gallon (U.S.)	kg/m^3	$2.641\,724 \times 10^{-1}$	g/dm^3	0.2642
pound/$foot^3$	kg/m^3	$1.601\,846 \times 10^{1}$	kg/m^3	16.02
pound/$inch^3$	kg/m^3	$2.767\,990 \times 10^{4}$	kg/dm^3	27.68
pound/gallon (Imp.)	kg/m^3	$9.977\,644 \times 10^{1}$	kg/dm^3	0.0998
pound/gallon (U.S.)	kg/m^3	$1.198\,264 \times 10^{2}$	kg/dm^3	0.1198
Power, heat flow				
Btu (IT)/hour	W	$2.930\,711 \times 10^{-1}$	W	0.2931
horsepower (550 ft · lbf/s)	W	$7.456\,999 \times 10^{2}$	kW	0.7457
horsepower (metric, CV, PS)	W	$7.354\,99 \times 10^{2}$	kW	0.7355
Pressure, stress (force per unit area)				
atmosphere (normal, 760 torr)	Pa	$1.013\,250 \times 10^{5}$	kPa	101.3
inch of mercury (60°F)	Pa	$3.376\,85 \times 10^{3}$	kPa	3.377
kilogram-force/$centimeter^2$	Pa	$9.806\,650 \times 10^{4}$	kPa	98.07
mm of mercury, 0°C (torr)	Pa	$1.333\,224 \times 10^{2}$	Pa	133.3
pound-force/$foot^2$	Pa	$4.788\,026 \times 10^{1}$	Pa	47.88
pound-force/$inch^2$ (psi)	Pa	$6.894\,757 \times 10^{3}$	kPa	6.895
Temperature interval				
degree Celsius	K	$1.000\,000 \times 10^{0}$		
degree Fahrenheit	K	$5.555\,556 \times 10^{-1}$	K	0.5556
Temperature				
temperature (°C)	K	°C + 273.15		
temperature (°F)	K	(°F + 459.67)/1.80	°C	(°F − 32)/1.80

Torque				
kilogram-force meter	$N \cdot m$	$9.806\,650 \times 10^{0}$	$N \cdot m$	9.807
pound-force foot	$N \cdot m$	$1.355\,818 \times 10^{0}$	$N \cdot m$	1.356
Velocity				
foot/second	m/s	$3.048\,000 \times 10^{-1}$	m/s	0.3048
kilometer/hour	m/s	$2.777\,778 \times 10^{-1}$	m/s	0.2778
mile/hour	m/s	$4.470\,400 \times 10^{-1}$	km/h	1.609
Viscosity				
centipoise	$Pa \cdot s$	$1.000\,000 \times 10^{-3}$	$mPa \cdot s$	1.000
centistoke	m^2/s	$1.000\,000 \times 10^{-6}$	mm^2/s	1.000
poise	$Pa \cdot s$	$1.000\,000 \times 10^{-1}$	$Pa \cdot s$	0.1000
stoke	m^2/s	$1.000\,000 \times 10^{-4}$	mm^2/s	100.0
Volume				
barrel (42 U.S. gallon)	m^3	$1.589\,873 \times 10^{-1}$	m^3	0.1590
$foot^3$	m^3	$2.831\,685 \times 10^{-2}$	dm^3	28.32
gallon (Imp.)	m^3	$4.546\,092 \times 10^{-3}$	dm^3	4.546
gallon (U.S.)	m^3	$3.785\,412 \times 10^{-3}$	dm^3	3.785
$inch^3$	m^3	$1.638\,706 \times 10^{-5}$	cm^3	16.39
liter	m^3	$1.000\,000 \times 10^{-3}$	dm^3	1.000
Volume per unit time				
$foot^3$/minute (cfm)	m^3/s	$4.719\,474 \times 10^{-4}$	dm^3/s	0.4719
$foot^3$/second	m^3/s	$2.831\,685 \times 10^{-2}$	dm^3/s	28.32
gallon (U.S.)/minute (gpm)	m^3/s	$6.309\,020 \times 10^{-5}$	cm^3/s	63.09

Notes:

1. Derived units such as that for torque (newton · meter, N · m) are written with a period between each component unit for clarity. In practice, the period is often omitted.
2. Derived from Mobil Technical Bulletin *SI Units, The Modern Metric System*. Copyright Mobil Oil Corporation, 1974. Sections reproduced courtesy Mobil Oil Corporation.

APPENDIX B

Ideal Gas Relationships

B.1 IDEAL GAS LAW

The gas species which make up the working fluids in internal combustion engines (e.g., oxygen, nitrogen, carbon dioxide, etc.) can usually be treated as ideal gases. This appendix reviews the relationships between the thermodynamic properties of ideal gases.

The pressure p, specific volume v, and absolute temperature T of an ideal gas are related by the ideal gas law

$$pv = RT \tag{B.1}$$

For each gas species, R is a constant (the gas constant). It is different for each gas and is given by

$$R = \frac{\tilde{R}}{M} \tag{B.2}$$

where $\tilde{R}$ is the universal gas constant (for all ideal gases) and M is the molecular weight of the gas. Since v is given by V/m, where V is the volume of a mass of gas m, Eq. (B.1) can be rewritten as

$$pV = mRT = \frac{m\tilde{R}T}{M} \tag{B.3}$$

B.2 THE MOLE

It is convenient to introduce a mass unit based on the molecular structure of matter, the mole:

> The mole is the amount of substance which contains as many molecules as there are carbon atoms in 12 grams of carbon-12. [a]

Thus, the number of moles n of gas is given by

$$n = \frac{m}{M} \tag{B.4}$$

and Eq. (B.3) becomes

$$pV = n\tilde{R}T \quad \text{or} \quad p = \rho RT \tag{B.5}$$

Values for the universal gas constant in different units are given in Table B.1. In the SI system, the value is 8314.3 J/kmol · K. The density of dry air at standard conditions (101.33 kPa and 25°C) is 1.184 kg/m ^{3}.

TABLE B.1 Values of universal gas constant $\tilde{R}$

8314.3 J/kmol · K
8.3143 J/mol · K
1.9859 Btu/lb-mole · °R
1543.3 ft · lbf/lb-mole · °R

B.3 THERMODYNAMIC PROPERTIES

It follows from Eq. (B.1) that the internal energy u [b] of an ideal gas is a function of temperature only:

$$u = u(T) \tag{B.6}$$

Since the enthalpy h is given by $u + pv$, it follows also that

$$h = h(T) \tag{B.7}$$

The specific heats at constant volume and constant pressure of an ideal gas, c_v and c_p, respectively, are defined by

$$c_v = \left(\frac{\partial u}{\partial T}\right)_v = \frac{du}{dT} \tag{B.8}$$

$$c_p = \left(\frac{\partial h}{\partial T}\right)_p = \frac{dh}{dT} \tag{B.9}$$

From Eq. (B.1) it follows that

$$c_p - c_v = R \tag{B.10}$$

The ratio of specific heats, γ, is a useful quantity:

$$\gamma = \frac{c_p}{c_v} \tag{B.11}$$

An *additional* restrictive assumption is often made that the specific heats are constants. This is not a necessary part of the ideal gas relationships.

In general, the internal energy and enthalpy of an ideal gas at a temperature T relative to its internal energy and enthalpy at some reference temperature T_0 are given by

$$u = u_0 + \int_{T_0}^{T} c_v(T)\, dT \tag{B.12}$$

and

$$h = h_0 + \int_{T_0}^{T} c_p(T)\, dT \tag{B.13}$$

The entropy at T, v, and p, relative to the entropy at some reference state T_0, v_0, p_0, can be obtained from the relationships

$$ds = \frac{c_v}{T}dT + R\frac{dv}{v} = \frac{c_p}{T}dT - R\frac{dp}{p} \tag{B.14}$$

which integrate to give

$$s = s_0 + \int_{T_0}^{T} \frac{c_v}{T} dT + R \ln \frac{v}{v_0} \qquad \text{(B.15}a\text{)}$$

and

$$s = s_0 + \int_{T_0}^{T} \frac{c_p}{T} dT - R \ln \frac{p}{p_0} \qquad \text{(B.15}b\text{)}$$

The properties u, h, and s can be evaluated on a per unit mass or per mole basis. On a mass basis, c_v, c_p, and R would have the units J/kg · K (Btu/lbm · °R); on a mole basis, u, h, and s are replaced by $\tilde{u}$, $\tilde{h}$, and $\tilde{s}$. R is then the universal gas constant $\tilde{R}$, c_v and c_p are replaced by $\tilde{c}_v$ and $\tilde{c}_p$, and $\tilde{c}_v$, $\tilde{c}_p$, and $\tilde{R}$ would have the units J/kmol · K (Btu/lb-mol · °R).

B.4 MIXTURES OF IDEAL GASES

The working fluids in engines are mixtures of gases. The composition of a mixture of ideal gases can be expressed in terms of the following properties of each component:

Partial pressure p_i. The pressure each component would exert if it alone occupied the volume of the mixture at the temperature of the mixture.

Parts by volume V_i/V. The fraction of the total mixture volume each component would occupy if separated from the mixture, at the mixture temperature and pressure.

Mass fraction x_i. The mass of each component m_i, divided by the total mass of mixture m.

Mole fraction $\tilde{x}_i$. The number of moles of each component n_i, divided by the total number of moles of mixture n.

From Eq. (B.5) it follows that

$$\frac{p_i}{p} = \frac{V_i}{V} = x_i \frac{M}{M_i} = \tilde{x}_i \qquad \text{(B.16)}$$

The thermodynamic properties of mixtures of ideal gases can be computed from the following relationships:

Molecular weight

$$M = \frac{1}{n}\sum_i n_i M_i = \sum_i \tilde{x}_i M_i \qquad \text{(B.17)}$$

Internal energy, enthalpy, and entropy

On a mass basis:

$$u = \sum_i x_i u_i \quad h = \sum_i x_i h_i \quad s = \sum_i x_i s_i \qquad \text{(B.18}a, b, c\text{)}$$

On a mole basis:

$$\tilde{u} = \sum_i \tilde{x}_i \tilde{u}_i \quad \tilde{h} = \sum_i \tilde{x}_i \tilde{h}_i \quad \tilde{s} = \sum_i \tilde{x}_i \tilde{s}_i \qquad \text{(B.19}a, b, c\text{)}$$

---

[a] This is the SI system definition of the mole; it was formerly called the gram-mole. The kilogram-mole (kmol) is also used; it is 1000 times as large as the mole.

[b] The symbol u will be used for internal energy per unit mass, $\tilde{u}$ for internal energy per mole, and U for internal energy of a previously defined system of mass m. Similar notation will be used for enthalpy, entropy, and specific heats, per unit mass and per mole.

APPENDIX C

Equations for Fluid Flow through a Restriction

In many parts of the engine cycle, fluid flows through a restriction or reduction in flow area. Real flows of this nature are usually related to an equivalent ideal flow. The equivalent ideal flow is the steady adiabatic reversible (frictionless) flow of an ideal fluid through a duct of identical geometry and dimensions. For a real fluid flow, the departures from the ideal assumptions listed above are taken into account by introducing a flow coefficient or discharge coefficient C_D, where

$$C_D = \frac{\text{actual mass flow}}{\text{ideal mass flow}}$$

Alternatively, the flow or discharge coefficient can be defined in terms of an effective cross-sectional area of the duct and a reference area. The reference area A_R is usually taken as the minimum cross-sectional area. The effective area of the flow restriction A_E is then the cross-sectional area of the throat of a frictionless nozzle which would pass the measured mass flow between a large upstream reservoir at the upstream stagnation pressure and a large downstream reservoir at the downstream measured static pressure. Thus

$$C_D = \frac{A_E}{A_R}$$

C.1 LIQUID FLOW

Consider the flow of a liquid through an orifice as shown in Fig. C.1 . For the ideal flow, Bernoulli's equation can be written

$$p_1 + \rho\frac{V_1^2}{2} = p_2 + \rho\frac{V_2^2}{2}$$

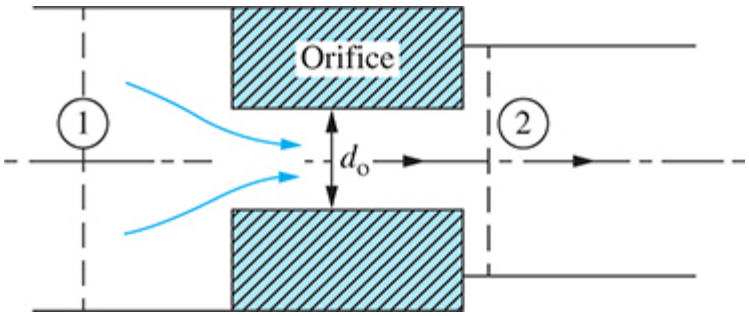


Figure C.1 Schematic of liquid flow through orifice.

For an incompressible flow, continuity gives $V_1 A_1 = V_2 A_2$ and the ideal mass flow rate through an orifice is given by

$$\dot{m}_{\text{ideal}} = A_2\left[\frac{2\rho(p_1 - p_2)}{1-(A_2/A_1)^2}\right]^{1/2} \qquad \text{(C.1)}$$

The real mass flow rate is obtained by introducing the discharge coefficient:

$$\dot{m}_{\text{real}} = C_D A_2\left[\frac{2\rho(p_1 - p_2)}{1-(A_2/A_1)^2}\right]^{1/2} \qquad \text{(C.2)}$$

The discharge coefficient is a function of orifice dimensions, shape and surface roughness, mass flow rate, and fluid properties (density, surface tension, and viscosity). The use of the orifice Reynolds number

$$\text{Re}_o = \frac{\rho V_2 D_2}{\mu} = \frac{V_2 D_2}{\nu}$$

along with orifice diameter ratio d_o/D_2, are correlating parameters for C_D, accounting for the effects of m over-dot, Greek *Rho*, Greek *Nu*, d_o, and D_2, to a good approximation. [1]

C.2 GAS FLOW

Consider the flow of an ideal gas with constant specific heats through the duct shown in Fig. C.2 . For the *ideal flow*, the stagnation temperature and pressure, T_0 and p_0, are related to the conditions at other locations in the duct by the steady flow energy equation

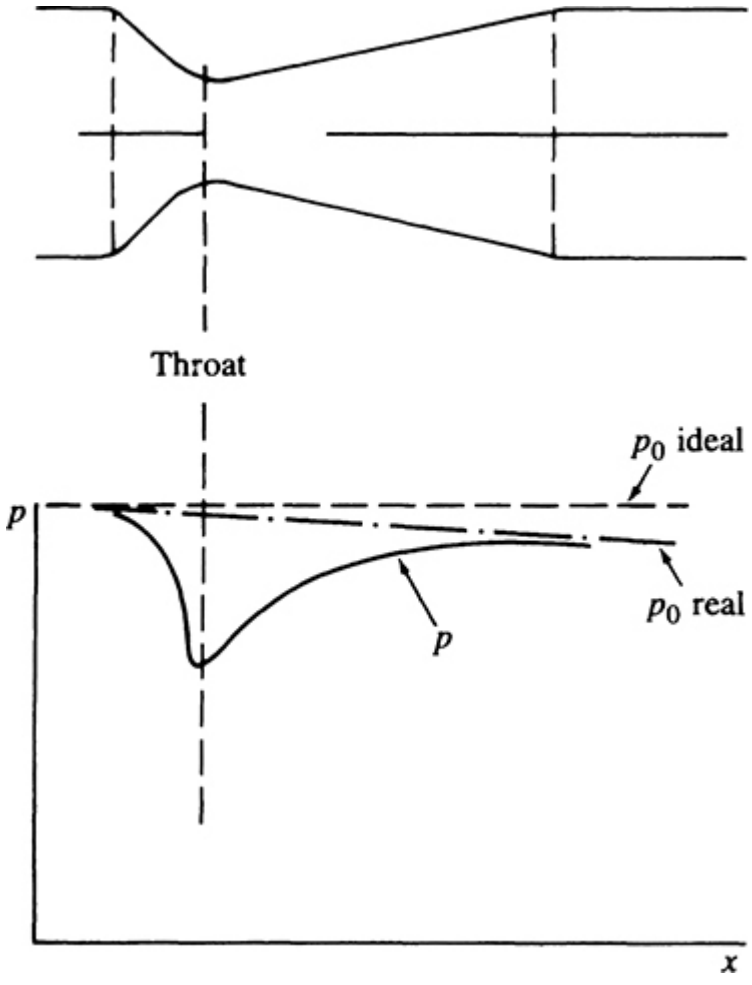


Figure C.2 Pressure distribution for gas flow through a nozzle.

$$T_0 = T + \frac{V^2}{2c_p}$$

and the isentropic relation

$$\left(\frac{T}{T_0}\right) = \left(\frac{p}{p_0}\right)^{(\gamma-1)/\gamma}$$

By introducing the Mach number $M = V/a$, where a is the sound speed $(=\sqrt{\gamma RT})$, the following equations are obtained:

$$\frac{T_0}{T} = 1 + \frac{\gamma - 1}{2} M^2 \qquad \text{(C.3)}$$

$$\frac{p_0}{p} = \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\gamma/(\gamma-1)} \qquad \text{(C.4)}$$

The mass flow rate $\dot{m}$ is

$$\dot{m} = \rho A V$$

With the ideal gas law and the above relations for p and T, this can be rearranged as

$$\frac{\dot{m}_{\text{ideal}}\sqrt{\gamma RT_0}}{Ap_0} = \gamma M\left(1 + \frac{\gamma - 1}{2} M^2\right)^{-(\gamma+1)/2(\gamma-1)} \qquad \text{(C.5)}$$

or

$$\frac{\dot{m}_{\text{ideal}}\sqrt{\gamma RT_0}}{Ap_0} = \gamma\left(\frac{p}{p_0}\right)^{1/\gamma}\left\{\frac{2}{\gamma - 1}\left[1 - \left(\frac{p}{p_0}\right)^{(\gamma-1)/\gamma}\right]\right\}^{1/2} \qquad \text{(C.6)}$$

For given values of p_0 and T_0, the maximum mass flow occurs when the velocity at the minimum area or throat equals the velocity of sound. This condition is called choked or critical flow. When the flow is choked the pressure at the throat, p_T, is related to the stagnation pressure p_0 as follows:

$$\frac{p_T}{p_0} = \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)}$$

This ratio is called the critical pressure ratio. For (p_T / p_0) less than or equal to the critical pressure ratio,

$$\frac{\dot{m}_{\text{ideal}}\sqrt{\gamma R T_0}}{A_T p_0} = \gamma\left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/2(\gamma-1)} \tag{C.7}$$

The critical pressure ratio is 0.528 for $\gamma = 1.4$ and 0.546 for $\gamma = 1.3$.

For a real gas flow, the discharge coefficient is introduced. Then, for sub-critical flow, the real mass flow rate is given in terms of conditions at the minimum area or throat by

$$\dot{m}_{\text{real}} = \frac{C_D A_T p_0}{\sqrt{RT_0}}\left(\frac{p_T}{p_0}\right)^{1/\gamma}\left\{\frac{2\gamma}{\gamma-1}\left[1-\left(\frac{p_T}{p_0}\right)^{(\gamma-1)/\gamma}\right]\right\}^{1/2} \tag{C.8}$$

For a choked flow,

$$\dot{m}_{\text{real}} = \frac{C_D A_T p_0}{\sqrt{RT_0}}\gamma^{1/2}\left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/2(\gamma-1)} \tag{C.9}$$

Equation (C.8) can be rearranged in the form of Eq. (C.2) (with $A_2 \ll A_1$) as

$$\dot{m}_{\text{real}} = C_D A_R[2\rho_0(p_0 - p_T)]^{1/2}\Phi \tag{C.10}$$

where Φ is given by

$$\Phi = \left\{\frac{[\gamma/(\gamma-1)][(p_T/p_0)^{2/\gamma} - (p_T/p_0)^{(\gamma+1)/\gamma}]}{1 - p_T/p_0}\right\}^{1/2} \tag{C.11}$$

Figure C.3 shows the variation of Φ and $(\dot{m}/\dot{m}^{*})_{\text{ideal}}$ with $(p_0 - p_T)/p_0$. $\dot{m}^*$ is the mass flow rate through the restriction under choked flow conditions (when the Mach number at the throat is unity). For flow rates less than about 60% of the choked flow, the effects of compressibility on the mass flow rate are less than 5%.

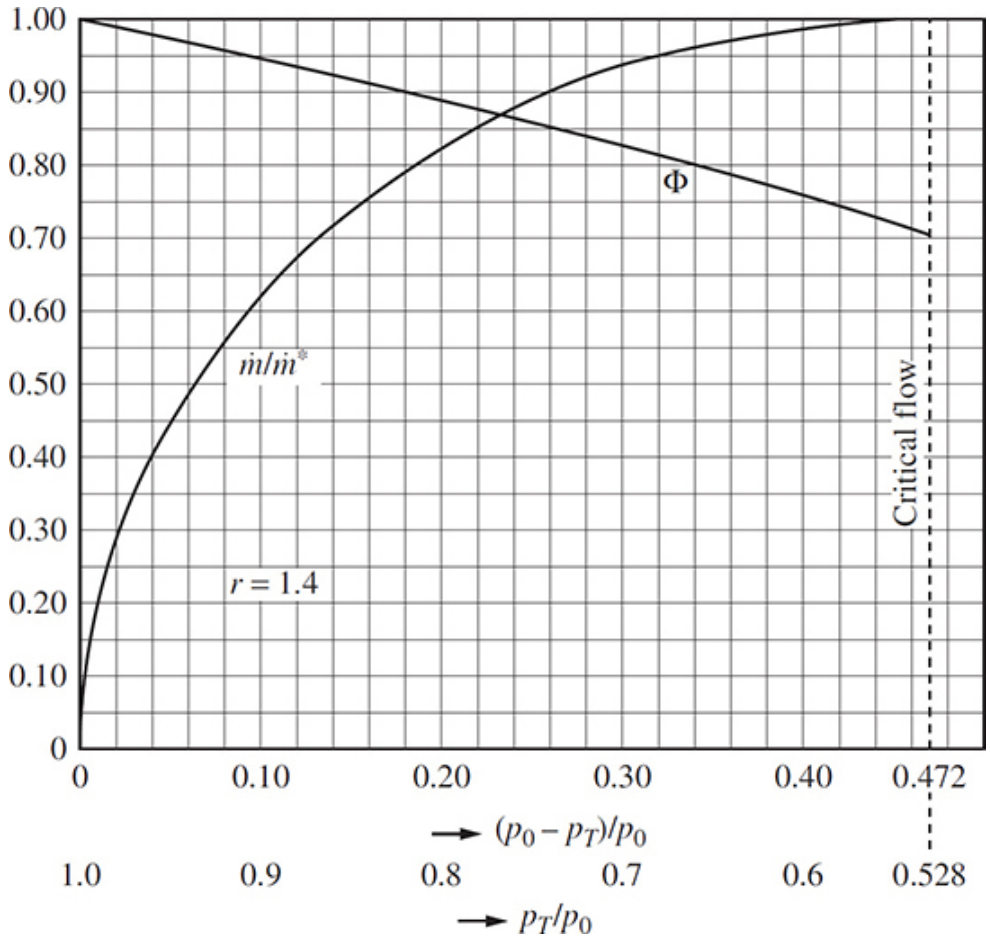


Figure C.3 Relative mass flow rate $\dot{m}/\dot{m}^*$ and compressible flow function Φ [Eq. (C.11)] as function of nozzle or restriction pressure ratio for ideal gas with $\gamma = 1.4$. (*From Taylor.* [2])

Flow coefficients are determined experimentally and are a function of the shape of the passage, the Reynolds number and Mach number of the flow, and the gas properties. For a Mach number at the throat less than about 0.7 and for passages of similar shape, the flow coefficient is essentially a function of Reynolds number only.

Orifice plates are frequently used to measure gas flow rates. Standard methods for determining flows through orifice plates can be found in Ref. 3 .

REFERENCES

1 . White, F. M.: *Fluid Mechanics*, 4th edition, McGraw-Hill, New York, 1999.

2 . Taylor, C. F.: *The Internal Combustion Engine in Theory and Practice*, vol. I, p. 506, MIT Press, Cambridge, MA, 1966.

3 . *Marks' Standard Handbook for Mechanical Engineers*, 8th ed., McGraw-Hill, New York, 1978.

APPENDIX D

Data on Working Fluids

TABLE D.1 Thermodynamic properties of air at low density †

			Ψ	Φ			c_p	c_v	
T, K	h, kJ/kg	u, kJ/kg	kJ/(kg · K)		p_r	v_r	kJ/(kg · K)		γ
250	409.9	338.1	4.4505	7.6603	38.81	1849.0	1.003	0.715	1.401
275	435.0	356.0	4.5187	7.7559	54.14	1458.0	1.003	0.716	1.401
300	460.1	374.0	4.5811	7.8432	73.39	1173.0	1.004	0.717	1.400
325	485.2	391.9	4.6385	7.9236	97.13	960.6	1.006	0.718	1.400
350	510.4	409.9	4.6919	7.9982	125.9	797.8	1.007	0.720	1.399
375	535.6	427.9	4.7416	8.0678	160.5	670.8	1.010	0.723	1.397
400	560.8	446.0	4.7884	8.1330	201.4	570.0	1.013	0.725	1.396
425	586.2	464.2	4.8324	8.1945	249.6	488.9	1.016	0.729	1.394
450	611.6	482.5	4.8742	8.2527	305.6	422.7	1.020	0.733	1.392
475	637.2	500.8	4.9139	8.3079	370.4	368.1	1.024	0.737	1.390
500	662.8	519.3	4.9518	8.3606	445.0	322.6	1.028	0.741	1.387
525	688.6	537.9	4.9881	8.4109	530.2	284.3	1.033	0.746	1.385
550	714.5	556.6	5.0229	8.4590	627.1	251.8	1.039	0.752	1.382
575	740.5	575.5	5.0565	8.5053	736.8	224.0	1.044	0.757	1.379
600	766.7	594.5	5.0888	8.5499	860.6	200.1	1.050	0.763	1.376
625	793.0	613.6	5.1201	8.5929	999.5	179.5	1.056	0.768	1.374
650	819.5	632.9	5.1503	8.6344	1155.0	161.5	1.061	0.774	1.371
675	846.1	652.3	5.1796	8.6745	1329.0	145.9	1.067	0.780	1.368
700	872.9	671.9	5.2081	8.7135	1521.0	132.1	1.073	0.786	1.365
725	899.8	691.7	5.2358	8.7512	1735.0	119.9	1.079	0.792	1.362
750	926.8	711.5	5.2628	8.7879	1972.0	109.2	1.085	0.798	1.360
775	954.0	731.6	5.2891	8.8236	2233.0	99.63	1.091	0.804	1.357
800	981.4	751.7	5.3147	8.8584	2520.0	91.12	1.097	0.810	1.354
825	1008.9	772.1	5.3397	8.8922	2836.0	83.52	1.103	0.816	1.352
850	1036.5	792.5	5.3641	8.9252	3181.0	76.71	1.108	0.821	1.350
875	1064.3	813.1	5.3880	8.9574	3559.0	70.58	1.114	0.827	1.347
900	1092.2	833.8	5.4114	8.9889	3971.0	65.07	1.119	0.832	1.345
925	1120.2	854.7	5.4342	9.0196	4419.0	60.08	1.124	0.837	1.343

950	1148.4	875.7	5.4566	9.0496	4907.0	55.58	1.129	0.842	1.341
975	1176.7	896.8	5.4786	9.0790	5436.0	51.49	1.134	0.847	1.339
1000	1205.1	918.1	5.5001	9.1078	6009.0	47.77	1.139	0.852	1.337
1025	1233.7	939.4	5.5212	9.1360	6629.0	44.39	1.144	0.856	1.335
1050	1262.3	960.9	5.5419	9.1636	7299.0	41.30	1.148	0.861	1.333
1075	1291.1	982.5	5.5622	9.1907	8020.0	38.48	1.152	0.865	1.332
1100	1319.9	1004.1	5.5821	9.2172	8797.0	35.90	1.157	0.870	1.330
1125	1348.9	1025.9	5.6017	9.2432	9632.0	33.53	1.161	0.874	1.329
1150	1378.0	1047.8	5.6209	9.2688	10529.0	31.35	1.165	0.878	1.327
1175	1407.1	1069.8	5.6399	9.2939	11490.0	29.36	1.168	0.881	1.326
1200	1436.4	1091.9	5.6585	9.3185	12520.0	27.51	1.172	0.885	1.324

†Abstracted with permission from *Thermodynamic Properties in SI* (Graphs, Tables, and Computational Equations for Forty Substances), by W. C. Reynolds, published by the Department of Mechanical Engineering, Stanford University, Stanford, CA 94305, 1979.

TABLE D.2 Standard enthalpy of formation and molecular weight of species

Species	Formula	Molecular weight, g/mole	State†	$\Delta\tilde{h}_f^\circ$ MJ/kmol	$\Delta\tilde{h}_f^\circ$ kcal/mol
Oxygen	O_2	32.00	gas	0	0
Nitrogen	N_2	28.01	gas	0	0
Carbon	C	12.011	solid	0	0
Carbon monoxide	CO	28.01	gas	−110.5	−26.42
Carbon dioxide	CO_2	44.01	gas	−393.5	−94.05
Hydrogen	H_2	2.016	gas	0	0
Water	H_2O	18.02	gas	−241.8	−57.80
Water	H_2O	18.02	liquid	−285.8	−68.32
Methane	CH_4	16.04	gas	−74.9	−17.89
Propane	C_3H_8	44.10	gas	−103.8	−24.82
Isooctane	C_8H_{18}	114.23	gas	−224.1	−53.57
Isooctane	C_8H_{18}	114.23	liquid	−259.28	−61.97
Cetane	$C_{16}H_{34}$	226.44	liquid	−454.5	−108.6
Methyl alcohol	CH_3OH	32.04	gas	−201.2	−48.08
Methyl alcohol	CH_3OH	32.04	liquid	−238.6	−57.02
Ethyl alcohol	C_2H_5OH	46.07	gas	−234.6	−56.08
Ethyl alcohol	C_2H_5OH	46.07	liquid	−277.0	−66.20

†At 298.15 K (25°C) and 1 atm.

TABLE D.3 Enthalpy of C, CO, CO_2, H_2, H_2O, N_2, O_2

	$\tilde{h}°(T) - \tilde{h}°(298.15)$, kcal/mol						
T (K)	C	CO	CO_2	H_2	H_2O	N_2	O_2
298	0.000	0.000	0.000	0.000	0.000	0.000	0.000
300	0.004	0.013	0.016	0.013	0.015	0.013	0.013
400	0.250	0.711	0.958	0.707	0.825	0.710	0.724
500	0.569	1.417	1.987	1.406	1.654	1.413	1.455
600	0.947	2.137	3.087	2.106	2.509	2.125	2.210
700	1.372	2.873	4.245	2.808	3.390	2.853	2.988
800	1.831	3.627	5.453	3.514	4.300	3.596	3.786
900	2.318	4.397	6.702	4.226	5.240	4.355	4.600
1000	2.824	5.183	7.984	4.944	6.209	5.129	5.427
1100	3.347	5.983	9.296	5.670	7.210	5.917	6.266
1200	3.883	6.794	10.632	6.404	8.240	6.718	7.114
1300	4.432	7.616	11.988	7.148	9.298	7.529	7.971
1400	4.988	8.446	13.362	7.902	10.384	8.350	8.835
1500	5.552	9.285	14.750	8.668	11.495	9.179	9.706
1600	6.122	10.130	16.152	9.446	12.630	10.015	10.583
1700	6.696	10.980	17.565	10.233	13.787	10.858	11.465
1800	7.275	11.836	18.987	11.030	14.964	11.707	12.354
1900	7.857	12.697	20.418	11.836	16.160	12.560	13.249
2000	8.442	13.561	21.857	12.651	17.373	13.418	14.149
2100	9.029	14.430	23.303	13.475	18.602	14.280	15.054
2200	9.620	15.301	24.755	14.307	19.846	15.146	15.966
2300	10.212	16.175	26.212	15.146	21.103	16.015	16.882
2400	10.807	17.052	27.674	15.993	22.372	16.886	17.804
2500	11.403	17.931	29.141	16.848	23.653	17.761	18.732
2600	12.002	18.813	30.613	17.708	24.945	18.638	19.664
2700	12.602	19.696	32.088	18.575	26.246	19.517	20.602
2800	13.203	20.582	33.567	19.448	27.556	20.398	21.545
2900	13.807	21.469	35.049	20.326	28.875	21.280	22.493
3000	14.412	22.357	36.535	21.210	30.201	22.165	23.446

Source: JANAF Thermochemical Tables, National Bureau of Standards Publication NSRDS-NBS37, 1971.

TABLE D.4 Data on fuel properties

Fuel	Formula (phase)[a]	Molecular weight	Specific gravity (density,[b] kg/dm^3)	Heat of vaporization, kJ/kg[c]	Specific heat		Higher heating value, MJ/kg	Lower heating value, MJ/kg	LHV of stoich. mixture, MJ/kg[d]	$(A/F)_s$	$(F/A)_s$	Fuel octane rating	
					Liquid, kJ/kg · K	Vapor c_p, kJ/kg · K						RON	MON
Practical fuels[e]													
Gasoline	$C_nH_{1.87n}$(l)	∼100	0.72–0.78	350	2.4	∼1.7	46.3	43.0	2.76	14.6	0.0685	91–99	82–89
E10 gasoline	$C_nH_{1.94n}O_{0.043n}$(l)	∼100	∼0.75	400	2.4	∼1.7	44.5	41.5	2.82	13.7	0.0730	96[e]	85[e]
Light diesel	$C_nH_{1.8n}$(l)	∼170	0.78–0.84	270	2.2	∼1.7	46.1	43.2	2.79	14.5	0.0690	—	—
Heavy diesel	$C_nH_{1.7n}$(l)	∼200	0.82–0.88	230	1.9	∼1.7	45.5	42.8	2.85	14.4	0.0697	—	—
Natural gas	$C_nH_{3.8n}N_{0.1n}$(g)	∼18	(∼0.79)	—	—	∼2	50	45	2.9	15.7[f]	0.0637	—	—
Pure hydrocarbons													
Methane	CH_4(g)	16.04	(0.72)	509	0.63	2.2	55.5	50.0	2.72	17.23	0.0580	120	120
Propane	C_3H_8(g)	44.10	0.51 (2.0)	426	2.5	1.6	50.4	46.4	2.75	15.67	0.0638	112	97
Isooctane	C_8H_{18}(l)	114.23	0.692	308	2.1	1.63	47.8	44.3	2.75	15.13	0.0661	100	100
Cetane	$C_{16}H_{34}$(l)	226.44	0.773	358	2.2	1.6	47.3	44.0	2.78	14.82	0.0675	—	—
Benzene	C_6H_6(l)	78.11	0.879	433	1.72	1.1	41.9	40.2	2.82	13.27	0.0753	125	115
Toluene	C_7H_8(l)	92.14	0.867	412	1.68	1.1	42.5	40.6	2.79	13.50	0.0741	120	109
Alcohols													
Methanol	CH_4O(l)	32.04	0.792	1100	2.6	1.72	22.7	20.0	2.68	6.47	0.155	109	89
Ethanol	C_2H_6O(l)	46.07	0.785	900	2.5	1.93	29.7	26.9	2.69	9.00	0.111	109	90
Other fuels													
Carbon	C(s)	12.01	∼2	—	—	—	33.8	33.8	2.70	11.51	0.0869	—	—
Carbon monoxide	CO(g)	28.01	(1.25)	—	—	1.05	10.1	10.1	2.91	2.467	0.405	106	—
Hydrogen	H_2(g)	2.015	(0.090)	—	—	1.44	142.0	120.0	3.40	34.3	0.0292	120–140	—

[a](l) liquid phase; (g) gaseous phase; (s) solid phase.
[b]Density in kg/m^3 at 0°C and 1 atm.
[c]At 1 atm and 25°C for liquid fuels; at 1 atm and boiling temperature for gaseous fuels.
[d]Between 3.5 and 3.7 MJ/m^3.
[e]10% ethanol adds some 2–5 octane numbers: so RON and MON depend on rating of base gasoline.
[f]$(A/F)_s$ influenced by N_2 and CO_2 content.
RON, research octane number; MON, motor octane number.

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