FOURTH EDITION

LIGHT & HEAVY VEHICLE TECHNOLOGY

M J NUNNEY

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Light and Heavy
Vehicle Technology

When theory and practice do not agree, you should examine the facts to see what is wrong with the theory

Charles F. Kettering, General Motors (1934)

Light and Heavy Vehicle Technology

Fourth edition

M.J. Nunney

CGIA, MSAE, MIMI

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Contents

Figure 27.29 Disc pad neglect in service (*Ferodo*)

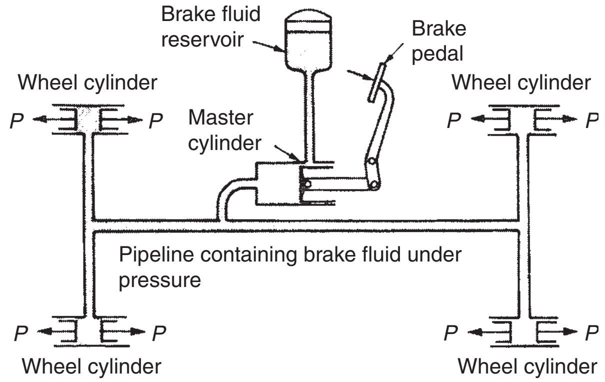


Figure 27.30 Diagrammatic layout of a motor vehicle hydraulic braking system

because the pressure applied on a fluid in a closed circuit equalizes itself throughout that circuit. However, it does not correct for variations in brake torque output arising from any differences in friction between the rubbing elements.

3 With the advent of IFS systems in the 1930s the problems of employing geometrically accurate mechanical linkages to the brakes became more difficult, unless enclosed cables were used. This was because the actuating system had to allow not only for independent but also substantially greater movements of the wheels relative to the car structure owing to the softer springing. The flexible hoses of the hydraulic braking system provided the ideal answer to this requirement.

4 In hydraulic actuation the replacement of mechanical linkages by fluid pipelines offers a greater versatility and ease of installation. Such a system also readily lends itself to the incorporation of a brake servo or booster unit to reduce the operating effort required from the driver.

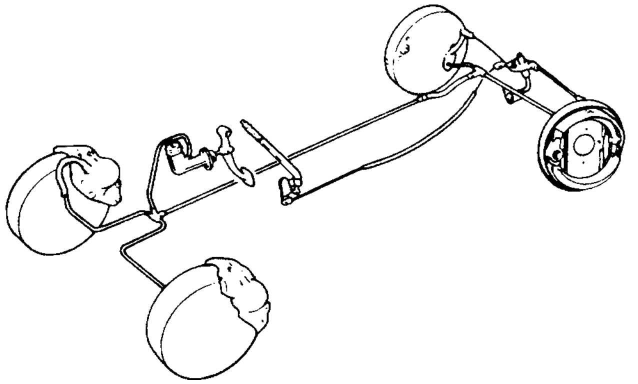


Figure 27.31 A disc front and drum rear, single-line, hydraulic brake installation (*Girling*)

Single-line brake circuit

This represents a very basic hydraulic system in which the emphasis is on the transmission of load, rather than motion, between the driver-operated brake pedal and the friction elements of the wheel brakes. To understand the principle upon which such a system operates, it is necessary to recall the work of the French scientist Blaise Pascal who, in 1658, discovered that pressure on a confined liquid is transmitted equally in all directions and acts with equal force on all equal areas. It thus follows that a perfect pressure balance can be obtained between the actuating units of motor vehicle hydraulic brakes (Figure 27.30), although this does not necessarily mean that the system overall will be perfectly balanced, as will be evident later.

The layout, or hook-up as it is sometimes known, of the now obsolete single-line hydraulic braking system comprises the following components (Figure 27.31).

Master cylinder
For motor cars this unit is usually mounted on the engine side of the body bulkhead structure, through which a push-rod connection is made to a down-hanging or pendant brake pedal that is spring returned to the off or brakes released position. In some light commercial vehicle applications (and earlier car practice) the master cylinder is mounted on the chassis frame where it similarly connects via a push-rod to an up-standing brake pedal. The master cylinder acts as a hydraulic pressure transmitter, because it contains a sealed piston that slides within a cylindrical bore whenever the brake pedal is depressed.

Wheel cylinders
These act as hydraulic slave cylinders in as much as they are pressure receivers containing sealed pistons which slide in response to the movement of the master cylinder piston resulting from the hydraulic pressure developed when the brake pedal is depressed. The wheel cylinders serve either to expand a pair of friction-lined shoes against their brake drum or to clamp a pair of friction-lined pads vice-like against their brake disc. In the latter case, each pad is of sector shape with the object of providing a uniform rate of both heat generation and wear over its surface area. Either drum or disc, or a combination of both types of brake, may be used according to the layout chosen for any particular vehicle.

Connecting pipes and hoses
These serve as hydraulic pressure conduits and connect the wheel cylinders to the master cylinder. The brake fluid is conveyed mainly by rigid metal tubing suitably formed and sometimes plastic coated to follow the installation path on either the body structure or the chassis frame.

Two important requirements of the piping installation are that the tubing must be supported against vibration and protected against corrosion.

Flexible reinforced rubber hoses are used finally to convey the fluid to the wheel cylinders which, in the brake assemblies of conventional layouts, move relative to the vehicle structure owing to suspension and steering movements of the road wheels.

Brake fluid
In the hydraulic brake system, force is transmitted from the brake pedal to the friction elements of the wheel brakes through the medium of a column of virtually incompressible liquid known as brake fluid.

Although this incompressibility is a characteristic of liquids in general, brake fluid is specially formulated to meet additional and stringent requirements as described later. A reserve of brake fluid is contained in a reservoir connected to the master cylinder.

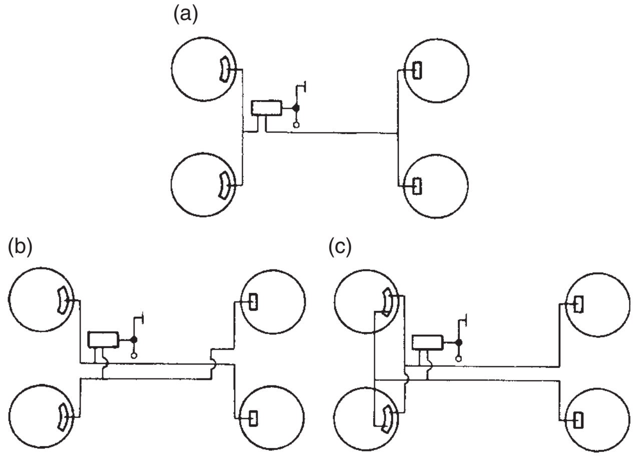


Figure 27.32 Divided-line brake circuits: (a) F/R-split (b) X-split (c) L-split

Divided-line brake circuit

It should be realized that if a serious fluid leakage occurs in the single-line layout the entire hydraulically operated footbrake system can be rendered ineffective. To avoid this potential hazard, the brake system legislative requirements in Europe and the United States now demand a divided-line brake circuit. A failure of either line will still leave two service or wheel brakes in action.

The basic idea of using a divided-line brake circuit is by no means of recent origin. Among some of the early users of the original Lockheed hydraulic brake system there still remained a certain lack of faith in its absolute safety, which could be attributed to the problems of fluid leakage and air ingress that had bedevilled the less successful attempts by other manufacturers to produce a satisfactory hydraulic brake system. This led to a few manufacturers of high-performance cars, commercial vehicles and especially racing cars to adopt a divided-line circuit, so that one line supplied the front brakes and the other one the rear brakes, thereby increasing the safety of the overall system should either line lose its fluid (Figure 27.32a). In practice, this was usually achieved by using a tandem master cylinder, which in effect comprised two master cylinders in line without any fluid communication between them. Less commonly, except on racing cars, two entirely separate single master cylinders were mounted side by side and operated via a balance lever that straddled them. This lever had a restricted movement at its pivot point connection with the brake pedal, so that in the event of one line losing its fluid, the pedal effort could still be transmitted to the other master cylinder.

There were no significant further developments of the divided-line brake circuit until 1963, when the Swedish manufacturer SAAB introduced a diagonally split or X-split brake circuit on their front-wheel-drive SAAB 96. In this system each output from the tandem master cylinder supplies one front brake and its diagonally opposite rear brake, so that in the event of either line failing one-half of the normal maximum braking still remains available from one front and one rear wheel (Figure 27.32b). To maintain car stability under these circumstances it is necessary for the steering geometry to incorporate either centre point steering, or the later developed negative-offset steering, as described in Section 24.2.

Another version of divided-line brake circuit, again from a Swedish manufacturer, was the L-split system introduced on the rear-wheel-drive 1966 Volvo 144. This more elaborate system is so arranged that each output from the tandem master cylinder supplies one-half of each front brake and a single rear brake (Figure 27.32c). In the event of either line failing there is less imbalance of the brakes than with the X-split circuit.

An important advantage of the X- and L-split circuits is that whichever line fails, the reduction in normal maximum braking remains the same; whereas with the simple front-rear split the reduction that accompanies failure of the front line is much greater than if the rear line fails. The divided-line brake circuit can also be split in other ways with varying degrees of elaboration, each with its advantages and disadvantages, but for modern front-wheel-drive cars the X-split circuit appears to have become established practice, while the simple front-rear split circuit is still generally used on rear-wheel-drive cars.

Single master cylinders

It is now necessary to examine in more detail what is happening in the master cylinder not only when the brakes are being applied, but also when they are being released. For this purpose, the construction and operation of single master cylinders used in residual pressure and non-residual pressure hydraulic brake systems must receive consideration.

Residual pressure type of master cylinder

Master cylinders of the residual pressure type maintain a small pressure in the region of 55 kN/m^2 ($8/lbf/in^2$) in the pipelines and wheel cylinders after the brakes have been released.

They were standard equipment in the once conventional all-drum car braking systems, or at least in those where the shoes were positively retracted from the drums when the brakes were released. Master cylinders of this type are now confined mainly to light commercial vehicles employing a similar system of drum brakes. The purpose of maintaining a residual pressure is twofold:

1 To prevent the ingress of air past the wheel cylinder seals when the brakes are released.
2 To ensure a quick response from the brakes during their initial application.

The valve arrangement that maintains that residual pressure also assists in purging air from the hydraulic system when brake bleeding is carried out. This operation becomes necessary when either some part of the system has been temporarily disconnected or after the brake fluid has been renewed.

The construction of a typical residual pressure master cylinder is as follows (Figure 27.33):

Body The main body is bored axially to receive a sliding piston and is drilled radially to provide compensating and supply ports which communicate with either side of the piston head when the brakes are not applied.

Piston This is operated by means of a push-rod connected to the brake pedal and carries two rubber seals, known as the primary and secondary cups. Between these sealing cups the piston skirt is reduced in diameter and the piston head is provided with a series of axial drillings.

Seals The primary cup, against which pressure is generated in the pipelines and wheel cylinders, is acted upon by the piston head but is prevented from adhering to it by a separating washer. The secondary cup is fitted over the flanged push-rod end of the piston to prevent fluid leakage from the open end of the cylinder, which is also protected by a rubber boot dust excluder.

Check valve This is a combination inlet and outlet valve spring loaded against the head of the pressure chamber formed by the master cylinder bore. The same compression spring also serves to return the piston fully against its end retaining ring. Although constructional details vary, a check valve can take the form of an annular rubber cup inlet seal through the centre of which acts a spring-loaded poppet outlet valve.

Fluid reservoir This unit is either formed integrally with, or separately connected to, the master cylinder. It contains a reserve of brake fluid and affords automatic gravity replenishment for the hydraulic system. To meet the demand for increased vehicle safety, the reservoir is generally made from a transparent plastic material, so that the fluid level is readily visible.

The operation of the residual pressure master cylinder during application and release of the brakes is as follows.

Brake application
When the brake pedal is depressed, the initial movement of the push-rod-operated piston causes the sealing lip of the primary cup to pass over the compensating port and henceforth seal off the pressure chamber of the master cylinder. At the same time, the increasing pressure on the primary cup flattens its separating washer, which thus prevents the seal from being extruded through the drillings in the piston head. As pressure builds up it overcomes the spring loading on the outlet check valve, thereby allowing the passage of fluid under pressure into the pipelines and wheel cylinders to actuate the brake assemblies.

Brake release
When the brake pedal is released, the inlet check valve opens to allow the passage of fluid returning under pressure to the master cylinder by the action of the brake shoes pull-off springs. The valve remains open as long as the pressure in the wheel cylinders and pipelines exceeds the required residual pressure of the system: that is, a pressure sufficient to ensure that the sealing cups in the wheel cylinders are kept expanded against their bores. Before the spring-loaded and rapidly returning piston uncovers the compensating port, the pressure drop in the cylinder allows the primary cup to collapse away from the piston head. The separating washer then bows to allow the passage into the pressure chamber of additional fluid drawn from

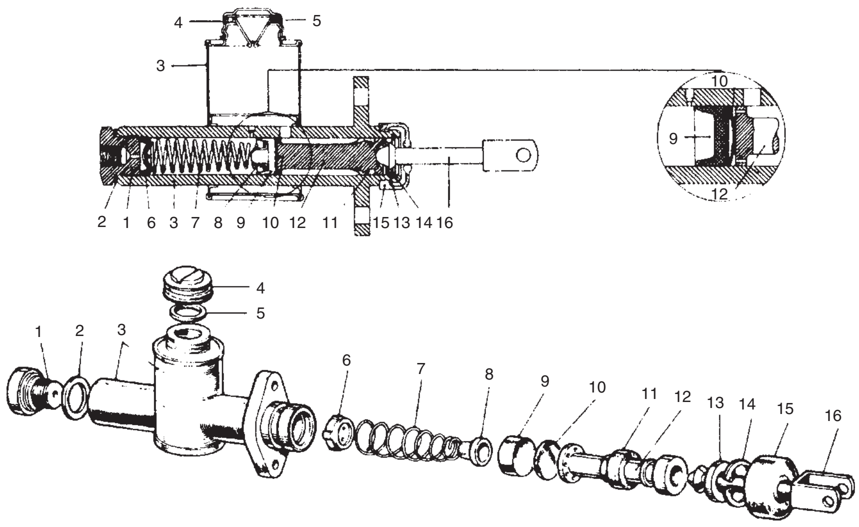


Figure 27.33 Constructional features of a residual pressure master cylinder

1 barrel end plug
2 end plug gasket
3 barrel and tank assembly
4 filler cap
5 filler cap washer
6 check valve assembly
7 return spring
8 return spring retainer
9 main cup
10 piston washer
11 secondary cup
12 piston
13 and 16 push-rod assembly
14 circlip
15 rubber boot

the annular space formed by the reduced skirt of the piston via the holes in the piston head. Fluid is supplied to this source through the supply port of the master cylinder. As the piston finally comes to rest, the primary cup uncovers the compensating port through which the additional fluid drawn into the pressure chamber is released into the reservoir.

Since the quantity of fluid in the wheel cylinders and pipelines is subject to heating and cooling, another function of the compensating port is to allow fluid to flow between the master cylinder and reservoir as it expands or contracts when the brakes are not being used.

Non-residual pressure types of master cylinder

With master cylinders of the non-residual pressure type, the only pressure remaining in the pipelines and wheel cylinders after the brakes are released is a hydrostatic one due to the head of brake fluid in the system. Here it should be appreciated that the *hydrostatic pressure* exerted by a column of liquid is measured in terms of its contained height or *head* and is not related to its cross-section.

The effect of this hydrostatic pressure is such that with drum brakes it can be sufficient to maintain the shoes in light rubbing contact with their drums, while with disc brakes it is insufficient to hinder the elastic recovery of the wheel cylinder seals retaining the pads in close proximity to their discs. To facilitate the bleeding operation, non-residual pressure master cylinders may still incorporate a valve similar to that used in residual pressure versions but modified with a permanent bleed passage to permit final equalization of pressure in the wheel cylinders, pipelines and master cylinder.

Several different constructions of non-residual pressure master cylinders may be encountered. These are the moving seal types with either radial or axial compensating ports, and the stationary seal type with radial compensating ports. A modified check valve with a permanent bleed passage may be fitted in some cases, its sole purpose being to prevent aerated fluid from entering the master cylinder when the brake pedal is pumped during bleeding of the system (Figure 27.34b).

Moving seal with radial compensating port
In both construction and operation this type of master cylinder is similar to its residual pressure counterpart, but differs from it in that the check valve is either omitted or is retained and provided with a permanent bleed passage (Figure 27.34a).

Moving seal with axial compensating port
In this alternative construction, the compensating port is situated at the head of the master cylinder pressure chamber and is controlled by a centre valve with rubber seal (Figure 27.35). A side outlet connection to the wheel cylinders is provided towards the head of the pressure chamber.

When the brake pedal is depressed, the initial movement of the master cylinder piston relieves tension on the centre valve pull-rod and allows the spring-loaded valve to close off the compensating port. Continuing movement of the sealed piston then serves to increase fluid pressure in the master cylinder, pipelines and wheel cylinders. When the brake pedal is released the system pressure falls as the piston retreats and finally intercepts the centre valve pull-rod. This overcomes the closing spring load on the centre valve, which then opens away from the compensating port so that communication is established between the master cylinder and reservoir.

Stationary seal with radial compensating ports
This form of construction is an inversion of the others, because the master cylinder piston moves through a stationary primary or recuperating seal to generate pressure in the hydraulic system (Figure 27.36). The piston carries a secondary sealing cup to prevent fluid leakage from the open end of the cylinder. A series of compensating ports is drilled radially through the hollow front portion of the piston. The primary seal is in fact permitted a very limited end float relative to its seating and is also separated from it by a special shim washer. A series of axially drilled holes is incorporated in the seating immediately behind the shim.

When the brake pedal is depressed, the initial movement of the piston allows its compensating ports to move beyond the primary seal lip. Further movement of the piston generates an increasing pressure in the master cylinder, pipelines and wheel cylinders. The shim washer prevents the pressurized seal from being extruded through the drilled holes in its seating.

When the brake pedal is released, the spring-loaded and rapidly returning piston can produce a pressure drop in the cylinder which allows the primary seal to float slightly

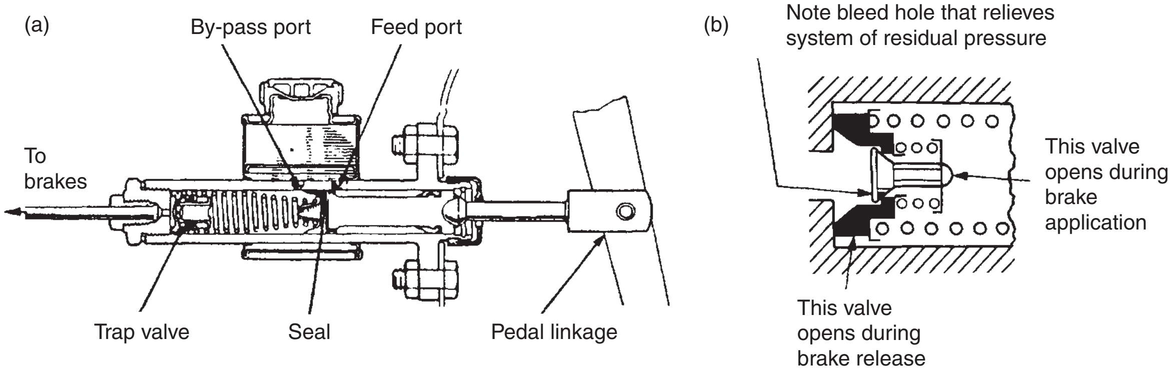


Figure 27.34 (a) Constructional features of a non-residual pressure master cylinder with radial compensating port (b) another form of check or trap valve that facilitates brake bleeding

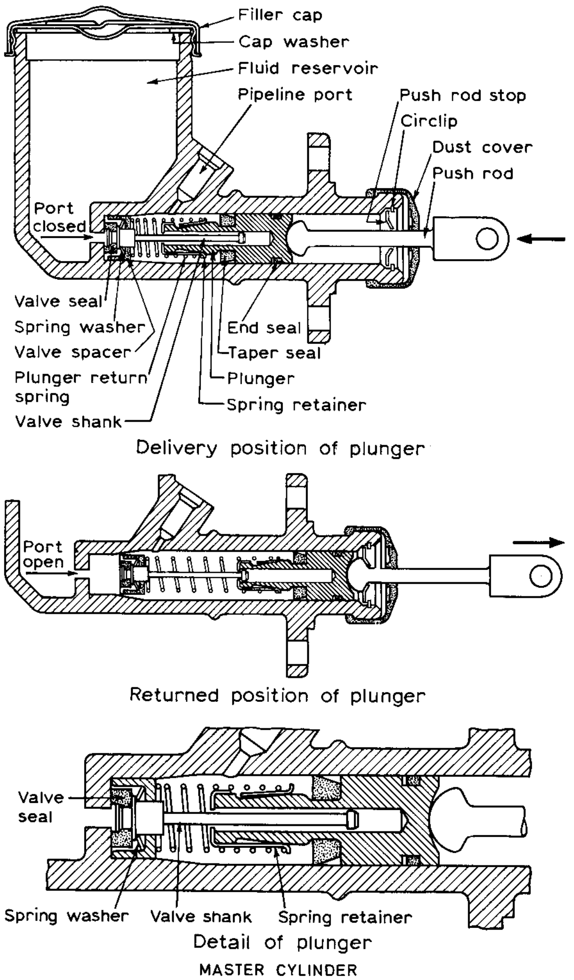


Figure 27.35 Construction and operation of a non-residual pressure master cylinder with an axial compensating port (*Girling*)

forward of its seating so that additional fluid then passes into the pressure chamber. As the piston finally comes to rest, its compensating ports are uncovered by the primary seal and the additional fluid is then released to the reservoir.

An advantage of the second and third forms of master cylinder construction is that the primary seal lip does not have to pass over a compensating port drilled in the wall of the pressure chamber, which action may eventually lead to nibbling of the seal. A further advantage of the third form of master cylinder is that the primary seal acts against a sliding male surface to minimize seal wear, it being found more economical to develop the high standard of surface finish on the piston than in its bore.

Tandem master cylinders

Tandem master cylinders vary in detail construction and especially in respect of their compensating port control. Basically they comprise a common housing that is usually produced from cast iron for optimum service life, in the cylinder bore of which slide two spring-returned pistons with rubber seals. These pistons may either be completely independent in their movements (Figure 27.37) or be linked together such as to have only limited independence (Figure 27.38) both arrangements being able to compensate automatically for any unequal displacement of fluid in operation. The piston that resides near the mouth of the cylinder bore is conventionally referred to as the primary piston and the other one as the secondary piston, neither term intending to imply the normal operating sequence of the pistons. According to design requirements, the return travel of the secondary piston may be limited by a stop pin or screw. The operating chambers of the two pistons are supplied with fluid from either an integral or a separately attached reservoir, the latter being produced from a translucent plastic moulding to provide a ready visual indication of fluid levels. In both reservoir constructions an internal vertical baffle is used to preserve independence of fluid supply for each line of the divided brake circuit.

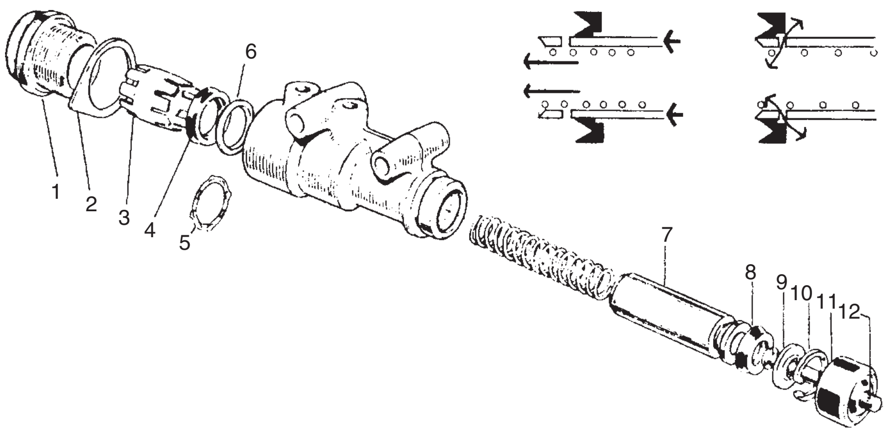


Figure 27.36 Construction and operation of a stationary seal type of master cylinder (*Girling*)

1 end cap
2 gasket
3 seal support
4 recuperating seal
5 steel shim
6 nylon backing ring
7 plunger
8 end seal
9 washer
10 circlip
11 dust cover
12 push-rod

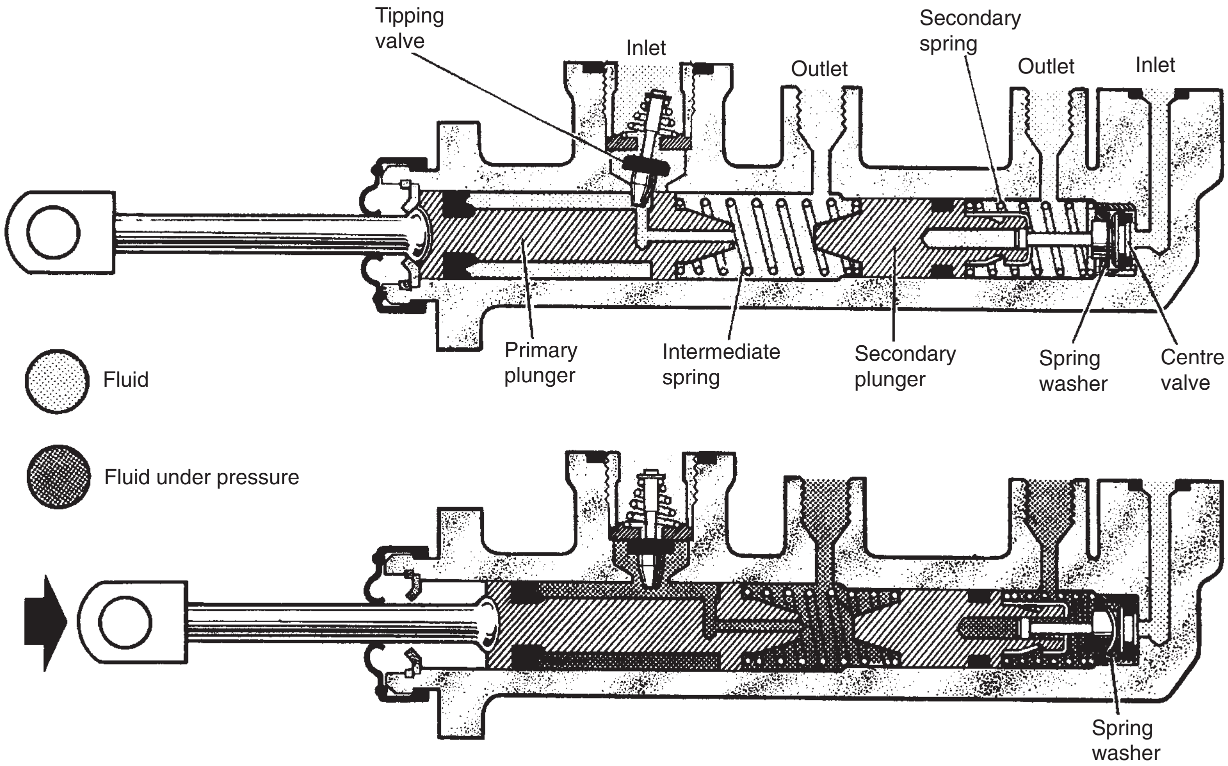


Figure 27.37 Tandem master cylinder with tipping and centre valves (*Girling*)

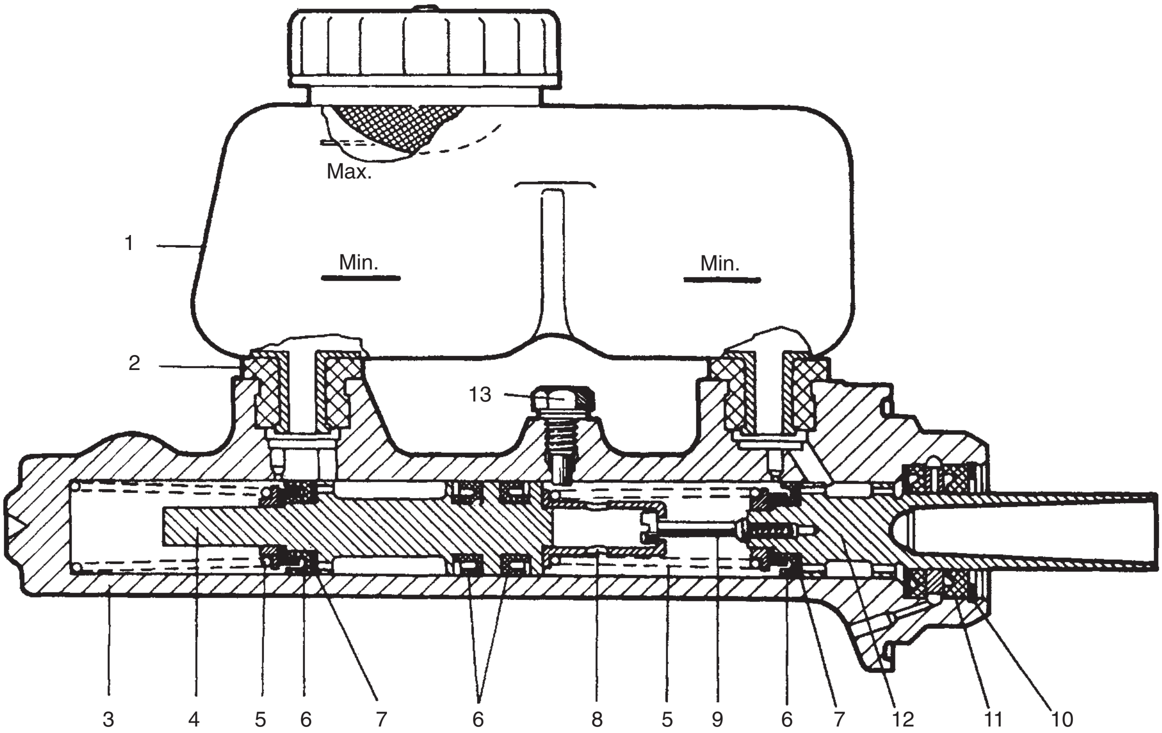


Figure 27.38 Tandem master cylinder with linked pistons (*ATE-SAAB*)

1 container
2 rubber seal
3 housing
4 secondary piston
5 spring
6 piston seals
7 washer
8 sleeve
9 screw
10 lock ring
11 seal ring
12 primary piston
13 screw

When the brakes are applied in normal operation, the movement of the master cylinder push-rod urges the primary piston forwards. This action closes the compensating port for the primary pressure chamber, either as a direct result of piston seal movement, or by piston movement allowing a tipping valve to reseat. The ensuing rise in pressure ahead of the primary piston causes a similar forwards movement of the secondary piston. This action likewise closes the compensating port for the secondary pressure chamber, either again as a direct result of piston seal movement, or by piston movement in this case allowing a centre valve to reseat. The same pressure is therefore generated ahead of both pistons and therefore in each line of the divided circuit.

In the event of pressure being lost in the line served by the primary piston, this piston will continue to move forwards until it contacts the secondary piston, thereby ensuring that pressure in the other line is maintained. Conversely, if pressure is lost in the line served by the secondary piston, this piston will move forwards until it bottoms against the blind end of the cylinder, so that pressure build-up between the primary and secondary pistons is still available in the other line. Clearly, the travel of the master cylinder push-rod will become greater whichever line of the divided circuit suffers loss of pressure, but in such an emergency it will still be capable of transmitting force to the piston serving the undamaged line and therefore actuate the respective brakes hydraulically.

The earlier reference to linking of the primary and secondary pistons in some tandem master cylinder constructions deserves further explanation. The advantages claimed for this feature are twofold. First, it enables the primary piston spring to be preloaded in position so that initial movement of the primary piston is transmitted without relative compression to the secondary piston, thereby closing both compensating ports simultaneously and allowing the same rate of pressure rise in each line. And second, the linking anchor screw prevents the weaker spring of the secondary piston from remaining compressed when the brakes are released, since otherwise the compensating port for the secondary pressure chamber could remain closed owing to incomplete return of the piston.

A pressure differential warning actuator may be required with tandem master cylinders (Figure 27.39). It comprises a self-centring shuttle valve that becomes hydraulically unbalanced in the event of either line losing pressure, when it moves to complete an electrical circuit and illuminate a facia-mounted warning light.

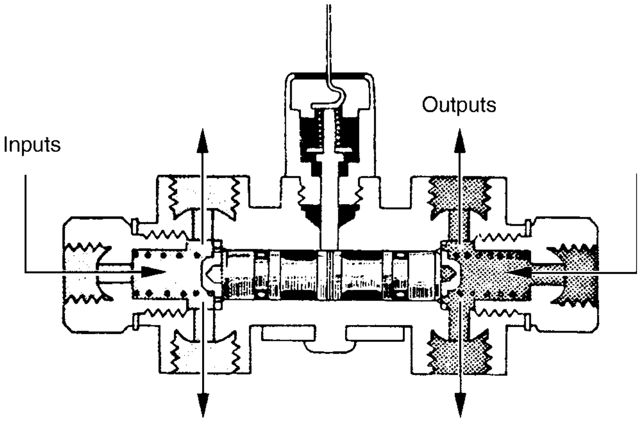


Figure 27.39 Differential pressure warning actuator

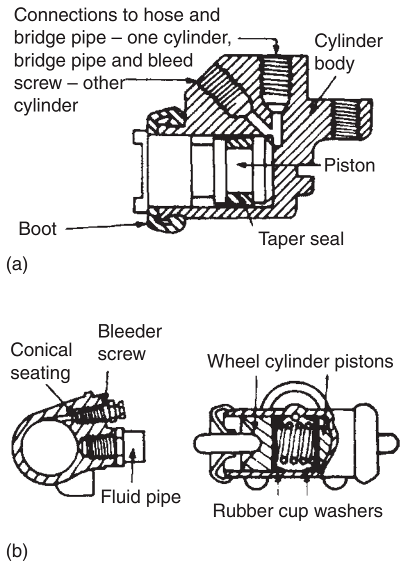


Figure 27.40 Hydraulic wheel cylinders: (a) single (b) double piston

Wheel cylinders for drum brakes

The construction of a typical single-piston wheel cylinder is as follows (Figure 27.40a):

Body The wheel cylinder body is usually spigot located into the brake back plate assembly and retained by a pair of setscrews. It has a blind-ended cylinder bore to receive the sliding piston and is drilled radially to provide threaded connections for the fluid pipeline and bleeder screw.

Piston The outer end of the piston engages the toe of the adjacent brake shoe either directly or indirectly through the medium of a short push-rod. It is usually spring loaded against the brake shoe at all times.

Seals A pressure seal is provided at the inner end of the piston and may take the form of a simple cup washer abutting against the piston head, or an annular sealing ring engaging with a groove in the piston body. In the former case a cup filler, or less commonly a spreader washer, is interposed between the piston seal and return spring to maintain the cup lip in contact with the cylinder wall. The cup filler also reduces the working volume of fluid in the cylinder that is subject to heating. A rubber sealing boot is fitted to exclude dust from the open end of the cylinder.

In operation, depression of the brake pedal displaces fluid from the master cylinder through the connecting pipes to the wheel cylinders. The additional brake fluid entering each cylinder pushes the sealed sliding piston outwards, a movement transmitted simultaneously to the adjacent brake shoe which is thus forced against the drum. The greater the pedal effort exerted by the driver the higher will be the pressure generated in the wheel cylinder, as will be the piston force

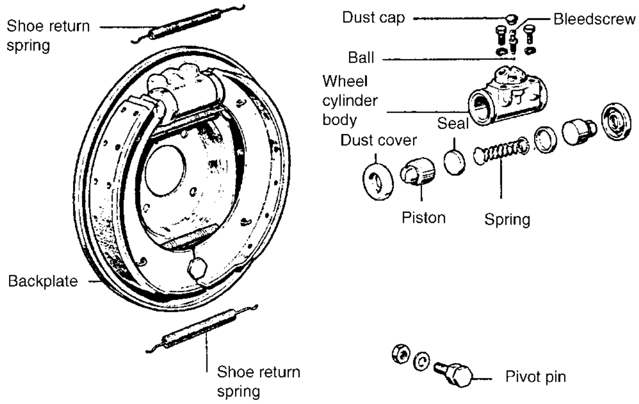


Figure 27.41 Hydraulically operated leading and trailing shoe drum brake (*Girling*)

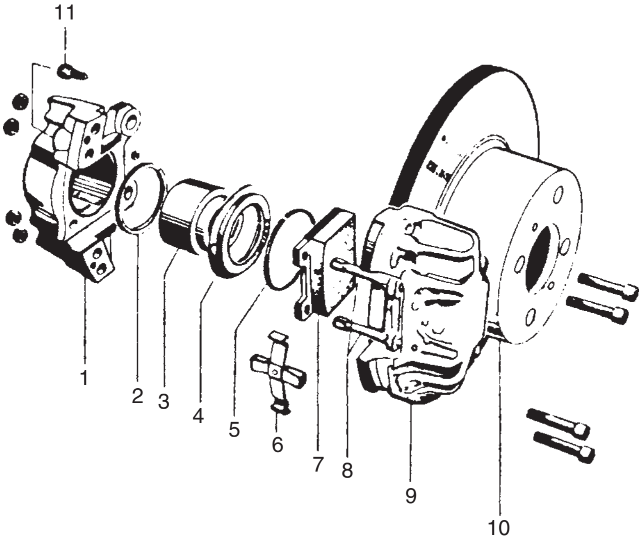


Figure 27.42 The hydraulically operated disc brake in practice, front-wheel assembly (*ATE-SAAB*)

1 brake housing half
2 piston seal
3 piston
4 gasket
5 gasket retainer
6 retaining spring
7 brake pad
8 locking pins
9 brake housing half
10 brake drum and disc
11 bleeder screw

transmitted to the brake shoe. When the brake pedal is released, the brake shoe pull-off springs return the piston to its initial position in the wheel cylinder and this forces excess brake fluid back to the master cylinder.

Although a single-piston wheel cylinder is usually single acting, in some brakes it is made double acting by slidably mounting its body on the back plate, as described later (Figure 27.54).

The detail construction and mounting of a typical double-piston wheel cylinder (Figure 27.40b) is similar to that of the single-piston unit, apart from the obvious differences of the body being through bored and the duplication of its piston and sealing arrangements. Likewise in operation, both pistons are pushed outwards against their adjacent brake shoes, because the wheel cylinder is double acting. The pistons are returned to their initial position in the usual manner by pull-off springs acting on the brake shoes (Figure 27.41).

Wheel cylinders for disc brakes

We take for example fixed caliper designs. Two opposed hydraulic cylinders are incorporated in either a single- or two-piece caliper construction which straddles the disc and is attached to the axle flange. In the former case, the cylinder bores are machined through from one leg of the caliper to a blind end in the other leg, a screwed plug being used to seal the open-ended cylinder. Two-piece caliper constructions are generally favoured in modern practice so that blind-ended cylinder bores can be machined into each leg of the caliper, thereby dispensing with the need for a screwed sealing plug (Figure 27.19 and 27.42).

A hydraulic pressure balance between the two opposed cylinders is obtained either by an external bridging pipe or internal drillings through the caliper bridge itself. The sliding movement of each hollow piston in its cylinder is sealed by a stationary rubber ring retained within a groove near the rim of the cylinder. By virtue of its enclosed air space, a hollow piston minimizes the amount of heat transferred to the brake fluid from the pad backplate during braking. The operation of the disc brake is similar to that of the drum brake except, of course, that the thrust forces developed at the wheel cylinder pistons arise from pressing the pad liners into rubbing contact with the brake disc. Thin steel shims with cut-outs are generally used between the piston and pad back plate, so as to secure the best distribution of pressure on the friction lining and avoid brake squeal.

It was earlier inferred in Section 27.2 that the disc brake is virtually non-self-energizing; but why only virtually and not totally, considering that we are simply bringing together two plane surfaces with a clamping load applied normal to them? In fact a mild self-energizing effect inevitably arises from the thickness of the friction pad, which imposes an offset between its rubbing surface and the abutment for its back plate. During braking the frictional drag developed between the pad and its disc therefore creates a turning moment about this offset (Figure 27.43), so that the leading edge of the pad makes a relatively harder contact against the disc than the trailing edge, even though the pad receives a centrally applied clamping load. Of greater significance than the mild self-energizing effect is the unequal wearing over the length of each pad and its influence on pedal feel. In a more recent Japanese disc brake installation this problem has been countered by providing each front caliper with four opposed pistons, the pair of pistons towards the leading edge of the pads being smaller in diameter than the pair towards the trailing edge (Figure 27.44). This arrangement is claimed to confer a better pedal feel under all braking conditions.

Wedge-operated brake expanders

In a wedge-operated brake expander mechanism, the lever and rotary cam method of transferring the actuating force to the brake shoe is replaced by a sliding wedge and tappets (Figure

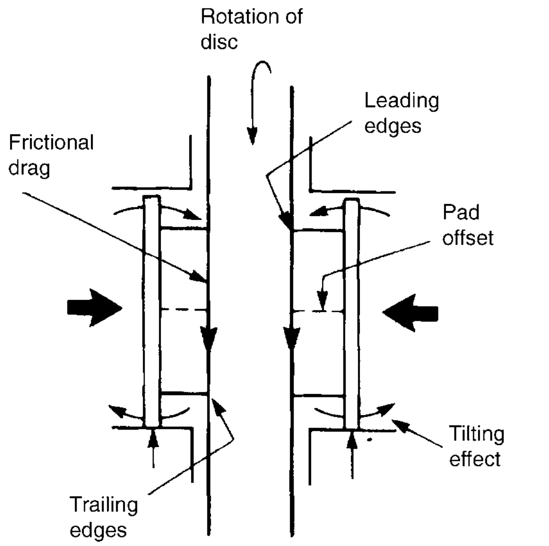


Figure 27.43 Tilting effect on disc brake pads during braking

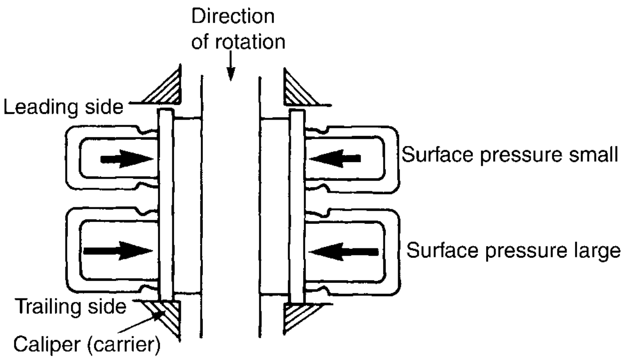


Figure 27.44 Disc pad apply pistons graduated in diameter for better pedal feel (*Colt Mitsubishi*)

27.8). Both the sliding wedge and tappets are of cylindrical form, the difference between them being that the wedge has a conical waisted portion whereas the tappets each have an inclined end face. The action is simply that of drawing the wedge partially through the tappets, which directly engage the toes of the brake shoes and thus bring the friction liners into contact with the drums.

Expander mechanisms of this type offer the following advantages:

1 They possess a high mechanical efficiency, because the friction between the wedge and tappets can be reduced to a minimum by interposing rollers.
2 The wedge expander unit can be easily lubricated and sealed, thus enabling it to retain its efficiency.
3 It can readily provide a floating-cam effect, either by allowing the expander body to slide on the back plate or by incorporating a sliding insert in the wedge.
4 By virtue of acting at right angles to the plane of the brake shoes, wedge expanders are convenient to use with mechanical (especially the handbrake) hydraulic and compressed air braking systems.

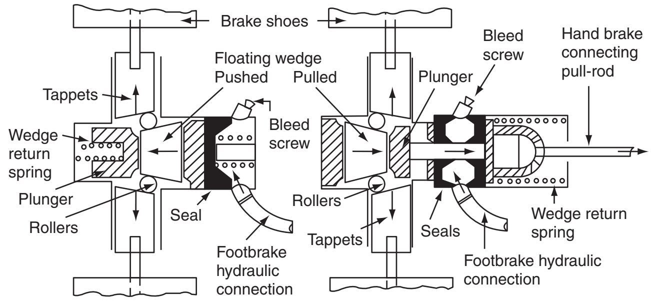


Figure 27.45 Pusher and puller hydraulically operated wedge expanders in diagrammatic form

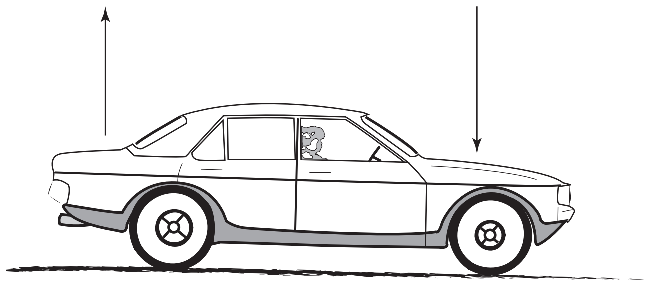


Figure 27.46 Weight transfer during braking

A possible disadvantage of the wedge expander unit has sometimes been its tendency towards sticking, i.e. not returning fully to the off position following release of the brakes.

Although wedge-type expanders were initially developed for mechanical braking system applications on cars, they were eventually supplanted by the purely hydraulic system with which they were originally intended to compete. However, wedge-type expander units continued to find application on commercial vehicles. For medium duty they are hydraulically operated with transversely mounted pusher wheel cylinders on the front brakes and similarly mounted puller wheel cylinders on the rear brakes, the latter arrangement providing convenient lost-motion pull-rod connections to the handbrake system (Figure 27.45). For application to heavy vehicles the wedge-type expander is used in conjunction with compressed air operation as described in Section 28.

Brake apportioning valves

To achieve the shortest emergency stopping distance, all four wheels of a car need to be almost on the point of locking, so that in theory the braking force developed at the road surface level should always be divided between the front and rear wheels in the same ratio as their respective vertical loading. In practice this ideal state of affairs is difficult to attain, because apart from being dependent on the position and amount of load carried, the relative vertical loading is also influenced by the transfer of weight from the rear to the front wheels during actual braking (Figures 19.41 and 27.46). Since this dynamic weight transfer is proportional to the deceleration of the car, it follows that in a simple conventional hydraulic braking system with a fixed ratio of braking to the front and rear wheels, the ratio chosen will be correct

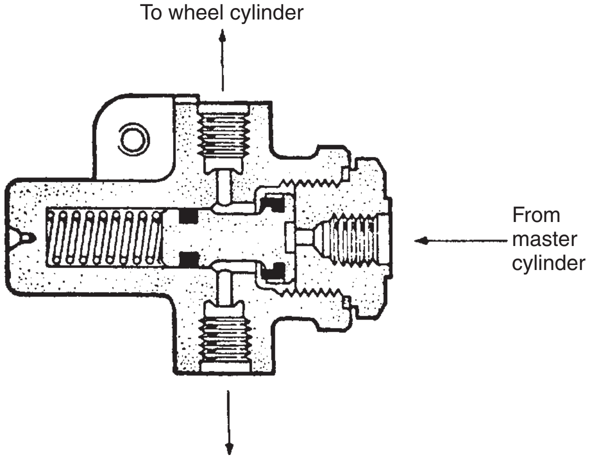


Figure 27.47 Lockheed pressure limiting valve

at one rate of deceleration only. Whereas such a fixed ratio, which usually approximates to the static distribution of weight between the front and rear wheels, is quite satisfactory for most normal braking conditions, under emergency braking the rear wheels could lock prematurely and seriously affect car controllability.

An established approach to reducing this problem, especially on nose-heavy front-wheel-drive cars, has been to impose an additional control over the line pressure supplied to the rear brakes. This may be accomplished by the following types of sensing valve:

Pressure limiting valve Under light braking this type of valve allows the free passage of fluid under pressure to the rear brakes in the normal manner (Figure 27.47). With heavier braking, once the fluid pressure exceeds a predetermined maximum it overcomes the return spring loading on the valve plunger and allows it to move forwards. This seals off the two outlets to the rear brakes, thereby preventing them from locking prematurely, any further increase in system line pressure then being transmitted to the front brakes only.

Pressure reducing valve This represents a refinement of the simple pressure limiting valve, because it maintains a constant braking force at the rear wheels despite thermal expansion of their brake drums during prolonged heavy braking. Under these conditions, a ball valve is unseated by the plunger and allows fluid to pass until the dropping pressure trapped in the rear brake line has been restored to its predetermined maximum, thereby reducing any tendency for the rear wheels to lock prematurely.

Inertia conscious valve A simple version of this type of deceleration sensitive valve consists of a body, a steel ball and a valve that is lightly spring returned. The angular mounting of the valve body is so contrived that under light braking the ball remains at the lower end of its inclined ramp and maintains the valve open, thereby allowing the free passage of fluid under pressure to the rear brakes (Figure 27.48). With heavier braking the inertia force acting on the ball causes it to roll up its bore and close the valve. This seals off the outlet to the rear brakes so that again any further increase in line pressure

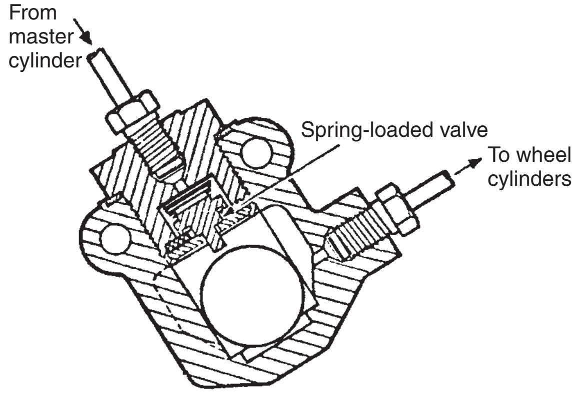


Figure 27.48 Lockheed inertia valve

will be transmitted to the front brakes only. The inertia conscious valve also provides automatic correction of braking ratio on down or up gradients.

Load conscious valve Valves of this type are typically pressure reducing in construction, but connect to the rear axle via a lever arm and flexible link. Relative changes in ride height are thus gradually transmitted to the valve plunger, which can then reposition itself so that the proportion of line pressure received by the rear brakes will be either increased or decreased to reflect loading conditions on the rear axle. This type of pressure modulating valve is particularly suited to commercial vehicle application, where variations in the distribution of static load are usually more important than weight transfer during braking.

Brake system apportioning valves must be compatible with divided-line brake circuits, so that two rear apportioning valves can be required for an X-split circuit (Figure 27.32). They must also be compatible with anti-lock brake systems (Figure 29.6) and may be electronically operated in conjunction with the system itself.

27.5 HYDRAULIC BRAKE FLUIDS

The properties of brake fluids

Although synthetic fluids in their various formulations are no strangers to the hydraulic equipment engineer, that fluid which is used in the hydraulic braking systems of most motor vehicles has long been regarded with a certain amount of curiosity by automotive service engineers. This is probably because the manufacturers of brake fluids do not normally volunteer the chemical composition of their products, but in any event such information can be meaningful only to a lubrication chemist. However, since we are required to have some knowledge of the constituents and properties of brake fluid, the following basic information should suffice.

Constituents

A conventional polyglycol ether brake fluid is socalled because it includes the following chemical components:

1 Glycols to ensure that the fluid is compatible with the rubber materials in the braking system.

2 Glycol ethers to enable the fluid to flow freely and therefore have the desired viscosity.
3 Polyglycols to confer the required lubricity on the fluid.
4 Corrosion inhibitors to protect the metal parts of the braking system.

Freezing and boiling points
A typical high-quality brake fluid would remain chemically stable over a temperature range of −65°C to +220°C.

Viscosity
The maximum acceptable viscosity of a brake fluid is limited by its satisfactory behaviour at −40°C. That is, the brakes should not respond sluggishly during either application or release at this low temperature.

Specifications
Since 1945 the Society of Automotive Engineers (SAE) in America has issued specifications relating to the minimum required performance of brake fluids, as also, does the Department of Transportation (DOT) in that country. More recently the European International Standards Organization (ISO) has issued its standard on brake fluid. These specifications are internationally recognized and are therefore generally labelled on the sealed container in which the fluid is sold, being expressed as SAE J1703, DOT 3,4 and 5, and ISO 4925 for conventional brake fluids of the polyglycol ether and the later developed borate ester types. The specifications are similar and differ only in respect of low temperature viscosity, minimum 'dry' boiling point of the pure brake fluid, and minimum 'wet' boiling point of the brake fluid after treatment with moisture. As a matter of background interest, a DOT 3 brake fluid has minimum dry and wet boiling points of 205°C and 140°C respectively, which compare to 230°C and 155°C for a DOT 4 fluid. They are periodically revised to take into account the increasing severity of duty imposed upon brake fluids, especially in terms of heat input not only from application of the brakes themselves as car performance has risen, but also from higher underbonnet temperatures where some of the brake equipment is housed. A high quality brake fluid may, of course, be formulated to surpass the requirements of these specifications.

Uncommonly a fluid with a mineral basis may be specified, such as the LHM (Liquide Hydraulique Minérale) fluid used in the central hydraulic system that includes the brakes on certain Cirtroën models. This fluid is identified with a green dye and cannot be mistaken with other coloured dyes for conventional brake fluid. It is vitally important to consult the vehicle manufacturer's recommendations in respect of brake fluid, because only the brake fluid specified for a given system is fully compatible with the rubber components and metals in contact with the fluid. The result of using incorrect fluid could be malfunctioning of the brakes with potentially disastrous consequences.

A brief mention must also be made of silicone based brake fluids, which are sometimes used for specialized applications. Brake fluids of this type are non-hygroscopic (a term explained later) and can therefore meet a DOT 5 specification by virtue of having a higher minimum wet boiling point. They also possess good low-temperature viscosity characteristics, but their bulk compressibility is greater and may create a spongy feel to brake pedal operation. Silicone based brake fluids are not compatible with conventional brake fluids.

The enemies of brake fluid

The satisfactory operation of a hydraulic braking system must ultimately depend upon the integrity of its brake fluid, which can deteriorate in the presence of air, water and dirt for the following reasons:

Air Pure brake fluid is virtually incompressible. If it contains air then the fluid mass becomes compressible to an extent corresponding to the amount of air entrained. This results in the brakes becoming less responsive during application and is indicated by a combination of excessive travel and spongy feel of the brake pedal.
Water Conventional (as opposed to mineral based) brake fluid is by its nature hygroscopic, which simply means that is tends to absorb moisture from the surrounding air. As a result the boiling point of the fluid can be seriously lowered, especially when it is considered that a water content of 5 per cent by weight will virtually halve the boiling point, and this may occur after three years' service life. The important point to appreciate is that if during severe usage of the brakes a fluid reaches its boiling point, gas pockets are formed in the top of the wheel cylinders. At worst the presence of this vapour will, between brake applications, force fluid back into the reservoir until suddenly insufficient fluid can be trapped between the master cylinder and wheel cylinder pistons to operate the brakes at all. It is to avoid such a dangerous situation as this that brake equipment manufacturers normally recommend the system fluid be changed every 18 months to 2 years.
Dirt This is, of course, the enemy of all hydraulic systems, where consistent operation depends upon maintaining fine clearances and effective seals. For this reason scrupulous cleanliness must be exercised when servicing hydraulic brake systems.

Removing air from hydraulic brakes

The purging of air from a hydraulic brake system is commonly referred to as bleeding the brakes (Figure 27.49). Unless the presence of air is indicated by a spongy feel of the brake pedal, bleeding should only be required after some part of the system has been disconnected, or following renewal of the brake fluid.

For the purpose of bleeding the brakes a series of air bleed valves with bleeder screws are employed, which communicate with the interior chambers of the various hydraulic units at their uppermost points. These units include in all cases the wheel cylinders of drum and disc brakes and in some cases the master cylinder as well. The expulsion of air from each unit is achieved by attaching what is termed a bleeder tube to the particular air bleed valve and immersing its other end in a small quantity of new fluid contained in a glass jar. If then the

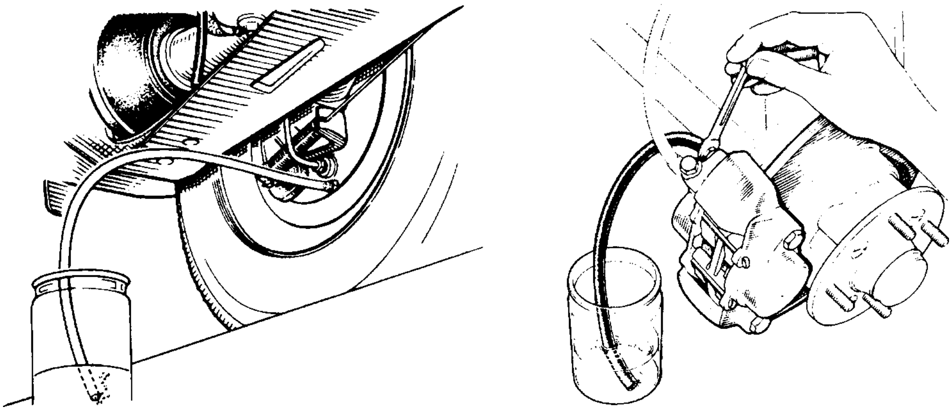


Figure 27.49 Brake bleeding (*Lockheed*)

air bleed valve is opened a fraction of a turn and the brake system pressurized either manually by repeatedly depressing the brake pedal, or automatically by means of a commercial pressure bleeding unit connected to the master cylinder reservoir, any air leaving the system will be made evident by the bubbles rising in the jar of fluid. When the bubbles no longer appear the bleeder screw of the air valve is retightened on to its seating. The procedure is, of course, repeated at the other air bleed valves in the system.

Not so many years ago, in the era of the very basic single-line hydraulic braking system described earlier, it would have been possible to include here a simple set of instructions that would have covered the brake bleeding procedure for practically all vehicle applications. Nowadays matters are a good deal more complicated, because from safety and other considerations the modern hydraulic braking system has reached a high level of sophistication. This in turn has led to many variations in brake bleeding procedures, which can require different pedal actions during manual bleeding, different sequences of bleeding that relate to divided-line safety systems, and with certain of the latter simultaneous bleeding of the front and rear brakes. Furthermore, there are additional factors that usually have to be taken into account when bleeding anti-lock brake systems (ABS) (Section 29), and especially those systems where a pump charged accumulator provides a reserve of pressurized fluid. Also, ABS system modulators must never be drained of fluid, because as with other hydraulic equipment of this nature, there can be difficulty in purging them of air when the system is refilled, and it is for this reason that replacement modulator units are supplied already primed with fluid and temporarily sealed. It is therefore most important that the manufacturer's instructions are faithfully adhered to when carrying out brake-bleeding operations.

Caution Care should be taken in the handling of brake fluids, since they contain agents that damage painted surfaces. They are also toxic to varying degrees, so that contact with sensitive parts of the body must be avoided. Medical advice must be sought immediately if brake fluid is inadvertently swallowed.

27.6 VACUUM SERVO-ASSISTED BRAKING

General background

Until the increasing weight and speed of commercial vehicles dictated the use of power-operated compressed air brakes, as opposed to power-assisted vacuum brakes, the latter proved to be a competent and reliable system of braking. In brief, the installation comprised a master servo, two front wheel servos and a vacuum reservoir. The piston-operated master servo converted vacuum, created by either manifold depression or exhauster pump according to whether the vehicle was petrol or diesel engined, into power for assisting brake application. It was so designed that its output was proportional to the effort applied at the brake pedal. The master servo was incorporated in a simple mechanical linkage from the brake pedal to the rear brakes, and thus assisted the driver's effort. There was no mechanical connection between front and rear brakes, the two front wheel servos being mounted on the stub axles and communicating via flexible vacuum hose pipes directly with the master servo. As the degree of vacuum admitted to the cylinder of the master servo was proportional to the driver's effort on the brake pedal, the same degree of vacuum necessarily existed in the wheel servos to apply the front brakes. Hence, the mechanically operated rear brakes and the vacuum-operated front brakes were servo assisted. The vacuum reservoir had sufficient capacity to provide at least four brake applications with the engine stationary. A non-return valve was used to allow air to be drawn from the vacuum reservoir, but not in a reverse direction. Where used the exhauster pump was typically of the sliding vane rotary type, and was engine driven either via the timing gears or in tandem with the fuel injection pump, or alternatively belt driven from the engine or the gearbox.

A pioneer in the application of vacuum servo assistance to the braking systems of both commercial vehicles and cars was the Belgian engineer Albert Dewandre, who first developed the idea in the mid 1920s.

Servo assistance for car brakes

Sir Henry Royce is credited with having once said that 'the speed at which a car can be driven with safety is determined by its ability to stop.' It is therefore perhaps to have been expected that, since the advent of four-wheel braking in the mid 1920s, the braking system has always been one of the outstanding features of Rolls-Royce cars, combining reliable stopping power with light pedal pressures. Although such attributes are now taken for granted in the braking systems of most modern cars, this was certainly not true before the general adoption of servo-assisted disc brakes. It is therefore relevant to try and account for the superiority of the earlier Rolls-Royce braking systems. The explanation may be found, first, in the use of strictly non-self-energizing drum brakes at the wheels, and second, in the addition of a friction servo motor to multiply the effort exerted on the brake pedal by the driver. In other words, the system used low-factor wheel brakes to provide consistent braking, together with a servo mechanism to reduce what would otherwise be a high pedal effort to operate them.

Since these two basic approaches can now be found in the servo-operated disc brake systems of many modern cars, they likewise account for their much improved braking performance, or at least greatly contribute to it. The reasons for the use of a servo unit in the braking system may therefore be summarized as follows:

1 It permits the use of low-factor wheel brakes with less sensitivity to changes in lining friction as they become heated.
2 The driver effort on the brake pedal can be reduced even further than with some high-factor, non-servo-assisted brakes.
3 A reduction can be made in the total pedal travel required because leverage in the system can be made much less.

Modern vacuum servo units

These may be either of the indirect- or direct-acting variety. The former is actuated by hydraulic pressure from the master cylinder and pressurizes the wheel cylinders via a hydraulic slave cylinder. This type of servo unit may therefore be installed in any convenient space available. For example, twin servo units of this type are housed in a front wing compartment of the high-grade Bristol car. A direct-acting (or integral) servo unit is, as its description suggests, directly actuated by the brake pedal and better lends itself on most cars to the modern requirement for dual hydraulic braking systems.

In either event its purpose is to reduce the driver effort required on the brake pedal. It performs this function by exposing a servo piston to the difference between atmospheric and engine intake manifold pressures. The piston is provided with either a sliding or a rolling seal against its cylindrical housing or chamber.

If used in conjunction with a diesel engine, the servo unit must be connected to either an engine or electrically driven vacuum pump. With earlier designs of vacuum servo units, each side of the piston was exposed to atmospheric pressure when the brakes were released and then manifold depression was allowed to act on the leading side of the piston during their application. This was known as the air suspended type of vacuum servo. In established modern practice, however, the vacuum suspended type of vacuum servo is used in which each side of the piston is exposed to manifold depression when the brakes are released and then atmospheric pressure is admitted to the trailing side of the piston during their application.

The reasons for this reversal in roles are threefold:

1 Less demand on the source of depression during brake application provides a more responsive control.
2 Since only the trailing side of the piston need be exposed to the atmosphere there is a reduced risk of damage by the ingress of foreign matter.
3 A simpler valve arrangement may be used requiring only one port each for the vacuum and atmospheric connections.

A fail-safe feature common to all types of vacuum servo-operated braking systems is that, in the event of failure of the vacuum source, the hydraulic circuit(s) remain unaffected although, of course, the driver pedal effort must be increased.

Operation of vacuum suspended servo unit

Referring first to the general arrangement drawing that names the parts of a typical direct-acting servo unit (Figure 27.50) and then to the simplified diagrams (Figure 27.51), the operating principle may be summarized as follows:

1 With the brakes released the servo piston is fully retracted by its return spring. Similarly the input rod connection to the brake pedal is retracted by the return spring for the latter. In this position of the input rod the vacuum port is open and there is a depression acting on both sides of the servo piston, which can thus be described as being suspended in vacuum.
2 When the brakes are applied the pedal urges the input rod forward, so that first the piston control valve closes the vacuum port and then its continued movement opens the atmospheric port. Air then enters the servo unit with the result that atmospheric pressure acts behind the servo piston and assists the input rod in travelling forwards; the piston in turn actuates the output push-rod connecting to the master cylinder plunger. During release of the brakes the air that entered the servo unit is exhausted via the opening vacuum port, first to the front portion of the chamber and thence past the non-return valve to the intake manifold.

Two additional points that deserve mention are the following:

3 When the brakes are held on, as distinct from being applied by increasing effort on the pedal, both vacuum and atmospheric ports remain closed so that a state of equilibrium exists between the master cylinder and the servo unit.
4 The function of the rubber reaction disc is to impart a certain degree of feel or feedback into the pedal effort being exerted by the driver, because when this is sufficient to open the atmospheric port of the piston control valve, it also means that the valve plunger head has penetrated the reaction disc. Since the latter then acts as a return spring to eject the valve plunger, the driver is made aware of this by an opposing force felt at the brake pedal.

Emergency brake assist

An emergency brake assist system can serve a two-fold purpose. First, it provides a built-in means of overcoming the

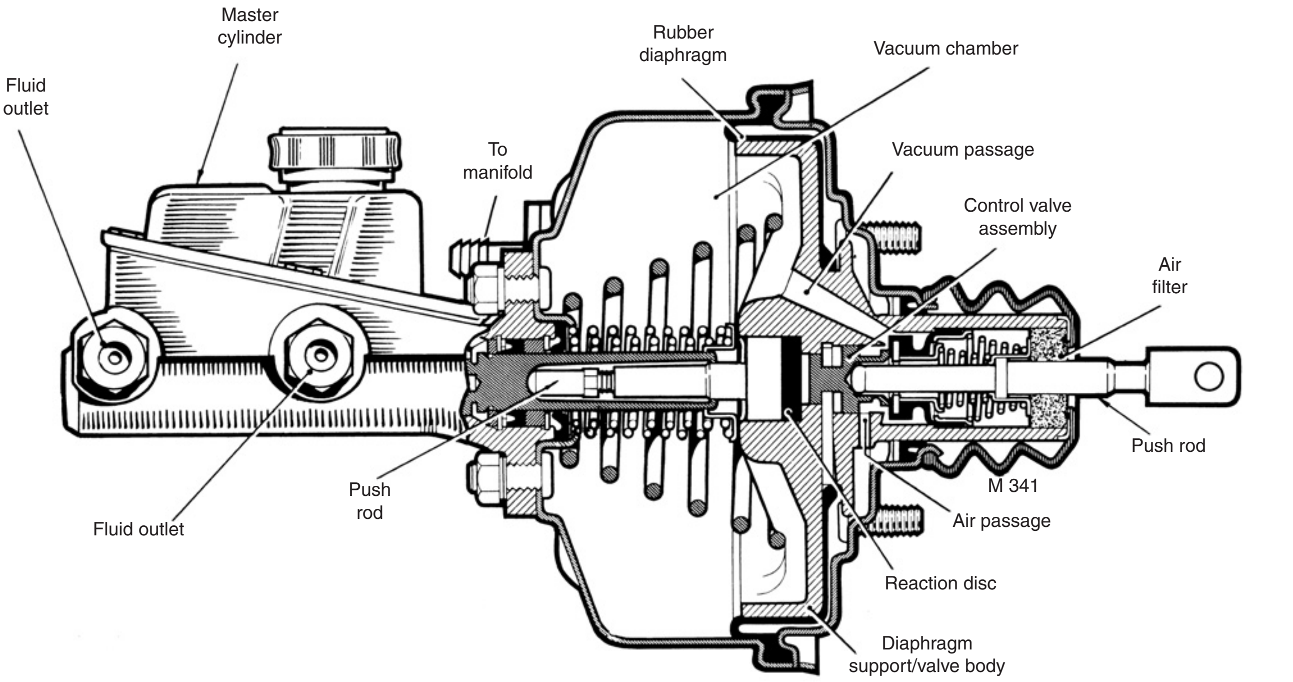


Figure 27.50 Cross-section through a direct-acting suspended vacuum brake servo (*Lockhead*)

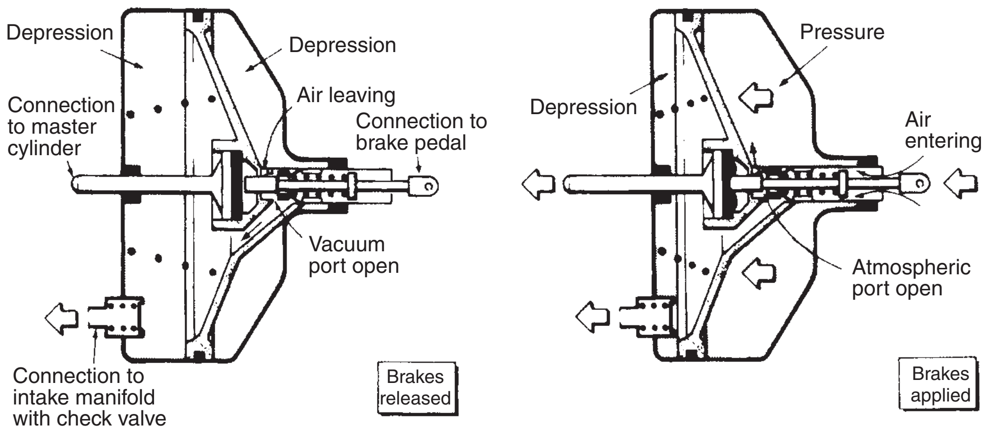


Figure 27.51 Operating principle of a direct-acting suspended vacuum brake servo

psychological barrier that may exist, when a driver is reluctant to brake hard enough in an emergency situation for fear of losing control of the vehicle: and second, to compensate for any physical limitations that a driver may have in exerting sufficient pedal effort under these circumstances. In both cases the system assists the driver to slow the vehicle in a shorter period of time, thereby exploiting the full potential of anti-lock brakes.

Brake assist systems, now sometimes abbreviated to BAS, generally fall into two categories. The simplest arrangement comprises a two-stage vacuum servo or booster unit; while a more comprehensive installation includes an independent power source of pressurized brake fluid, which can be delivered to the main hydraulic operating system when required.

A two-stage suspended vacuum servo unit is essentially a modification of a conventional unit. In one version a pair of depression chambers are mounted in tandem with the hydraulic master-cylinder, the rear chamber being larger in diameter than the front one. Under normal braking the diaphragm in the rear chamber acts on the push-rod of the master-cylinder (Figure 27.52a). This occurs when air admitted by the plunger control valve linked to the brake pedal acts behind the diaphragm and creates a pressure difference across it. During hard braking, such as in an emergency, a pre-set coil spring in the plunger control valve can then be overcome. This allows increased travel of the plunger control valve, which admits air not only behind the diaphragm in the rear chamber, but also behind the smaller diameter diaphragm in the front chamber

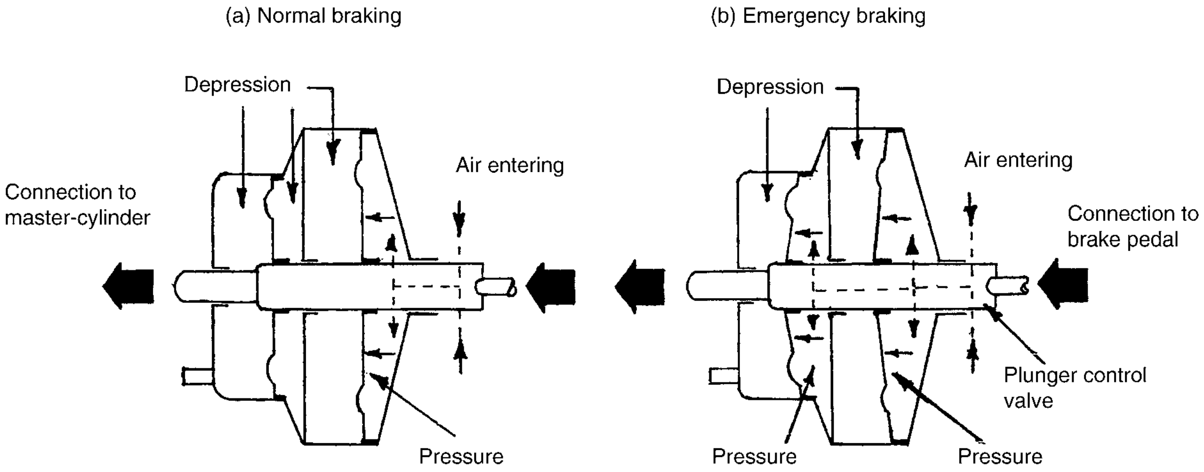


Figure 27.52 Basic principle of emergency brake assist vacuum servo unit

(Figure 27.52b). Hence, there is an additional apply force transmitted to the push-rod of the master-cylinder, which utilizes more effectively the driver effort at the brake pedal for greater stopping power.

In another version of a two-stage vacuum servo unit a single depression chamber and diaphragm is retained, but the plunger control valve incorporates a solenoid. This is actuated by an electronic control system that senses rapidity of brake pedal movement. Under emergency braking conditions, the solenoid is therefore actuated to allow full, instead of partial, opening of the plunger control valve. The immediate build-up of air pressure behind the diaphragm therefore maximizes servo output for greater stopping power, which otherwise may not have been available if related purely to driver effort at the brake pedal. This type of two-stage vacuum servo unit acts in the manner of a conventional unit for normal braking.

Where an independent power source of pressurized brake fluid is employed for brake assist, it is controlled electronically and may also be available for other vehicle functions such as traction control. The purpose of the electronic control is to monitor by means of sensors the speed and force with which the brake pedal is pressed, and compare this data to a stored reference value. It can then determine whether additional line pressure for the brakes is required to give maximum emergency braking. In this event, the brake assist system is commanded to release high-pressure fluid from an accumulator into the main braking system, thereby increasing line pressure above that otherwise related to driver pedal effort. An electrically driven pump is used to charge the hydraulic accumulator, which stores fluid at a pressure in the region of 17.5 MN/m^2 (2500 lbf/in^2).

Brake assist systems do, of course, reduce their contribution to brake application, when the driver eases pedal effort and the emergency has passed.

27.7 THE PARKING BRAKE SYSTEM

General layout

The regulations governing the construction and use of motor vehicles requires their brakes to have a second independent means of operation. This function is performed by a mechanically actuated parking brake system, which in conventional practice is hand operated. Hydraulic connections are not employed for the handbrake system owing to the possibility of the fluid leaking slowly past the seals when the brakes are left applied for any length of time.

In early practice, the control linkage of car handbrake systems generally comprised a series of rods, shafts and levers, but it has long since become established practice to use a simpler arrangement of part-enclosed cables. The following describes the layout and construction of a modern handbrake system.

Driver control

This takes the form of either a pivoted or a sliding handbrake lever incorporating a pawl and ratchet mechanism. As the lever is pulled towards its on position, the spring-loaded pawl slides over the stationary ratchet teeth. When the handbrake is fully applied and the driver releases hold on the lever, the pawl transmits a locking force to the ratchet teeth owing to their special form and thus prevents the lever from returning to the off position. To return the handbrake to its off position, pressure on a release button or catch overcomes the spring load on the pawl to disengage it from the ratchet.

Operating cables

These provide a flexible connection between the handbrake lever mechanism and the wheel brake assemblies. They comprise a cable inner core that is always loaded in tension and connects between the handbrake lever mechanism and the relay levers on the wheel brake assemblies. Partly enclosed within a spirally wound outer casing, the cable is anchored by this casing at one end to the wheel brake assembly and at its other end to the understructure of the car. The purpose of the casing is to serve as a continuous guide and thus prevent the inner cable from straightening. Since it is only necessary to provide control movement in the brake apply direction, the inner cable is not subject to compressive loads that could buckle it and is pulled to its off position by return springs in the brake assemblies.

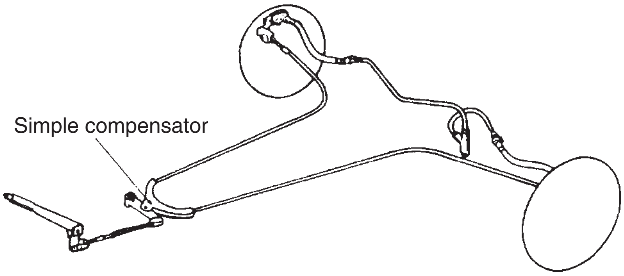


Figure 27.53 Handbrake system with simple compensator

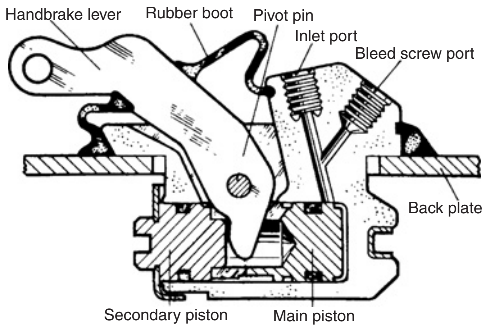


Figure 27.54 Single-piston wheel cylinder with integral handbrake mechanism

Equalizer mechanism

In order to ensure that each of the two brake assemblies operated by the handbrake is applied with the same force, the inner cables are not connected immediately to the handbrake lever mechanism, but through a simple compensating or equalizer device (Figure 27.53). This usually takes the form of a centrally pivoted balance beam with the cables connecting to its outer ends. Alternatively, a continuous inner cable may be used to connect the two brake assemblies, in which case its apex is guided around either a stirrup-mounted pulley or a saddle piece equalizer. To provide an overall adjustment for the handbrake system a screw-thread device is incorporated in series with the handbrake lever mechanism and equalizer device. A silicone grease is typically used to maintain lubrication for the enclosed part of the cable run.

Hydraulic wheel cylinder with integral handbrake mechanism

It was earlier mentioned that a single-piston hydraulic wheel cylinder can be made double acting by allowing its body freedom to slide on the back plate. In one design of leading and trailing shoe rear brake a wheel cylinder of this type is used with added provision for mechanical operation of the shoes by the handbrake (Figure 27.54). Although this makes for a more complicated design of wheel cylinder it does avoid the need for a separate lever-operated connecting linkage between the shoes, where the available space is usually fairly restricted by the bearing housings for the rear hubs. However, the latter type is the more widely used (Figure 27.15).

Mechanical operation of the single-piston hydraulic wheel cylinder is accomplished by making the piston in two halves. The sealed inner half is acted upon by the footbrake hydraulic system, while the outer half, or buffer piston, is actuated by a cranked lever pivoting in the cylinder body and connected to the handbrake system. A specially shaped slot is cut out of the two-piece piston to accept the inner end of the cranked lever. The length of the slot is such that hydraulic operation of the piston assembly does not disturb the cranked lever.

When the brake pedal is depressed, the build-up of hydraulic pressure in the wheel cylinder produces a two fold action. First, it moves the sealed inner piston which in turn carries the outer buffer piston with it and so applies the leading shoe. Second, it causes an opposite movement of the slidably mounted cylinder body, the effect of which is to apply the trailing shoe.

When the handbrake is operated, the inner end of the cranked lever forces only the buffer piston to move and apply the leading shoe, while the reaction force at the lever pivot causes the cylinder body to slide and apply the trailing shoe.

Handbrake arrangement for disc brakes

The addition of a handbrake mechanism to the disc brake poses certain problems, which partly account for the mixed system of front disc and rear drum brakes being used in conventional practice, although an alternative approach is to build a small drum handbrake mechanism into the disc hub, as mentioned earlier.

Problems arise because the disc brake is to all intents and purposes non-self-energizing in its action and does, in fact, have the lowest brake factor of all automotive brakes. Hence, it follows that the force required to operate it is necessarily the highest. As far as the handbrake system is concerned, this implies that a large leverage must be provided between the driver control and the disc pads. Therefore a small amount of wear at the pads can be magnified into a greatly increased travel at the handbrake lever. It is for this reason that an automatic adjusting device is generally specified for such systems. Also, the operating linkage must be made as rigid as possible, because any unwanted flexing would again increase the travel at the handbrake lever.

In a modern example of automatic handbrake adjuster for a floating caliper disc brake, the input lever actuates a cam and push-rod that transmits an apply force, via the adjusting mechanism (Figure 27.55), to the disc pad piston. When a predetermined amount of pad wear has occurred, and in the course of normal operation of the service brake hydraulic piston, the adjuster pawl clicks over the piston ratchet to establish a constant adjustment for the handbrake. As in other automatic adjusters for brakes, there is a small amount of in-built lost motion to prevent over-adjustment and also permit proper relaxation of the pads after each operation of the service hydraulic brakes (their mode of self-adjustment having been explained earlier).

Electric parking brake

More recently there has been a trend towards electric parking brakes for some high-grade passenger cars. The system basically comprises two electromechanical actuators that are integrated with the floating calipers for the rear disc brakes. For this type of system a clamping force on the brake pads can be derived from a jacking screw and screwed sleeve mechanism.

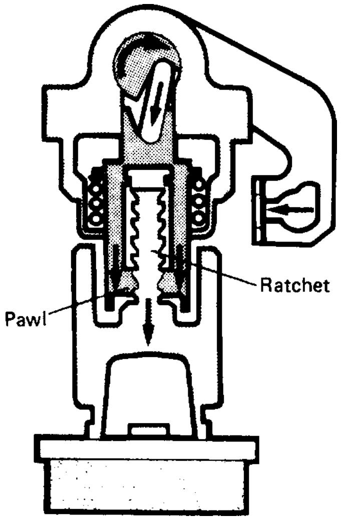


Figure 27.55 Automatic adjuster for disc handbrake (*Girling*)

The jacking screw acts directly upon the brake pad backplate and is prevented from rotating, whilst the screwed sleeve is axially constrained and receives a rotational movement from a concentrically mounted electric motor. Therefore when the handbrake is operated and the motor energized, the rotation of the screwed sleeve compels the jacking screw to transmit an endwise force on the pad backplate to apply the brake. A conventional hydraulic service brake is retained with this system. By virtue of its electrical operation the parking brake can be computer controlled, so that rolling back on hill starts can be avoided and emergency braking can be signalled via a stability control system.

27.8 HYDRAULIC POWER BRAKES

General background

So far we have considered only those hydraulic brake systems where the transmission of apply pressure to the wheel cylinders is either wholly, or partially, dependent upon the effort expended by the driver on the brake pedal. Another category of hydraulic brake system is the fully powered type, the operation of which is analogous to the driver merely depressing a pedal-controlled admission valve which either indirectly, or directly, subjects the wheel cylinders to a build-up of hydraulic pressure from a separate energy source and hence applies the brakes. In a practical system it is however arranged for the depression of the brake pedal to be resisted by a moderate force that is proportional to the forces being applied to the brakes, so that the driver can better feel the amount of braking being used.

Although the idea of hydraulic power braking had received investigation in an earlier period of motor vehicle history, it was not until the mid 1940s that such a system became available, originally on public service vehicles and then some ten years later on passenger cars, albeit on a limited scale. A notable example of hydraulic power braking for a city bus was the Lockheed system, which was later developed in conjunction with London Transport for installation on their successful Routemaster bus in the early 1960s. As compared with an alternative air brake system, the hydraulic power brakes better met their requirements for a lighter-weight installation, a more accurate and quicker responding pedal control, and also greater reliability since condensation troubles and entrainment of foreign matter in the system are avoided.

For passenger cars the introduction of hydraulic power brakes by Citroën on their revolutionary DS19 model in 1955 was a significant step forward in the concept of a central hydraulic system, which additionally provided an energy source for the automatic control of the clutch and gear change, power assistance for the steering, and automatic height control for the suspension. Since the late 1970s there has been an increasing incentive to develop hydraulic power brakes for cars. This has arisen from the legislative requirements for emission control equipment, the associated use of petrol injection systems and the wider use of diesel engines, all of which have tended to diminish the available intake manifold depression for operating a conventional vacuum servo unit. Indeed, a separately driven exhauster pump is necessary for this purpose in the case of diesel-engined cars. More recently, there has been the further incentive of combining the power source with electronically controlled solenoid valves to provide an anti-lock braking system, as described in Section 29.

Types of hydraulic power brake

In terms of being fluid power control systems, there are basically two circuit arrangements that may be used for hydraulic power brakes, which are classified as ‘open centre’ and ‘closed centre’.

Open-centre control system

This represented the earliest type of hydraulic power brake and simply comprised three units: a hydraulic pump of the positive displacement type that was permanently driven from the transmission system, a conventional master cylinder combined with a follow-up servo valve of the piston type that was linked to the brake pedal, and a fluid supply tank.

With the brake released there was no end contact between the master cylinder and servo valve pistons, so that there was a free circulation of fluid from the pump to the supply tank via the central passage of the servo valve (Figure 27.56a). When the driver applied the brakes by depressing the pedal, the forward movement of the servo valve closed its mouth against a seating formed on a rearward extension of the master cylinder piston, thereby stopping what had otherwise been the free return of fluid through its central passage to the supply tank. A build-up of pressure from the pump therefore occurred between the exposed areas of the larger-diameter master cylinder and smaller-diameter servo valve pistons (Figure 27.56b), which in the former case applied the brakes in the manner of a conventional hydraulic brake system and in the latter case imposed a moderate resisting force at the pedal to provide the driver with a sense of feel as to the amount of braking being used. The magnitude of the pressure build-up in the hydraulic power system and therefore the braking effort generated indirectly by the master cylinder system remained proportional to the effort expended by the driver on the brake pedal. In other words, whenever the separating force on the servo valve

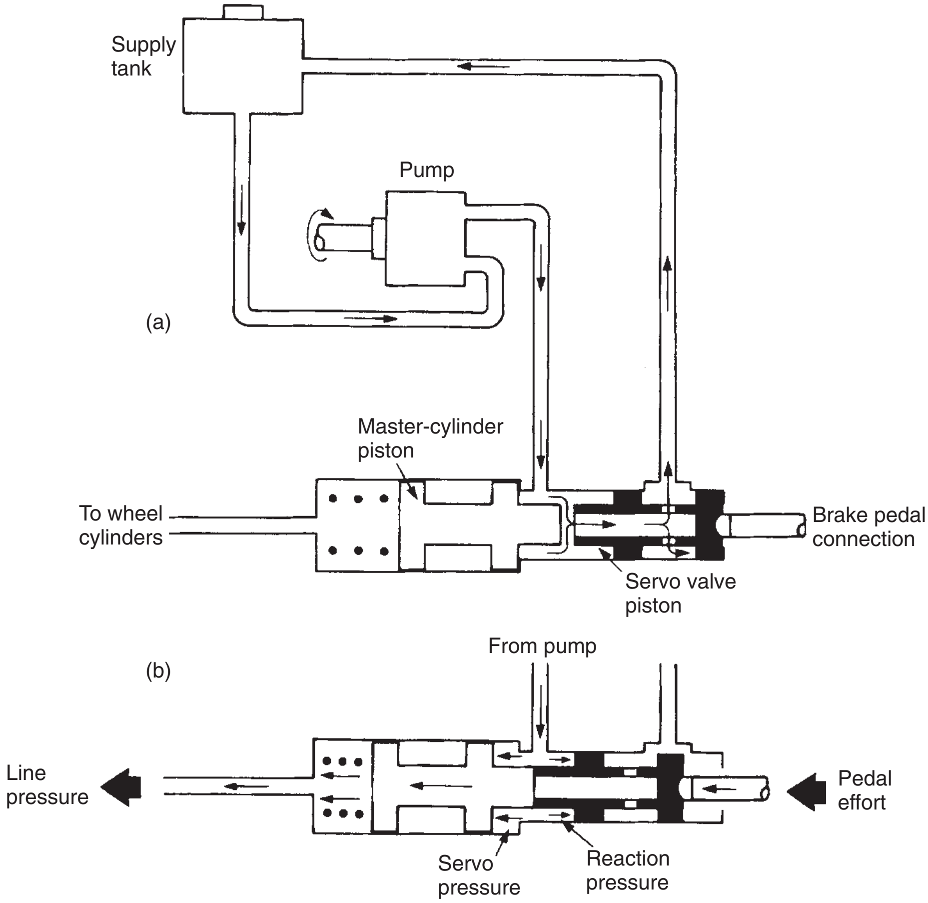


Figure 27.56 Schematic arrangement and operation of early-type open-centre control for hydraulic power brakes

exceeded that due to the opposing pedal force, the servo valve opened just sufficiently to allow a restricted amount of the fluid being pumped to return via its central passage to the supply tank, which prevented any further rise in pressure in both the hydraulic power and master cylinder circuits.

In later applications of the open-centre control system a pedal-operated and spring-returned distribution valve of the spool type is used. This still allows free circulation of fluid between the pump and supply tank when the brakes are released (Figure 27.57a), but establishes a direct connection between the pumped fluid and the wheel cylinders when line pressure is required to apply the brake (Figure 27.57b). A true hydraulic power brake system is therefore obtained, as there is now no need for a separate master cylinder circuit. The driver is again provided with a feel of the amount of braking being used, because the distribution spool valve has a restricted central passage that allows the brake line pressure to act beneath the valve and exert a moderate reaction force against brake pedal effort.

With an open-centre system it will therefore be clear that the working pressure for hydraulic power braking is created by the servo or distribution valve, which provides a modulated restriction to the pumped fluid returning to the supply tank. In this respect it may be compared with the control valve of a power-assisted steering system (Section 25.3). This type of hydraulic power braking system has the merits of simplicity and, since the system is in a depressurized state with the brake released, it assists in limiting external leaks and extending the life of the components. However, it does have the significant disadvantage that pump delivery is reduced at low vehicle speeds, which then makes it difficult to maintain powerful and responsive application of the brakes. To overcome this problem, later open-centre systems have been used in combination with a closed-centre type, the arrangement of the distribution spool valves for such a system being shown in Figure 27.59.

Closed-centre control system

This represents the modern and more comprehensive type of true hydraulic power brake system. It basically comprises a supply tank from which the fluid is fed to a high-pressure pump driven from the engine. The pump then delivers fluid at a pressure higher than the highest pressure required by the braking system to a gas-charged hydraulic accumulator, where the fluid under pressure is stored until it is released to apply the brakes. This is accomplished by depressing a pedal-operated and spring-returned distribution valve of the spool type, which releases the fluid directly to the wheel cylinders (Figure 27.58b). As in the case of the later type of open-centre control system, the distribution spool valve

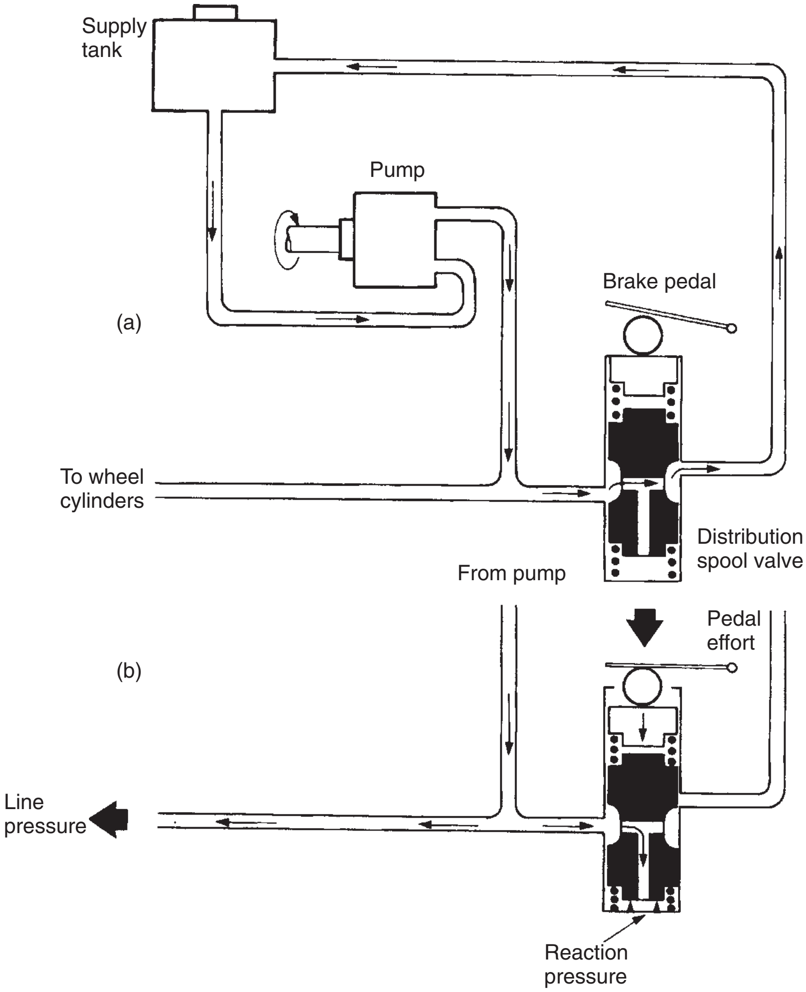


Figure 27.57 Schematic arrangement and operation of later-type open-centre control for hydraulic power brakes

has a restricted central passage that allows the brake line pressure to act beneath the valve and therefore create a reaction force to provide the driver with braking feel. When the brakes are released the fluid is returned via the distribution spool valve to the supply tank (Figure 27.58a). A pressure regulation valve is included in the system to direct the excess output from the pump back to the supply tank; in addition, the accumulator has a non-return valve so that the brakes may be applied a limited number of times after the pump has stopped. Another important advantage that is available from any true hydraulic power braking system is that there is virtually an unlimited displacement of fluid to operate the brakes, which makes the system less sensitive to fluid vaporization, air entrainment and maladjustment of the brakes.

In contrast to an open-centre control system, it will therefore be noticed that when the brakes are released there is no free circulation of fluid through the distribution spool valve. It is only when the latter is depressed by the pedal sufficiently to establish connection between the fluid stored under pressure by the accumulator and the brake lines leading to the wheel cylinders that braking occurs. That is, the braking effort depends upon removing, rather than imposing, a restriction on the fluid being supplied under pressure.

As with other types of brake installation, hydraulic power brake systems have long been duplicated from safety considerations. In commercial vehicle practice this usually takes the form of a tandem brake valve, wherein a primary distribution spool valve moves above a secondary one to operate the front and rear brakes respectively. For passenger car installations two distribution spool valves are commonly mounted side by side with a balance beam linkage connecting them to the brake pedal.

27.9 MAINTENANCE OF HYDRAULIC BRAKES

Preventive maintenance

There is an ancient saying that 'decay is inherent in all material things', and this is no less true for the components of hydraulic brake systems. Servicing for brake safety in terms of preventive and general maintenance has therefore always received careful consideration from brake equipment manufacturers.

Preventive maintenance is concerned with the scheduled replacement, or overhaul, of hydraulic components when in the judgement of the manufacturer a reasonable working life has been achieved. Typically the scheduled periods for

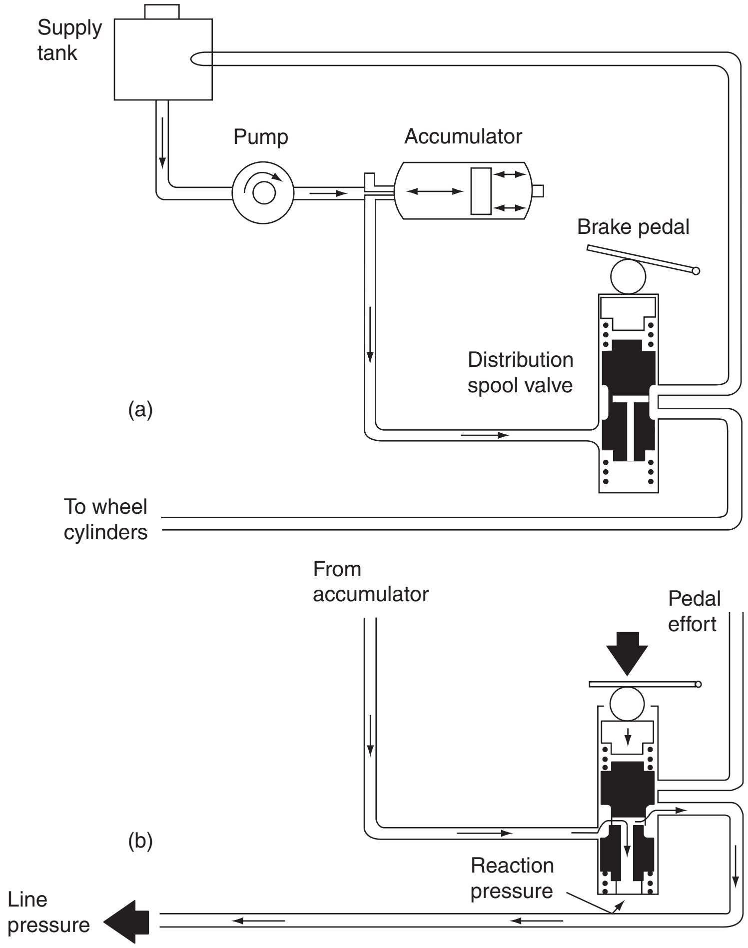


Figure 27.58 Schematic arrangement and operation of a closed-centre control for hydraulic power brakes

passenger cars and light commercial vehicles range from 40 000– 60 000 miles (64 000–96 000 km) or three years, whichever occurs first, and may depend for example on the type of brake calipers used. It is then advised that all hydraulic cylinders, hoses, apportioning valves and servo units are replaced by new, guaranteed components or, if this should not prove possible, the existing cylinders and servo unit may be overhauled using the appropriate service kit.

General maintenance

Weekly Check the level of the brake fluid in the reservoir and, if necessary, top up to the required level. A sudden fall in fluid level indicates leakage from the system, which must be investigated and rectified without delay.
Every 1000 miles (*1600 km*) Adjust drum brakes unless, of course, automatic adjustment is provided for them.
Every 5000 miles (*8000 km*) Remove the brake drums and check for wear of the linings, which must not be allowed to wear down to the rivets. In the case of bonded linings these should not be worn below 1.5 mm (1/16 in) from the metal shoe.
Every 5000 miles (*8000 km*) Check the disc pads for wear; if they have worn to 3 mm (1/8 in) thickness, replacements should be fitted. This check may not be required where on-board wear indicators are incorporated.
Every 5000 miles (*8000 km*) Examine carefully all flexible brake hoses and their unions and also the full length of each metal pipeline for fluid leakage and any signs of general deterioration, such as chafing or corrosion. Renew any unserviceable part without delay.
Every 10 000 miles (*16 000 km*) Clean down the back plate and lubricate the tips of brake shoes, the shoe platforms and the adjuster with an approved brake grease.
Every 18 months to 2 years Renew the fluid in the braking system.

The vehicle manufacturer's own recommended servicing schedule for the hydraulic brake system should always be consulted for any special requirements. For example, the air filter of a vacuum servo unit may need renewing at certain intervals.

Hydraulic brake misbehaviour in service

Assuming that any misbehaviour of the braking system is not being influenced by unequal tyre pressures, or defective operation of the steering or the suspension systems, a purely

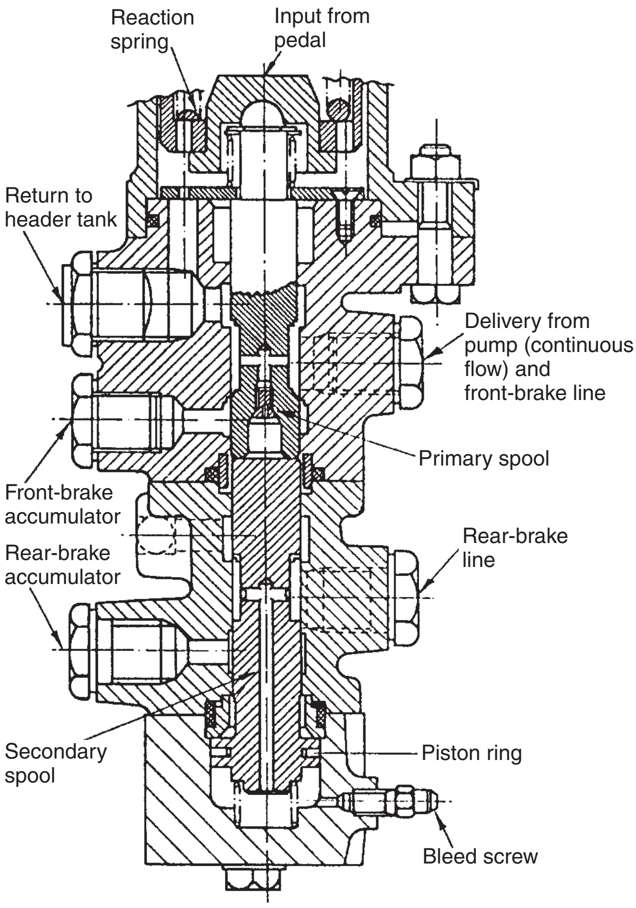


Figure 27.59 Tandem hydraulic power brake valve (*London Transport*)

general guide to the possible faults developed with hydraulic brakes is as follows:

Pedal travel excessive
Drum brakes need manual adjustment
Drum brakes' automatic adjusters seizing
Disc hubs too much end float
Discs running out-of-true
Distorted disc pad damping shims.
Pedal slowly sinks
External leakage from hydraulic system
Internal leakage past master cylinder main seal.
Pedal feels spongy
Air trapped in hydraulic system
Badly relined brake shoes.
Pedal feels springy
Shoe linings not bedded in
Master cylinder mounting defective.
Pedals feels hard
Glazed or contaminated shoe linings
Glazed or contaminated disc pads
Wheel cylinder pistons seizing
Caliper cylinder pistons seizing
Brake servo unit inoperative.
Brakes pulling to one side
Other side shoe linings contaminated
Other side disc pads contaminated
Other side wheel cylinder piston seizing.
Other side caliper cylinder piston seizing
Brakes locking prematurely
Brake system apportioning valve defective.
Brakes binding on release
Drum brakes over-adjusted
Handbrake over-adjusted
Handbrake mechanism seizing
Wheel cylinder piston seizing
Caliper cylinder piston seizing
Master cylinder reservoir vent blocked
Master cylinder compensating port blocked
No clearance at master cylinder push-rod.

Brake squeal

Complaints of brake squeal in service are as old as the motor industry itself. In the early years of the motor vehicle, the explanation and remedies for this phenomenon now seem deceptively simple. It was usually attributed either to dirt clogging the pores in the surface of the brake lining and causing it to become glazed, or to the presence of high spots between the lining and the brake drum. So in service it was the practice to remove the brake shoes and clean their linings, using a stiff wire brush and a flammable agent, which is perhaps best not recalled in the safety conscious climate of today. Alternatively, the linings were hand-bedded using a rasp to remove the high spots and especially to avoid premature heel and toe contact before readjusting the brakes. If these measures did not provide a convincing cure, then it was not unknown for owners to be assured that squealing brakes were at least efficient ones!

In referring to brake squeal in the 1930s a distinguished research engineer of that period, L.H. Dawtrey, wrote 'This is one of the most difficult problems which the designer has to contend with as evidenced by the large variety of vehicles which suffer from the disability.' By then it was realized that the explanation for squeal in drum brakes could involve a number of factors. These included insufficient rigidity of the brake drum and the backplate (or anchor plate as it was then often termed) on which the shoes were mounted, incorrect operating geometry for the shoes, too high friction coefficient for the brake lining and lack of vibration damping measures.

Later research by both vehicle manufacturers and brake equipment suppliers appeared to confirm that brake squeal is caused by a resonant vibration of the rotating and stationary parts of the brake assembly, whether it be of the drum or the later disc type. When the resonant frequency of vibration comes into the audible range, then it is heard as brake squeal. The source of the vibration energy must, of course, originate between the rubbing surfaces of the brake before it is transmitted to other parts of the assembly.

In practice the natural vibration period of the brake drums and, less commonly, the brake discs, has sometimes been suppressed by fitting a frictional damping ring around their peripheries. A classic past example of the former was the

Rolls-Royce spring-loaded damper ring, which encircled the drum near its open end and acted in the manner of a crankshaft torsional vibration damper (Section 1.9). The same principle may also be applied to modern brake discs, one method used being to spring a split steel ring into a deep groove machined in the periphery of the disc.

Again, as a matter of historical interest, Rolls-Royce also incorporated an elegant example of brake shoe damping, where a spring-loaded short auxiliary shoe with an offset pivot was mounted towards the toe of each main brake shoe, so that light application of the brakes was concentrated on the smaller auxiliary shoes. Less exotic ways of damping brake shoe vibration include attaching a lead weight to the shoe web, spring-loading the shoe web into contact with the backplate, and strategic slotting of the shoe web to alter its flexibility and hence the distribution of pressure between the lining and drum.

In the case of disc brakes, a similar redistribution of pressure between the pad and disc is required to dampen vibration. It is generally achieved by offsetting in the circumferential direction the centre of pressure on the pad lining. In effect the apply force from the caliper piston is then no longer concentrated in the middle of the pad area, as would otherwise be expected. To modify the pressure distribution a suitably positioned cutaway portion on the rim of the caliper piston may be used or, as in later practice (Section 27.4), a thin steel shim with an asymmetrical cutaway portion is inserted between the rim of the caliper piston and the pad backplate. Alternatively, the pad may be supplied with the lining already offset on the backplate, or the lining can be strategically tapered.

In all cases of brake squeal it is essential to seek advice from the vehicle manufacturer before departing from the recommended build instructions and friction material specifications.

Personal safety precautions in servicing the braking system

Before jacking up either the front or the rear wheels on to axle stands in order to work on the braking system, the other pair of wheels should be chocked securely against rotation. The inhalation of asbestos dust can be injurious to health, so when cleaning out drum brake assemblies on older vehicles the dust should be removed using a vacuum cleaner or by wiping with a damp rag. Further reference should be made to the precautionary guidance issued by the Health & Safety Executive in their publication HS(G)67 titled 'Health & safety in MOTOR VEHICLE REPAIR'. Brake fluid has already been described as a toxic substance and should be handled with care; in particular, keep it away from the eyes.

27.10 BRAKE EFFICIENCY AND TESTING

Basic considerations and requirements

The total retarding force acting on a vehicle includes braking force, tyre drag, friction losses in the wheel bearings and the transmission system, the force exerted by the vehicle to drive the engine, the effect of road gradient and wind resistance. If all these forces together so happen to equal the weight of the vehicle, then it would slow down or decelerate at the rate of approximately 9.8 metres per second per second ($9.8\,m/s^2$) or 32 feet per second per second ($32\,ft/s^2$). This is the same rate as that at which a body falling freely would accelerate, were it not for air resistance, under the force of gravity. The rate of acceleration or deceleration, due to a body being acted upon by a force equal to its own weight, is known as g. It is therefore convenient to talk in terms of $0.5\,g$, $0.85\,g$ and so on, instead of expressing the actual value in either m/s^2 or ft/s^2. Likewise it is convenient to talk of braking force as a percentage of the weight of the vehicle, so that for example a 50 per cent braking force would provide $0.5\,g$ deceleration on a level road.

In automobile practice, however, it has long been customary to refer to the braking force relative to the weight of the vehicle in terms of braking efficiency, so that a $0.5\,g$ deceleration is conveniently expressed as a 50 per cent braking efficiency. In theory, this relationship holds good only as long as the coefficient of friction or adhesion between the tyre and road surface is unity or, in other words, the horizontal braking force at the wheel divided by the downward load on the wheel is equal to one. This limitation is nevertheless accepted as being a reasonable basis on which the braking efficiency of a vehicle can be assessed. Summing up, then, if the brakes are producing a retarding force that is equal say to three-quarters of the weight of the vehicle, the braking efficiency is said to be 75 per cent.

In Britain under The Road Vehicles (Construction and Use) Regulations, the basic minimum requirements for braking efficiency that cover most categories of new vehicle are 50 per cent service brakes, 25 per cent secondary brakes and 16 per cent CV parking brakes. The EC minimum requirements for braking efficiency are higher for motor cars and lower for commercial vehicles, but are based on actual stopping distances.

Testing brake efficiency

Several methods of testing brake efficiency have been used in service, past and present, which may be classified under the following headings:

Pendulum decelerometer This type of instrument, which is usually identified as a Tapley meter, is basically a calibrated pendulum device. During braking the pendulum swings forwards because of its inertia, the angle of swing bearing a direct physical relationship to the deceleration of the vehicle (Figure 27.60a). The movement of the pendulum is amplified and held as a scale reading of percentage brake efficiency.

U-tube decelerometer With this type of instrument, deceleration of the vehicle is indicated by movement of a liquid column in a U-tube, which is mounted parallel to the longitudinal axis. Deceleration causes the transfer of liquid from one limb to the other of the U-tube (Figure 27.60b), the limb in which the liquid falls being calibrated and made of smaller bore to amplify the scale reading of percentage brake efficiency.

Platform brake-testing machine This form of a permanently installed equipment was originally introduced in America, where in some states the testing of brakes was early made compulsory. It comprises four spring-returned sliding platforms on to which the vehicle is driven at low speed, the brakes being applied when each wheel is on its appropriate platform (Figure 27.60c). The sliding reaction of the platforms is transmitted as

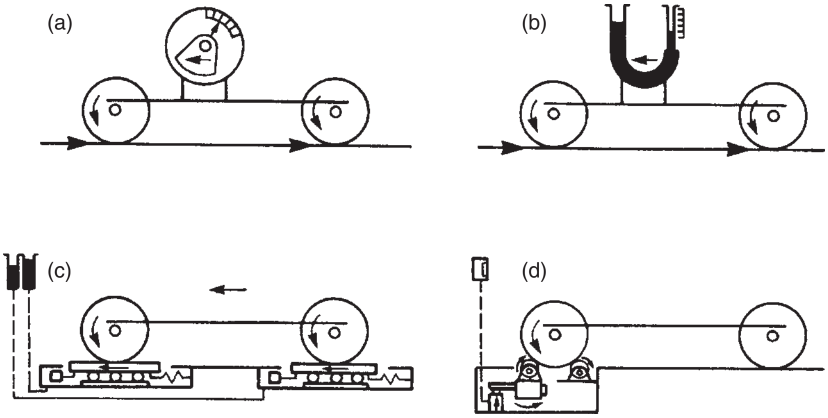


Figure 27.60 Schematic arrangements of brake-testing methods: (a) pendulum decelerometer (b) U-tube decelerometer (c) platform brake-testing machine (d) roller brake-testing machine

movement in liquid columns, which are calibrated to indicate braking force.

Roller brake-testing machine This is another form of permanently installed equipment, modern versions of which typically comprise a flush-fitting roller bed with two pairs of electrically driven rollers, these being so arranged that both wheels at either end of the vehicle can rest upon them and be tested simultaneously. During testing the braked wheels are forced to rotate by the rollers (Figure 27.60d), the torque reaction developed at the electric motors being sensed to provide an indication of braking force at each wheel, this being displayed on console gauges.

Since the testing of brake efficiency in service is usually now concerned with the DTp (referred to as MoT) brake performance test on a vehicle, it should be mentioned that this test has to be carried out using an approved roller brake-testing machine, except where this may not be appropriate as for example with vehicles having permanently engaged four-wheel drive. In this and other cases where approval has been granted, a decelerometer instrument is used. The former method of brake testing is regarded as being safer and more efficient for the DTp requirement. The DTp (MoT) *Tester's Manual* should be consulted for details of the brake performance test and the equipment used in accordance with the manufacturer's instructions. Since the operation of roller brake-testing machines or 'rolling roads' as they are now known and other brake testing equipment can pose potential safety hazards, unless suitable precautions are observed by personnel, it is advisable to consult the Health & Safety Executive publication HS(G)67 titled 'Health & Safety in MOTOR VEHICLE REPAIR'.

28 Air and endurance brake systems

28.1 PRINCIPLES OF AIR BRAKES

General background

It was as long ago as 1868 that the American engineer George Westinghouse first patented his invention for an automatic compressed air brake for railway trains, which soon proved superior to other types of braking system including his own earlier development of a vacuum-operated brake. Although compressed air brakes for railway trains were soon adopted in America and many other countries, another one hundred years passed before they appeared to any extent on railway trains in Britain. The change from vacuum to compressed air braking in fact coincided with the demise of the steam engine and the introduction of the diesel. With a steam engine the vacuum was created by an ejector, wherein the action of a jet of steam was made to exhaust air from the pipes, cylinders and reservoirs of the system. Therefore with a diesel engine either a vacuum pump was necessary, or it could be just as economical to fit a pressure pump for compressed air braking, with its greater power.

Rather interestingly a somewhat similar pattern of progress occurred with heavy vehicles on the road (although much more rapidly, of course) when the petrol engine was superseded by the diesel. With the petrol-engined vehicle the once popular vacuum-operated system utilized the depression existing in the intake manifold, but since the diesel engine could not offer this facility it similarly became advantageous to mount an engine-driven compressor, rather than an exhauster, and install compressed air brakes, again with their greater power.

Both railway and heavy-vehicle engineers have therefore considered it preferable to utilize a source of compressed air, generally at 700 kN/m^2 (100 lbf/in^2) or more, rather than a source of vacuum that must always be less than atmospheric pressure, or to put it another way a negative pressure no greater than 105 kN/m^2 (15 lbf/in^2). More specifically, the higher operating pressures that are made possible with compressed air brakes allow a reduction in size of the system components, accompanied by quicker application and release characteristics.

More recent years have seen the operation of heavy vehicles where both gross weight and speed capability are much greater than ever before. This in turn has led to an increasing degree of sophisticated engineering in air brake systems, not least to meet the safety directives of the EC (European Community). Indeed, it may be said that air brake technology has virtually become a separate branch of motor vehicle engineering.

Advantages of air brakes

For medium to heavy commercial vehicles, compressed air braking systems, or air brakes as they are generally known, offer the following advantages:

1 As an operating medium, air costs nothing and is always available.
2 The system will tolerate a certain amount of air leakage without failing completely.
3 Large operating forces to expand the brake shoes can readily be generated.
4 A supply of compressed air is a convenient source of energy to operate various ancillary equipment on the vehicle.

Basic installation of air brakes

Commercial vehicle air brake installations constitute a true power as distinct from power-assisted braking systems. This is because there is no direct connection in any mechanical or hydraulic sense between the brake treadle and the wheel brake chambers, although the driver is provided with a certain degree of feel related to system air pressure during braking. The functions of an air brake system are to compress, store, meter and deliver a volume of air under pressure to the wheel brake actuating chambers.

In simple terms, the operation of a basic single-circuit air brake system (Figure 28.1) is such that when the brake treadle is depressed one of two related control valves is opened, so that air under pressure from the reservoir can pass through the control valve and into each wheel brake actuating chamber. Here the compressed air acts against a diaphragm, its resulting movement being transmitted via a push-rod to either the operating lever of the brake camshaft, or the wedge of a brake expander unit, which forces the shoes against the brake drum. As the brake treadle is released, the previously mentioned control valve closes and the other one is opened, thereby allowing the air under pressure in the brake actuating chambers to be exhausted to the atmosphere and the shoe return springs to release the brakes. In the event of the system air pressure falling below a safe working minimum, either a warning light or a buzzer is automatically activated in the driving compartment.

The components of an air brake system are most conveniently considered under the following headings:

Compression and storage
System control
System actuation.

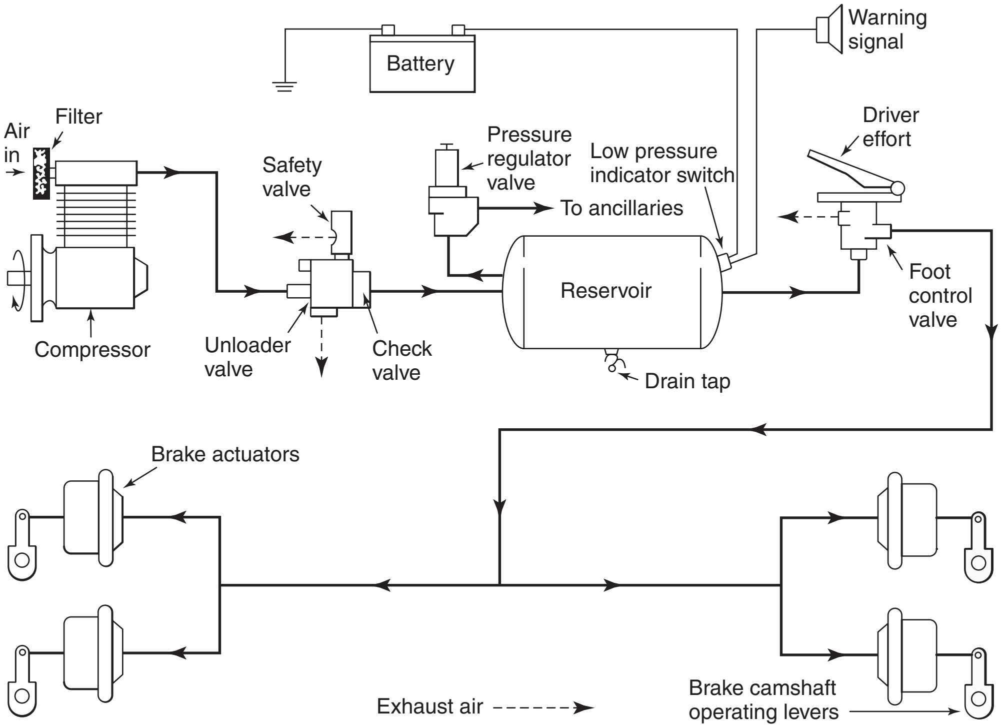


Figure 28.1 Layout of a basic single-circuit air brake system

28.2 COMPRESSION AND STORAGE

Air compressor

This generates a supply of compressed air to operate the braking system and also the vehicle ancillary services. In modern installations it is usually mounted on, and either belt or gear driven from the engine. It takes the form of a reciprocating piston pump that may have a single or twin in-line cylinders. The cylinder head is provided with inlet and delivery valves of the spring-loaded disc type, which are arranged to open in opposite directions. Of rugged construction, the main body of the air compressor is typically cast from high-grade iron. An air-cooled compressor can be recognized by cooling fins integrally cast into the cylinder block and head, while a water-cooled compressor is jacketed and has a smooth exterior with connections to the engine cooling system. The main and big-end bearings of the compressor receive oil under pressure from the engine lubrication system, the small-end bearing and cylinder wall being splash lubricated. Oil return and breathing is into the engine sump.

In operation, the compressor receives a clean supply of air either via the engine intake manifold or from a separate filter mounted on the engine cylinder block. During the down-stroke of the piston a partial vacuum or depression is created in the cylinder space above the piston, so that atmospheric pressure acting above the inlet valve forces it to open against its spring loading and allows air to enter the cylinder (Figure 28.2a). As the piston commences its up-stroke, the increasing pressure of air in the cylinder allows the inlet valve to close under the influence of its spring loading. The continued movement of the piston on its up-stroke then further increases the air pressure in the cylinder, until it becomes sufficient to overcome the spring loading acting on the delivery valve (Figure 28.2b). Air is then discharged under pressure through this valve for delivery to a storage reservoir. As the piston commences its next down-stroke, and simultaneous with the reopening of the inlet valve, the increasing depression in the cylinder allows the delivery valve to close under the influence of its spring loading, thus preventing the compressed air from returning to the cylinder.

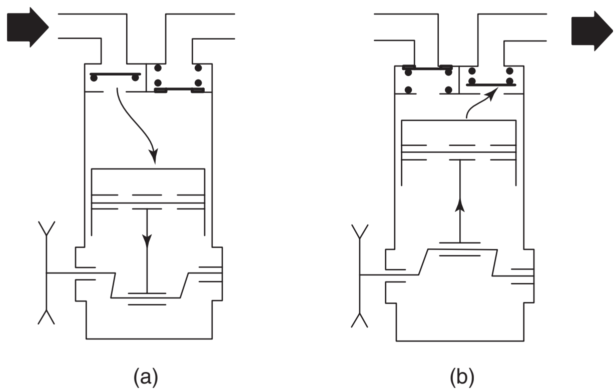


Figure 28.2 Operating principle of air compressor: (a) intake stroke (b) delivery stroke

Air reservoirs

Several reservoirs are necessary to serve the separate main service, emergency secondary and, where applicable, trailer brake circuits, which comprise the modern comprehensive air brake system. Their purpose is to provide storage capacity of compressed air at the required pressure for immediate effective braking and other ancillary services, and also to maintain the pressure supply after the engine has stopped. The total air storage capacity of the reservoirs is subject to legislative requirements. It is decided on such factors as the air delivery rate of the compressor, the size and number of the wheel brake actuating chambers, the frequency of brake application as related to vehicle type and operation, and the further demands of the ancillary services.

Since they act as pressure vessels, air reservoir tanks are designed with a large factor of safety and constructed from sheet steel with welded seams. Each reservoir is provided with either a plug or a drain valve at its lowest point, so that any condensation or sediment that accumulates may be regularly drained off. This is necessary because, depending on humidity, the air entering the compressor contains a certain amount of water vapour, and as it passes through the compressor it can also acquire oil mist from cylinder wall lubrication. The emulsion of water and oil can have adverse effects on air brake components, while any freezing of the water content in could weather could seriously affect the operation of the braking system.

Two related items of protective equipment that may be included in the air brake compression and storage system are the air dryer and alcohol injector. An air dryer is fitted in the compressor air discharge line to the reservoirs and is mounted external to the engine compartment, such that the motion of the vehicle induces a cooling air flow over the dryer body. Although the main purposes of an air dryer will be self-evident, it is also intended to exclude oil droplets, carbon particles and any other contaminants from the compressed air before it enters the system reservoirs, so that the provision of cool, clean and dry air for the control and actuation systems will increase both service life and operating efficiency of their components. An air dryer is a basically simple unit and operates in three stages that involve cooling, filtering and drying. The heated air entering from the compressor is first cooled by contact with the dryer body interior and deposits water and oil condensate into a lower sump. The cooled air then passes through an oil filter to remove any remaining liquid droplets or solid contaminants, before entering a desiccant chamber. This contains a cartridge of microcrystalline pellets that possess a strong affinity for water and present an exceedingly large surface area to the air flowing over them. By this means the water vapour still in the air is adsorbed by the desiccator, the air being progressively dried before finally entering an upper purge chamber from whence it flows, via an outlet check valve, to the reservoirs. Dried air remaining in the purge chamber is utilized to expel sediment from the sump of the dryer and to remove adsorbed moisture from the desiccant pellets. This process occurs whenever the reservoir air pressure signals the governor to unload the compressor, because air from the unloader line is then allowed to open a purge valve in the base of the dryer, which causes a sudden decompression and reverse flow of air through the unit.

The function of an alcohol injector is to deliver a positive and metered quantity of alcohol into the air being discharged under pressure from the compressor, so that it is carried by the air stream to the rest of the system. Its purpose is to lower the freezing point of any water vapour entrained in the compressed air supply, which reduces the risk of brake misbehaviour through moisture freezing in the system during low-temperature operation of the vehicle. An alcohol injector essentially comprises a simple spring-returned plunger pump with an alcohol reservoir. The injection process is accomplished by depression of the plunger over its working stroke, which occurs whenever the reservoir air pressure signals the governor to unload the compressor, since air from the unloader line is also directed above the enlarged head of the plunger. A non-return valve at the outlet of the injector isolates the entire unit from system air pressure when the pump is not being activated. Provision is made to lock the plunger out of action when the injector is no longer required after winter use, and, of course, the reservoir is then allowed to remain empty. Mineralized methylated spirit or ethyl alcohol are used in the injector unit, the latter generally being preferred. In either case it should be recognized that these substances are highly flammable and give off toxic fumes.

Governor valve

Also known as an air limiter valve, this is mounted in a return line between the compressor and reservoirs to maintain air storage pressure between normal operating limits; typical cut-out and cut-in pressures for a governor valve are 735 kN/m^2 (105 lbf/in^2) and 630 kN/m^2 (90 lbf/in^2) respectively. The governor valve exerts this control by directing air at cut-out pressure to an unloader plunger in the compressor head, which holds the inlet valve off its seat so that no further air compression is possible in the cylinder. Otherwise, the compressor would pump air continuously to the reservoirs regardless of operating pressure requirements. When the compressor is rendered inactive in this way it is said to run light.

The governor valve is intended to be progressive in its operation, which depends upon the opposing forces created by a control or pressure setting spring and air storage pressure acting against either a sliding piston or a flexible diaphragm. We take the latter construction as an example.

The diaphragm is clamped at its outer edges by the valve body and at its centre by the head of a hollow valve plunger, above which acts the control spring. A combination inlet and exhaust disc valve is initially spring loaded into contact with the lower tip of the valve plunger.

In operation, a supply of air at storage pressure is returned to the governor valve and enters a chamber beneath the diaphragm. As the pressure increases against the diaphragm it flexes to lift the valve plunger and compress the control spring. Continued lifting of the valve plunger will first allow the inlet and exhaust valve to cover the exhaust passage seat, following which the tip of the valve plunger will lose contact with the valve altogether. When this happens the compressed air acting beneath the diaphragm passes through the hollow valve plunger and is directed to the unloader plunger in the compressor (Figure 28.3). Simultaneous with this action, the air pressure exerts an additional upward force on the control spring by virtue of acting on the area of the plunger itself. It is the presence of this additional force that allows a substantial

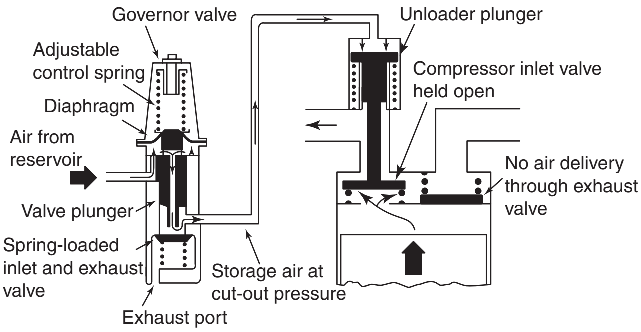


Figure 28.3 Schematic arrangement and operation of governor valve

drop in pressure of the storage air before charging recommences. In other words, it determines the required cut-out and cut-in pressures. When the storage air pressure drops below the cut-in point of the governor valve, the control spring first restores contact between the plunger tip and the inlet and exhaust valve to isolate the air supply to the unloader plunger. With reducing air pressure it then causes the plunger to unseat the inlet and exhaust valve and allow the air supply to the unloader plunger to be vented to atmosphere. The air compressor now resumes its normal operation of charging the reservoirs.

Unloader valve

An unloader valve is installed in the supply line between the compressor and reservoir where it performs a similar function to that of a governor valve. The essential difference between these alternative types of pressure control valve is that, with an unloader valve, the governor valve and unloader plunger are combined into a single unit. Its operation is such that instead of rendering the compressor inactive when cut-out pressure is reached, the compressor continues pumping air that is simply vented to the atmosphere via the unloader valve.

Safety valve

This provides a safeguard against any excessive build-up of air pressure in the storage system in the event of either a governor, or an unloader valve, failing to operate. It may be located at either the compressor, the unloader valve or the reservoir, and simply comprises a ball check valve that is spring loaded on to an orifice seating formed in a brass housing. If the storage air pressure rises above a preset limit, which is typically in the range of 896 kN/m^2 (130 lbf/in^2) to 1103 kN/m^2 (160 lbf/in^2), the ball check valve is lifted against its spring loading and allows excess air to be vented to the atmosphere through the exhaust port. The safety valve remains closed during all normal operation of the air brake system.

Safety tests

A periodic check for any abnormal leakage from the air brake system typically involves the following procedure. First, run the engine until the driver air gauges for the front and rear brake circuits show a pressure of 735 kN/m^2 (105 lbf/in^2) or 7 bar and stop the engine, then observe that the pressure does not drop more than 52 kN/m^2 (7.5 lbf/in^2) or 0.5 bar over a period of 4 minutes. Second, run the engine until the air gauges again show a pressure of 735 kN/m^2 (105 lbf/in^2) or 7 bar, next fully depress the brake pedal and stop the engine, then keeping the pedal depressed for 2 minutes observe that the pressure does not drop more than 41 kN/m^2 (6.0 lbf/in^2) or 0.4 bar. If these drops in pressure are exceeded then the system should receive urgent investigation.

The fitting of anti-lock brake systems to heavy vehicles (Section 29.5) is now a requirement of European Union legislation. This has made it necessary for the driver to check the functioning of the system before commencing a journey. For this purpose the satisfactory operation of the system is indicated by a warning lamp on the dashboard, a second warning lamp being provided for a trailer combination. The signal that appears after switching on should extinguish once the vehicle reaches a speed of about 6 mile/h (10 km/h), when the anti-lock facility normally becomes operative.

28.3 SYSTEM CONTROL

Foot-operated brake valves

These are treadle operated. Their function is to provide control of system air pressure, during application and release of the brakes which is precise and proportional to driver effort. Consider first a single foot brake valve. This essentially comprises a treadle-operated telescopic plunger and piston assembly, which acts upon a combination inlet and exhaust valve at its foot. The plunger acts upon the piston through the medium of what is called a graduating spring, this being provided with a retaining collar through which can slide the stem of the piston. Below the piston is a lighter return spring. The lower body of the piston is made both hollow and ported, so that not only does it serve as an exhaust valve seating on brake application, but also it can allow the passage of exhaust air from the actuating chambers on brake release. In the normal position, the combination inlet and exhaust valve is spring loaded against the inlet valve seat, the foot of the piston remaining clear of the upper exhaust valve.

When the driver depresses the brake treadle, a force is transmitted through an interposed roller to the spring-loaded telescopic plunger and piston assembly, the downward motion of which causes the exhaust seat of the piston to close on to the exhaust side of the combination valve. Further depression of the treadle causes the piston to unseat the inlet side of the combination valve, which then allows compressed air from the reservoir to pass through the valve and on to the wheel brake actuators (Figure 28.4a). This compressed air is also allowed access to the underside of the piston, via a bleed hole in the valve body. Therefore when this upward pressure just exceeds the downward one being exerted on the piston through the medium of the plunger and graduating spring, the piston lifts just sufficiently to allow the inlet side of the combination valve to close into its seating. At the same time the piston still maintains a closed exhaust passage against the exhaust side of the combination valve (Figure 28.4b). In this

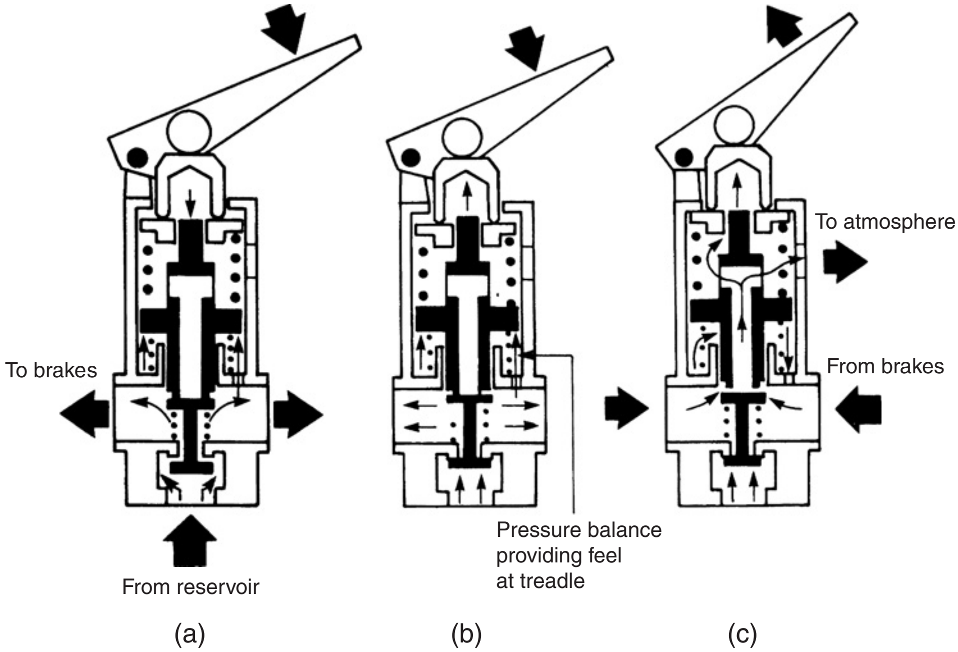


Figure 28.4 Schematic arrangement and operation of foot-operated brake valve: (a) brakes applied (b) brakes held (c) brakes released

balance or lapped condition the brakes are held on with a force that is proportional to the effort being applied by the driver, who is therefore provided with a sense of feel as to the amount of braking being used.

Any change in the braking effort exerted by the driver on the treadle will alter the balance point. If the treadle is depressed further, transmitting additional force through the graduating spring, the downward movement of the piston will again unseat the inlet side of the combination valve, thereby increasing the air pressure delivered to the brake actuators. Similarly, this additional pressure will act against the underside of the piston until it exceeds that imposed by the graduating spring force. The piston will then rise to allow the inlet side of the combination valve to close on to its seating with the exhaust side remaining closed. Another balance point is thus established with the brakes being more firmly held on, in accordance with the increased braking effort exerted by the driver.

Conversely, if the driver partially releases the treadle then the reduced force transmitted by the graduating spring will allow the piston to rise, owing to the air pressure beneath it and its return spring. When this occurs the inlet side of the combination valve remains closed and the piston retracts from the exhaust side, which allows some air to be exhausted from the brake actuators until the drop in pressure allows the piston to sink just sufficiently to close off the exhaust side of the combination valve. Once again a new point of balance is reached, but this time at a lower pressure as reflected by the reduced braking effort exerted by the driver. Of course, when the driver fully releases the treadle the piston returns to its highest position, so that the inlet side of the combination valve remains seated to cut off compressed air supply from the reservoirs, and the exhaust side is unseated to exhaust all

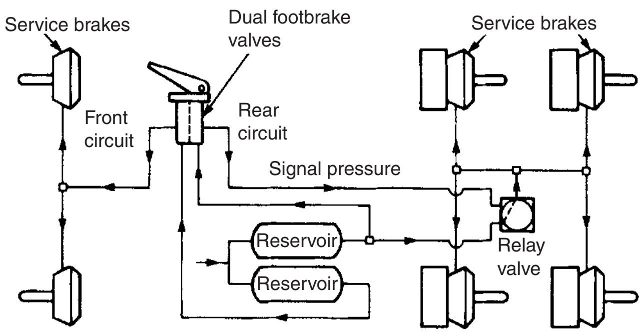


Figure 28.5 Layout of a basic dual-circuit air brake system for a six-wheel rigid vehicle

air pressure from the brake actuators to release the brakes (Figure 28.4c).

The air braking systems installed in modern heavy vehicles in Britain must be designed to meet The Road Vehicles (Construction and Use) Regulations and also EC Directives. One result of this has been the fitting of dual-circuit braking systems for rigid heavy vehicles, which was first encouraged during the mid 1960s and then some ten years later became mandatory. For example, the foot-operated service brake of a six-wheeled vehicle is split into a front service brake which is operative on the front axle brake assemblies, and a rear service brake which is operative on the foremost and rearmost rear axle brake assemblies (Figure 28.5). If a fault occurs in either of the two systems the other one must operate independently, so it becomes necessary to provide dual foot brake valves with each receiving air from its own reservoir.

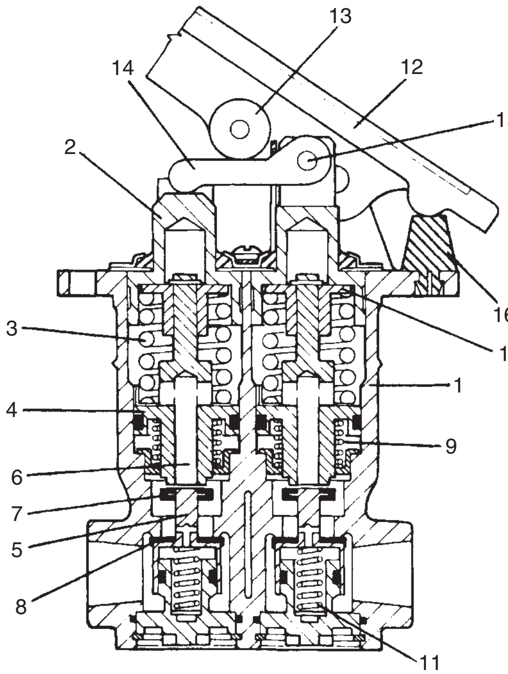


Figure 28.6 Section of Bendix Westinghouse dual foot-operated brake valves (*Seddon Atkinson*)

1 body	9 return spring
2 plunger	10 spring retainer
3 graduating spring	11 valve spring
4 piston	12 treadle
5 inlet/exhaust valve	13 roller
6 exhaust port	14 link
7 supply port	15 pivot pin
8 inlet valve	16 treadle stop

The valves may be arranged either in line with each other and known as a dual-concentric foot valve, or side by side and straddled by a balance beam. The latter is acted upon at its centre by the treadle roller to provide matched air pressures for the two circuits (Figure 28.6).

Pressure regulating valve

This type of valve is used to ensure that adequate air pressure exists in that portion of the system preceding the valve, before any pressure is allowed to build up in the system beyond it. For example, it can ensure that a brake reservoir receives charging priority over a reservoir that supplies the ancillary services. When this type of valve is installed between reservoirs it is also known as a pressure protection valve.

A pressure regulating valve is of simple construction and comprises a valve body, a regulating spring and either a diaphragm- or a piston-controlled delivery valve. In operation, the valve remains closed until reservoir air pressure acting on the underside of either its diaphragm or piston builds up sufficiently to overcome the regulating spring load. Air under pressure can then flow to the ancillary services (Figure 28.7). In the event of a system failure beyond the valve, the air pressure will drop until it reaches the cut-out pressure at which the valve

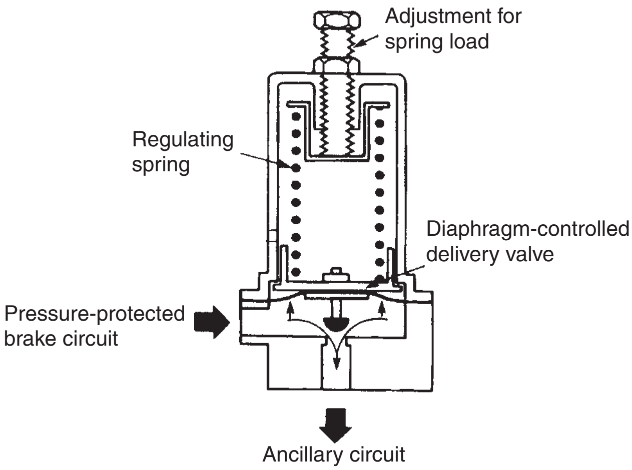


Figure 28.7 Schematic arrangement and operation of pressure regulating valve

closes. This prevents any further loss of air under pressure from the circuit being protected.

Single and double check valves

A single check valve is also known as a non-return valve, and its purpose is to allow the flow of air in one direction only. This type of valve is fitted at the entry to the air reservoirs according to system requirements, and prevents loss of pressure in the event of an air pipe or coupling suffering a fracture. The body of the check valve is screwed directly into the entry port of a reservoir, and its internal components comprise no more than a spring-loaded rubber valve with suitable guidance for the stem and a seating for the valve.

The valve is intended to operate when the pressure of air at the outlet side is equal to, or less than, the pressure at the inlet side. Air is therefore allowed to flow through the unit once the pressure at the inlet port forces the valve off its seat, but it cannot flow in the reverse direction because the combination of pressure and spring load will return the valve to its seating. To meet EEC requirements a more sophisticated version of check valve is now used, this being known as circuit protection valve. It has the characteristic of automatically isolating a defective circuit from the air supply line, so that air pressure in the remaining circuits is kept at a level high enough to maintain acceptable braking efficiency.

A double check valve performs a change-over function in allowing air under pressure from either one of two control valve systems to supply a brake actuator, while isolating the other in the event of failure. This type of valve simply comprises a body with two inlets and one outlet, the inlet passage being sleeved to guide a rubber shuttle valve, which is provided with a seating at each end of the passage.

In operation, the shuttle action of the double check valve ensures that only air at the higher of the two pressures delivered to its inlet connections will pass to the outlet and on to the brake actuator. This is because the air delivered at higher pressure will compel the valve to move over and seal off the opposite inlet connection where the air pressure is lower (Figure 28.8).

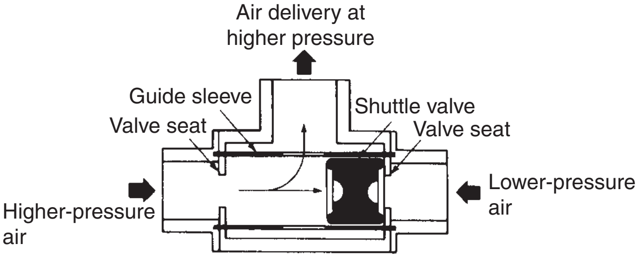


Figure 28.8 Schematic arrangement and operation of double check valve

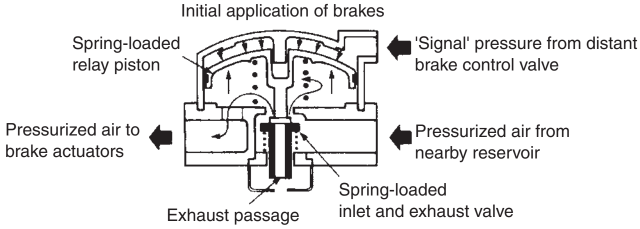


Figure 28.9 Schematic arrangement and operation of relay valve

Relay valves

For the effective operation of an air brake system it is important to obtain a rapid pneumatic balance or equalization of air pressure at all brake actuators. Otherwise, there would be an unacceptable delay in the application of the rear brakes of a rigid long-wheelbase vehicle. To overcome this difficulty it can be arranged for the rear brakes to receive their pressurized air supply more directly from a separate nearby reservoir via a relay valve. This valve can then be rapidly triggered by a signal pressure from a brake control valve, since only a small volume of air need be passed through a narrow-bore pipeline to provide the signal, and the apply pressure of the brakes remains proportional to the signal pressure.

The relay valve comprises an upper air chamber with a signal pressure port and delivery ports in the chamber base. Sliding within the chamber is a spring-loaded relay piston with a valve seat carried at the lower end of its stem. The relay valve lower body contains the by now familiar combination inlet and exhaust valve, which is made hollow and is normally spring loaded against the valve inlet port at the base of the air chamber. Valve inlet or reservoir ports are also provided in the lower body.

In operation, the large area of the relay piston ensures that there is a quick response to signal pressure from a brake control valve. The resulting movement of the piston against its spring first closes the exhaust passage through the combination inlet and exhaust valve, and then depresses this valve so that the inlet seat is uncovered. Storage air from the reservoir can now pass through the inlet port of the valve body into the air chamber beneath the piston, and leave via the delivery ports to supply the brake actuators (Figure 28.9). When the combined force of the piston and valve return springs, together with the air pressure acting beneath the piston, just exceeds the signal pressure acting above the piston it will lift sufficiently to allow reseating of the combination valve. With the inlet seat now covered by the combination valve and the exhaust passage through this valve sealed by contact with the piston stem, the relay valve is in a state of balance. That is, those brakes operated by air supplied via the relay valve are applied just as quickly and held on with the same force as the other brakes, in all cases this force being proportional to the effort exerted by the driver on the control valve.

Other more elaborate versions of relay valve may be used for certain applications. One example is the inverse relay valve that was once used with tractive unit and trailer combinations with a single-line connection. It was so arranged that air passed through it to the trailer reservoir and when storage pressure fell with brake application it triggered the relay valve, thereby admitting air to the trailer brakes from their nearby reservoir. This system had the disadvantage that no air could be supplied to the trailer reservoir during prolonged periods of downhill braking. Another example is the relay emergency valve, which is associated with modern tractive unit and trailer combinations. Apart from speeding up the application and release of the trailer brakes, it also has an emergency valve that allows automatic application of the trailer brakes should either the air pressure in the feed or emergency line to the trailer reservoir fall too low, or in the event of the trailer breaking loose from the tractive unit. For this reason it is known to American engineers as a breakaway valve.

Differential protection valve

Also known as an anti-compounding valve, its purpose is to prevent simultaneous application of both service air and secondary spring brakes, which otherwise could result in mechanical overstressing and damage to the actuating mechanism between the brake chambers and the brake shoes. This type of valve is installed after the relay valve associated with the hand control valve for the spring brake system, which will be described later.

The valve body is provided with inlet connections from the hand control and footbrake valves, and outlet connections to the ports of the spring brake chambers. In a typical construction the valve assembly comprises two concentric pistons, the outer one being spring loaded against a seating that controls the inlet port for service line air, and the inner one being spring loaded away from a seating that controls the inlet port for secondary line air. The latter arrangement is necessary to ensure that during normal driving a pressurized supply of air is made available to hold off the spring brakes (Figure 28.10). This secondary pressurized air also acts in conjunction with the spring loading on both pistons, so that the outer piston covers the service air inlet port.

The differential protection valve will be activated should the driver not only apply the service air brakes from the foot treadle, but also operate the hand control valve to release air from the secondary spring brakes and apply them as well. This unwanted application of the spring brakes is prevented because, as the pressure in the secondary line falls, the pressure in the service line overcomes the spring load on the inner piston so that it moves to cover its seating and prevents

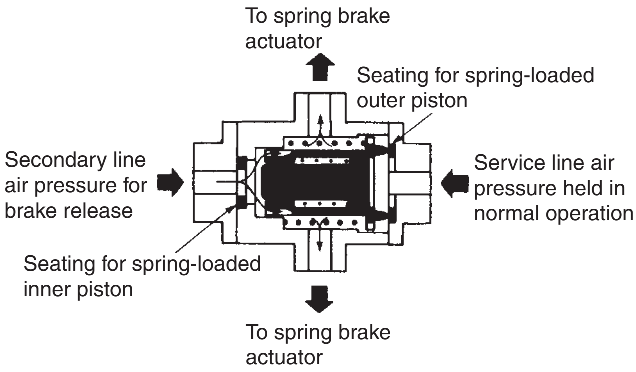


Figure 28.10 Schematic arrangement and operation of differential protection valve

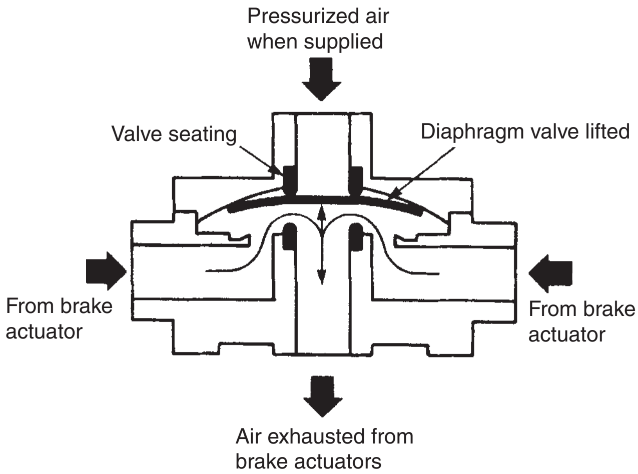


Figure 28.11 Schematic arrangement and operation of quick release valve

further loss of air through the secondary inlet port. Further increase in service line pressure then overcomes the spring load on the outer piston, which moves to uncover its seating so that service line air is admitted to the spring brake chambers. This supply of pressurized air from the service line thus replaces that exhausted from the secondary line, thereby ensuring that the spring brakes remain released during application of the service brakes. When the service brakes are released, the reduction in service line pressure allows first the outer piston to reseat and cover the service inlet port and next the inner piston to return and uncover the secondary inlet port. The secondary spring brakes will then be either applied or released according to driver operation of the hand control valve.

Quick release valves

These valves are installed adjacent to the wheel brake actuators, so that air can be rapidly evacuated from the actuators when the line pressure is released. This avoids the delay in response associated with air from all the actuators being evacuated simultaneously through long runs of air line, before reaching the atmosphere via the exhaust port of the control valve. Quick release valves can therefore perform two functions in an air brake system, since they can allow quicker release of the service air brakes and quicker application of the secondary spring brakes.

This type of valve is of simple construction, and typically comprises a valve body with two ports connecting to a pair of brake actuators and a third port that exhausts to the atmosphere. Another port is included in the cover of the valve body and connects to the brake control valve. Sandwiched between the valve body and cover is a flat rubber diaphragm that can seal against either an upper or a lower seating. When the service air brakes are applied, pressurized air is supplied to the port in the valve cover, which presses the diaphragm on to its lower seating to seal off the exhaust port. The incoming air now flexes the outer edge of the diaphragm and passes out of the ports connecting to the brake actuators. During release of the brakes the air pressure in the actuators will be greater than the reducing line pressure at the valve cover port, so that the diaphragm is lifted away from its lower seating and allows the air from the actuators to escape rapidly to the atmosphere through the valve body exhaust port (Figure 28.11).

Load sensing valve

The purpose of fitting a load sensing valve is to ensure that the braking force generated at a particular axle is proportional to the load borne by that axle. With an air brake system the valve will therefore control the air pressure in the brake actuators for the regulated axle, according to the load on that axle and brake line pressure. The valve senses the extent of rear axle loading by being made responsive to deflection of the road springs. It is therefore mounted on the chassis frame and connects to the axle through a shock absorbing linkage, so that it can ignore the effects of spring deflection caused merely by road surface irregularities.

In one type of load sensing valve a balance beam is connected at one end to a control piston and at the other end to a balance piston, the fulcrum of the beam being afforded by a pair of slidably mounted rollers. These rollers are automatically repositioned along the beam by a control rod, which is responsive to variations in axle load. The control piston is spring loaded towards the balance beam and incorporates a combination inlet and exhaust valve, while the balance piston is also spring loaded towards the balance beam and engages its end through a lost-motion fork and pin arrangement.

When the driver applies the brakes, the line pressure received by the load sensing valve acts against the control piston to force it downwards. This movement of the control piston allows the combination inlet and exhaust valve to cover the exhaust port in the valve body and uncover the inlet passage in the piston. Pressurized air can then flow through the control piston to the brake actuators and also act against the top of the balance piston. The latter is likewise forced downwards and tends to raise the control piston owing

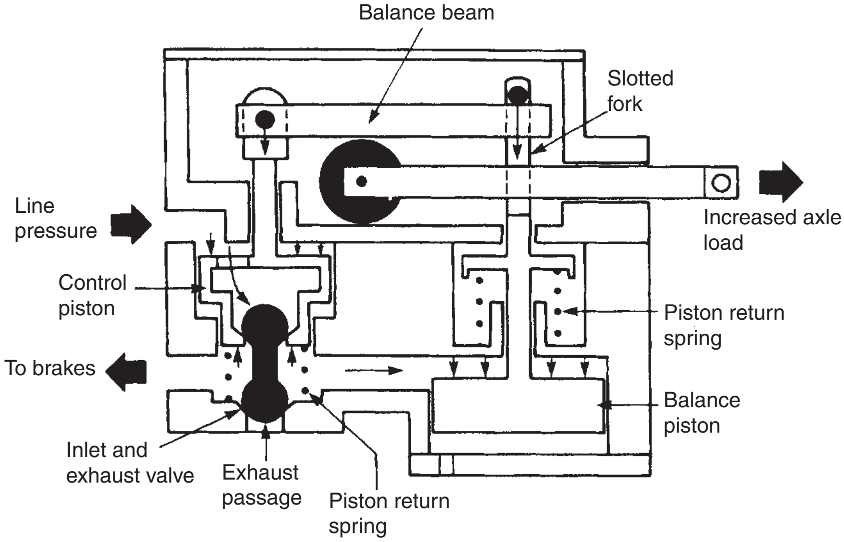


Figure 28.12 Schematic arrangement and operation of load sensing valve

to the see-saw effect of the balance beam. This upwards or return movement of the control piston continues until a point is reached where the combination valve covers both the inlet and exhaust passages (Figure 28.12). This point corresponds to a state of equilibrium between the control and balance pistons, with the air pressure in the brake actuators remaining constant. The brakes are therefore being held on with a force that is proportional not only to the effort being applied by the driver, but also to the load on the axle. Now if we increase the load on the regulated axle, the control rod of the load sensing valve will withdraw to reposition the fulcrum rollers closer to the connecting fork of the balance piston. It thus follows that when the brakes are applied, there will be a greater build-up of pressure in the actuators before it is held constant. This is because more pressure must be applied to the balance piston before a state of equilibrium is reached, owing to the less favourable leverage exerted by the balance piston on the beam.

In the Bendix type of load sensing valve the two-piece housing contains an upper assembly of sliding control piston, diaphragm and the familiar combination inlet and exhaust valve, which operates in conjunction with a lower assembly of a hollow sliding stem valve that is made responsive to suspension deflection and hence vehicle loading (Figure 28.13). There are two ingenious features associated with the operation of this type of load sensing valve. The first of these is an intermeshing arrangement of tapered radial fins for controlling the effective area of the piston diaphragm. One set of fins is fixed in the housing and the other set moves with the control piston, so that the effective area of the diaphragm and therefore the upthrust due to air pressure experienced by the piston depend upon the extent to which the fins of the latter lend support for the flexing diaphragm. Secondly, there is an air-pressure-controlled plunger that can temporarily clamp the ball-pin connection between the sliding stem valve and its torsionally flexible operating linkage. This isolates the action of the valve against minor suspension disturbances during the process of braking, while avoiding unnecessary wear in the connection by allowing it greater freedom of movement when the brakes are released.

When a brake application is made, the line pressure received by the load sensing valve therefore establishes a positive connection with the operating linkage, via an external pipe and the stem valve plunger, and also forces down the control piston until its spring-loaded combination inlet and exhaust valve is unseated by contact with the mouth of the stem valve. This results in the exhaust passage through the stem valve being closed off and the inlet passage in the control piston being opened up, so that air under pressure is not only directed through the load sensing valve to the brake actuators, but also gains access to the underside of the control piston diaphragm. When the force acting upwards on the diaphragm becomes equal to that acting downwards on the piston, the latter achieves a state of balance and the combined inlet and exhaust valve assumes a lapped condition with the air pressure in the brake actuators remaining constant. The effect of the vehicle being heavily laden will be to increase the height to which the stem valve rises in the housing. This means that the control piston will also be working in a higher position and by the same token giving less support to its diaphragm. Since this will reduce its effective area closer to that of the control piston itself, there needs to be a greater air pressure acting on the diaphragm before the opposing forces on the piston become balanced and the combination inlet and exhaust valve assumes a lapped condition. In other words, the output pressure to the brake actuators is now less reduced in relation to the input line pressure and thus corresponds with the heavier loading of the vehicle. When the brakes are released, the air pressure in the brake actuator line will lift the control piston and unseat the combination inlet and exhaust

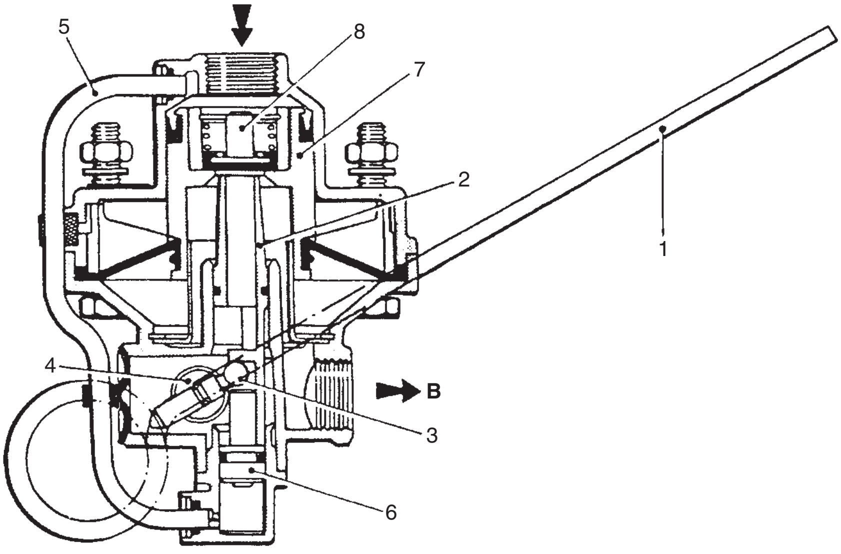


Figure 28.13 Section of Bendix Westinghouse load sensing valve (*Seddon Atkinson*)

1 operating lever
2 stem valve
3 ball-pin
4 operating shaft
5 external pipe
6 plunger
7 piston
8 inlet/exhaust valve

valve from the mouth of the stem valve, which allows the air to escape down through its exhaust passage and thence to the atmosphere, via a rubber check valve in the unit body.

Introduction to electronically-controlled air brakes

A new development in air brake system control for heavy vehicles is that in which a quicker acting electronic control is superimposed on the traditional pneumatic control, so that the latter essentially performs a safety backup function, although there are long-term development aims to make this feature redundant if legislation permits. Systems of this type are now designated as 'electronic braking systems' (EBS) or, in popular jargon, 'braking-by-wire systems'.

In basic principle the foot-operated brake valves assembly is also equipped with a potentiometer device, so that the extent to which the driver applies the brakes can be signalled to an electronic control unit (ECU). This then issues appropriate command signals to electro-pneumatic relay valves, which serve their nearby brake actuators. The function of these relay valves is first to isolate the slower acting pneumatic back-up control, second to admit air more rapidly at a regulated pressure to the brake actuators, and third to release this pressure and restore the back-up control when braking is no longer required.

By virtue of the brakes responding more quickly to the driver pressing the pedal, it will be evident that shorter stopping distances and hence improved vehicle safety is the primary objective of an electronic braking system. This type of control system can also be extended to confer other advantages, because the addition of load sensors can make it responsive to rear axle loading and load transfer during braking, so that overbraking on any particular axle can be avoided to the benefit of reduced liner or pad wear. Imbalance between tractor and trailer decelerations can likewise be sensed at the fifth-wheel coupling of an articulated vehicle or at the coupling arrangements of a drawbar combination and monitored by the ECU, which then issues command signals to the trailer electro-pneumatic relay valve that regulates the air pressure accordingly. Electronic braking systems are, of course, compatible with established anti-lock brake and traction control systems.

28.4 SYSTEM ACTUATION

Air brake actuators

These are also known as brake chambers. One is mounted externally to each wheel brake. Through the medium of a diaphragm element they convert the energy stored in the compressed air into the mechanical force and movement required to actuate the brake shoes. Owing to their bulky nature they cannot be accommodated within the brake drums, and therefore act upon either lever and cam or wedge and tappet shoe expanders instead of directly on the brake shoes.

Reference has been made in Section 27.1 to the S-cam and rollers type of brake expander and its use on air-braked heavy vehicles. A further development of the fixed-cam brake takes the form of a cam and struts type of expander mechanism. Here the shoe tip rollers associated with S-cam operation (Figure 27.4) are replaced by ball-ended struts and sliding tappets. Each strut or push-rod locates at one end in a spherical recess within the cam and at the other end in a spherical recess inside its sliding tappet (Figure 28.14). Rotation of the camshaft therefore causes the struts to separate the tappets and expand the shoe tips. The advantages claimed for this particular construction are that it provides a sealed and lubricated enclosure for the expander mechanism, and that the shoe tip

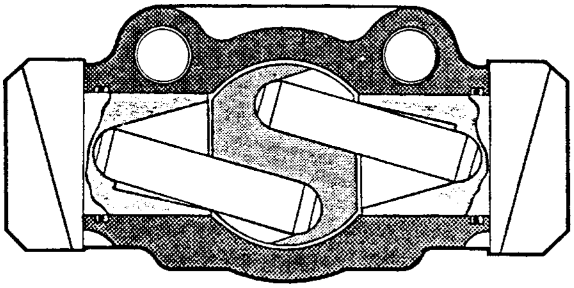


Figure 28.14 Air brake cam and struts expander (*Seddon Atkinson*)

forces are always constrained to act in-line with the sliding tappets, whereas with an S-cam and rollers the angularity of the shoe tip forces is such that their action is not independent of the direction of drum rotation.

Reference has also been made in Section 27.1 to the continuing use of wedge-type brake expanders for commercial vehicles. Those operated by compressed air for heavy vehicles were originally developed in America during the mid 1960s. The advantages of replacing the lever and cam by a wedge and tappets were perceived as reducing unsprung weight by virtue of eliminating the camshaft and slack adjuster; greater structural rigidity resulting from the brake actuator directly attacking the wedge; shorter actuating stroke with smaller air chamber, thereby minimizing air consumption and promoting quicker application and release; ready accommodation of both air brake and spring brake actuators; and allowing the use of dual-circuit air systems by incorporating twin wedge expanders in each brake. Hardly a disadvantage but more an essential requirement for an air-operated wedge brake is an in-built automatic adjuster, which ensures that the wedge travel between the sliding tappets never becomes excessive or runs out.

Various shoe arrangements may be employed with air brake wedge expanders. A single wedge expander may be used in conjunction with either a simple arrangement of leading and trailing shoes to give a floating-cam characteristic or, via a bell-crank and struts linkage, an arrangement that provides two leading shoes in the forwards direction and leading and trailing shoes in reverse, while the use of twin wedge expanders confers a two leading shoe action in both the forwards and reverse directions. Here it should be noted that in some countries these different shoe arrangements are better known as simplex, duplex and duo-duplex brakes respectively. A modern example of the latter type of air wedge brake is the Girling Twinstop. In this design the air brake actuators are stem mounted from the wedge expanders, their mounting tubes being screwed directly into the wedge expander bodies. Each expander incorporates an integral push-rod and wedge with the usual rollers interposed between the wedge and the inclined faces of its sliding tappets (Figure 28.15). The wedge also provides a positive caged guidance for the rollers, so as properly to control their movements during brake actuation. An automatic adjustment facility with a manual override is embodied in one of the sliding tappets in each expander, the opposing sliding tappet remaining solid.

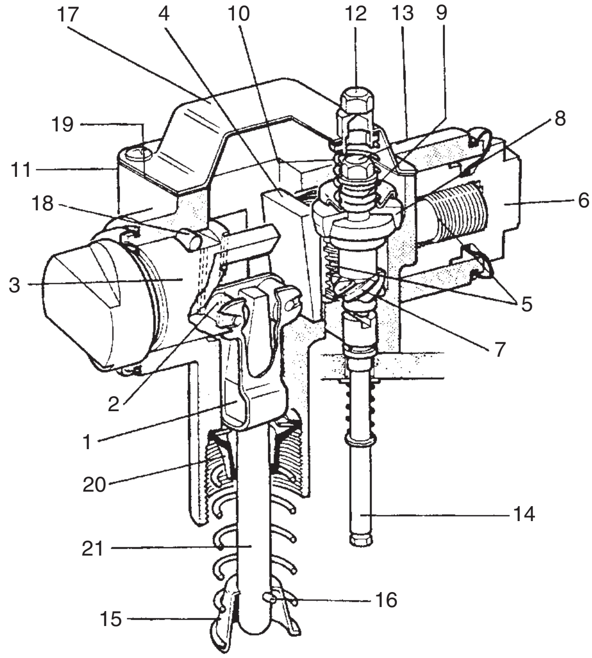


Figure 28.15 Air brake wedge expander (*Lucas Girling*)

1 wedge	12 manual override socket
2 roller	13 manual override pinion head
3 plain tappet	14 manual override stem
4 roller tappet	15 spring retainer
5 adjuster screw	16 pin/circlip
6 input tappet	17 expander cover
7 adjuster pinion	18 tappet stop pin
8 drive cone	19 gasket
9 overload spring	20 seal
10 cone spring	21 wedge push-rod
11 housing	

During normal brake application the wedge is urged inwards, via its push-rod, by the air brake actuator. It therefore imposes a separating force on the sliding tappets through the interposed rollers. Since there are two wedge expanders, each shoe receives movement at both ends and is bodily brought into contact with the rotating drum. Once this occurs both shoes move round slightly, until the trailing end of each is arrested by its adjacent sliding tappet, which has been forced back against an abutment in the expander housing. The opposing tappets then continue to transmit a shoe tip force to the leading ends of the shoes, so that a two leading shoe action is obtained. This same sequence of events does, of course, occur in the opposite sense with reverse rotation of the drum and again confers a two leading shoe action on the brake.

Of simple construction, a single-diaphragm brake actuator comprises a reinforced rubber diaphragm that is sandwiched between a two-piece pressed steel casing, which is held together by a clamping ring and provided with suitable mounting studs. The diaphragm is furnished with a push-rod connection to the shoe expander mechanism and is spring returned to the brakes released position (Figure 28.16).

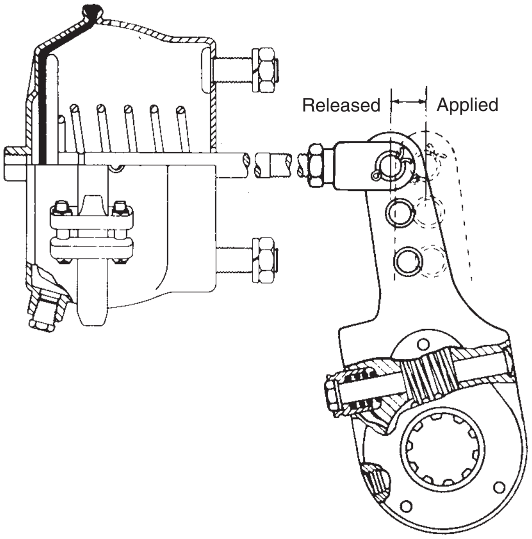


Figure 28.16 Air brake actuator and slack adjuster (*Bendix Westinghouse*)

In some installations where a rear brake actuator operates in conjunction with a mechanical handbrake, the diaphragm utilizes a pull-rod connection to the shoe expander mechanism.

In operation, the pressurized air enters the inlet port of the brake actuator and compels the diaphragm and pushrod assembly to move against the return spring and apply the brakes. The apply force developed is proportional to the effective area of the diaphragm and the pressure of the air admitted to the actuator. As the brakes are released and the air pressure reduces, the diaphragm and push-rod assembly is returned to its original position by the actuator return spring and the pull-off springs for the brake shoes.

During the mid 1960s the legally permitted gross weights of heavy vehicles were increased, and this led to various improvements being incorporated in their air brake systems to meet more stringent regulations. Included among these improvements was the use in some systems of double-diaphragm brake actuators for improved safety. With this arrangement the two diaphragms are separated at their clamping edges by a spacer ring, so that pressurized air can be admitted through a rear inlet port to the secondary diaphragm and a side inlet port to the service diaphragm. The secondary diaphragm has a forward thickening of its central portion to complement the thickness of the spacer ring, and it is also provided with a small-diameter rear lip seal that covers the rear inlet port. In the event of the service diaphragm leaking then clearly the brake can still be operated by using the secondary system, since this part of the chamber remains self-contained. The situation is a little different with a leaking secondary diaphragm, because in this case the return spring or pressurized air acting on the central portion of the diaphragm forces its sealing lip to cover the rear inlet port and prevents the escape of air into the secondary system. Since this then provides the equivalent of a self-contained chamber, normal operation of the service diaphragm is restored.

Spring brake actuators

Another improvement that was introduced into air brake systems during the mid 1960s was the secondary and parking spring brake, this type of brake having previously been used in American heavy-vehicle braking systems. A spring brake actuator utilizes the stored potential energy of a powerful compression spring to apply the wheel brake. During normal driving the spring must therefore be held in a compressed state to ensure that the brake remains released. For this purpose the spring brake actuator is provided with a supply of pressurized air via a hand control valve, the latter being described later. Since air pressure is reduced to apply a spring brake actuator and increased to apply an air brake actuator, the air brake engineer distinguishes between the two forms of air supply by referring to them as inverse air and upright air respectively.

Although air pressure is released from a spring brake for brake application, there still remains sufficient compression in the expanded spring to exert the required force for actuation of the brake. A spring brake may be either applied gradually by the hand control valve for the purpose of secondary or emergency braking, or applied fully to hold the brakes on for parking, thereby replacing the once conventional handbrake that had direct mechanical linkage to the wheel brakes. The spring brake also possesses an important fail-safe feature, because the brakes will be automatically applied should a failure occur in the air pressure circuit for the secondary and parking brake functions.

A spring brake is typically mounted in tandem with a conventional single-diaphragm service air brake, each brake operating independently of the other (Figure 28.17). The service brake chamber is therefore supplied with upright air from the footbrake valve and the spring brake chamber with inverse air from the hand control valve; simultaneous application of service and spring brakes is prevented by the differential protection valve described earlier. In construction the spring brake chamber contains either a diaphragm or a sliding piston with air seal, so that when these are subject to increasing air pressure the powerful spring is compressed to release the brake. When air pressure is reduced, either the diaphragm or the piston moves in the opposite direction under the influence of spring load. Its central stem then bears against the back of the service brake diaphragm, thereby actuating the brake through the usual push-rod connection. Some form of screw release mechanism is provided for manually relieving the brake assemblies of spring brake load, which allows the brake assemblies to be safely serviced, or if necessary the vehicle to be moved in the absence of air pressure.

Lock actuators

Introduced as an alternative to the spring brake, the lock actuator was intended to simplify the parking brake function of heavy vehicles. When signalled to do so from a hand control valve, the lock actuator will arrest the return movement of the guided stem shaft of an air brake diaphragm to hold the brakes applied. Similar to the spring brake, air pressure is released from a lock actuator to hold the brake applied (Figure 28.18).

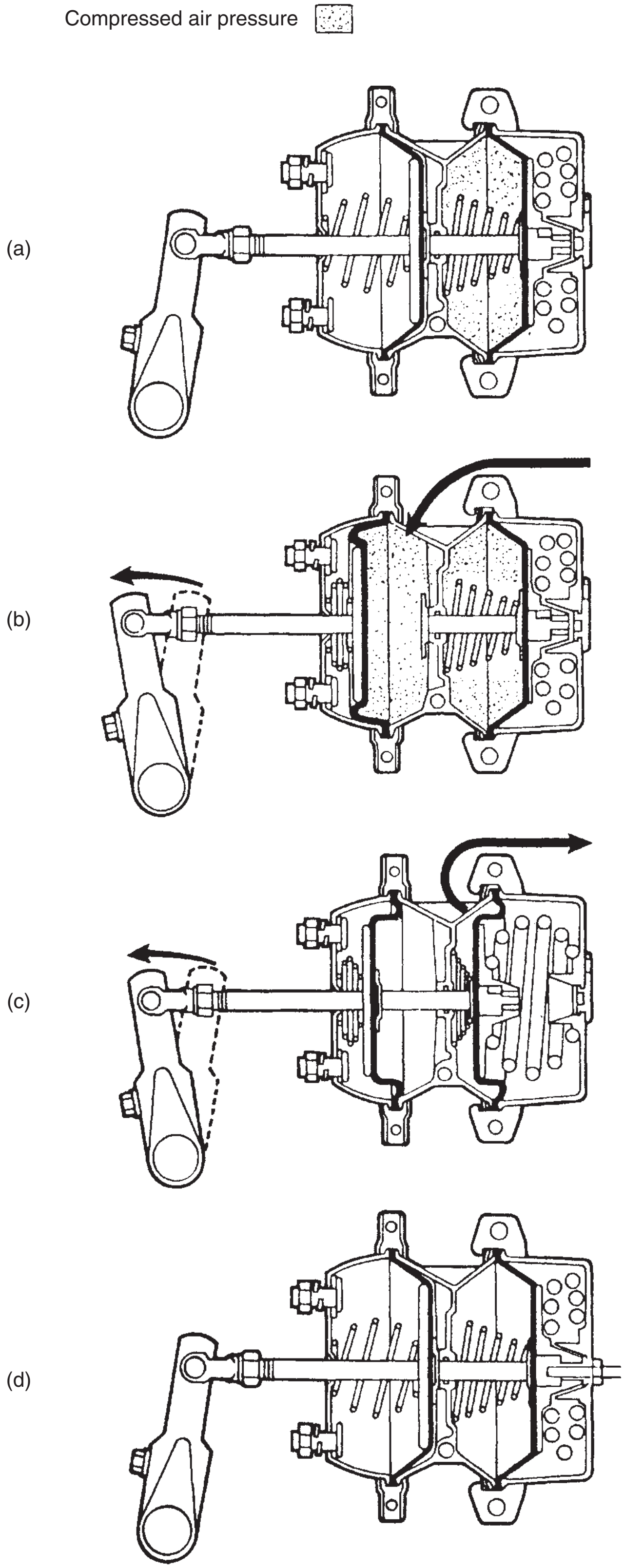


Figure 28.17 Spring brake actuator operation (*Seddon Atkinson*): (a) normal driving (b) service braking (c) secondary/park braking (d) mechanical release

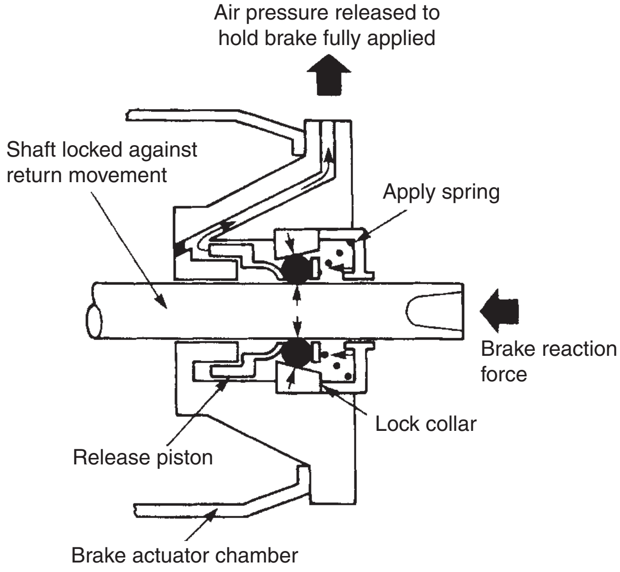


Figure 28.18 Schematic arrangement and operation of lock actuator

With a lock actuator the stem shaft of the air brake diaphragm is encircled by hourglass-shaped rollers, which can be either spring loaded against the conical bore of a lock collar so their wedging effect prevents return movement of the stem shaft, or forced away from the conical bore of the lock collar by air pressure acting against a release piston and thereby restoring free movement of the stem shaft. In a typical installation for a double-diaphragm air brake, the lock actuator is released by first applying the secondary brake to allow the rollers to unwedge themselves, following which they remain free on the shaft as long as air pressure is maintained against their release piston. The secondary brake is then, of course, released to drive away the vehicle.

Air brake adjusters

Maintaining correct adjustment in an air brake system is a matter of considerable importance, because it is related to what is sometimes termed brake force build-up time, during which the braking force at the wheels is building up to a maximum value. If adjustment is neglected and the clearance between the shoe linings and the drum becomes excessive, the amount of air that must pass into the actuator chamber to apply the brakes is necessarily greater, which therefore lengthens the brake force buildup time and increases the stopping distance.

The adjustment of cam-operated air brakes is accomplished by what is known as a slack adjuster, the mechanism of which is embodied in the lever arm used between each brake actuator push-rod and expander cam for the shoes. By means of a lockable worm and wheel mechanism the rotational position of the expander cam may be manually adjusted relative to that of the lever arm (Figure 28.16). Hence the most favourable operating geometry for the actuating linkage may be preserved throughout the life of the shoe linings, and by the same token the actuator diaphragm

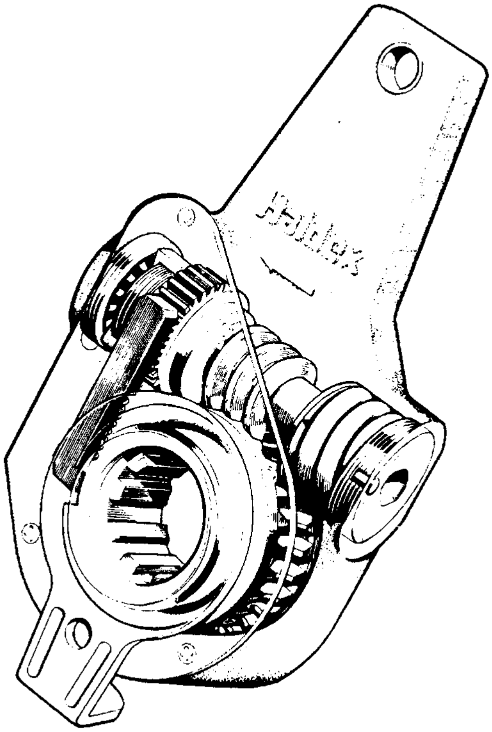


Figure 28.19 Cut-away view showing internal mechanism of Haldex automatic slack adjuster (*Haldex Division, Garphyttan Ltd*)

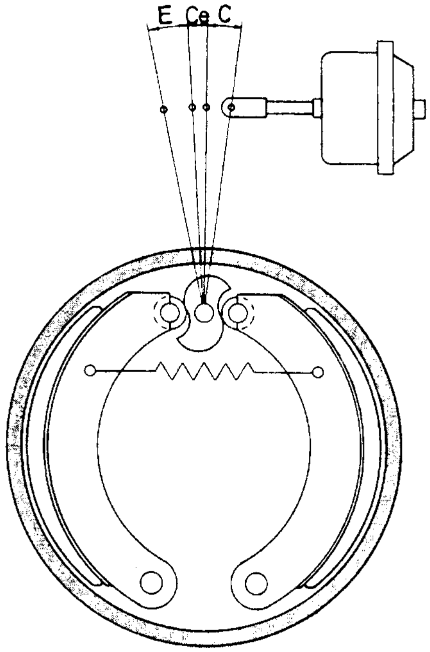


Figure 28.20 Identifying the angular movements of an air brake slack adjuster (*Haldex Division, Garphyttan Ltd*)
(C) – Clearance angle (Ce) – Excess clearance angle
(E) – Elasticity angle

is never over-extended. During normal brake operation the entire lever arm cum slack adjuster does, of course, rotate bodily with the brake camshaft.

Automatic slack adjusters have increased in popularity since the late 1970s, because they can maintain almost constant the running clearance between the shoe linings and their drums, whilst making allowance for thermal expansion of the drums during heavy braking by not taking up all the perceived excess slack. In more recent years they have become especially relevant to the responsive operation of anti-lock air brake systems, and indeed since late 1994 automatic slack adjusters have been a mandatory fitment under EC legislation for all newly registered heavy vehicles and trailers. Automatic slack adjusters basically incorporate a lost-motion ratchet and pinion drive to a spring-loaded worm shaft and friction clutch arrangement. Since the ratchet reacts against a fixed link that serves as a datum point, any excess clearance due to lining wear is taken up on the return stroke of the brake actuator. A visual indication of lining wear may also be incorporated in the mechanism of these slack adjusters. A long-established and widely used type of automatic slack adjuster for cam-operated air brakes is that known as the Haldex (Figure 28.19), which confers almost constant and correct clearance between the brake linings and the drum. To accomplish this the action of the slack adjuster is able to distinguish between three phases of angular movement (Figure 28.20) as follows:

Clearance angle (C) This corresponds to the normal running clearance between the brake linings and the drum, which is necessary to prevent any brake drag and to allow cooling between the friction surfaces.

Excess clearance angle (Ce) This is additional to the normal clearance angle and appears as the brake linings continue to wear in service, prior to the normal running clearance being restored by automatic adjustment.

Elasticity angle (E) this occurs as the brake shoes are expanded with increasing force against the brake drum temporarily dilating it, the effect of which must be excluded from the adjustment process to prevent over-adjustment and brake drag.

The take-up of the Haldex automatic slack adjuster is determined by the gear ratio of the lever arm worm and the camshaft wheel. It is, of course, most important that the manufacturer's advice be sought in relation to the correct installation and maintenance of an automatic slack adjuster.

As earlier mentioned, automatic shoe adjustment is an essential feature of the air-operated wedge brake. For modern duo-duplex versions it is accomplished by incorporating a jacking screw device within one of the sliding tappets in each expander unit. This mechanism takes the form of an adjuster screw that can be either turned directly within a threaded tappet, or translated by a threaded sleeve that can be turned within a counterbored tappet, the latter in both cases being restrained from rotation. Automatic provision is then made for either the screw itself, or its threaded sleeve, to receive a small degree of turning when the return movement of the tappet increases owing to wear of the shoe lining. This results in the shoe tip being jacked further away from the end of the sliding tappet, so that upon release of the brake the correct shoe to drum clearance is restored. The automatic means of turning either the adjuster screw, or its threaded sleeve, basically relies upon an additional helical connection being established between the sliding tappet and its adjuster, combined with a lost-motion pawl and ratchet or friction clutch control system, so that excess return movement of the sliding tappet causes a turning moment to be exerted on either the screw or its sleeve. An example of the former system is shown in Figure 28.15, where the required helical connection is established between a skew gear that turns the adjuster screw and a meshing adjuster pinion, which also provides a manual override adjustment. When the outward movement of the sliding tappet exceeds the predetermined backlash between the gear and pinion, the reaction from their helical teeth lifts and declutches the pinion against its spring loading, thereby allowing it freedom to turn; but on the return movement of the tappet the pinion drops and is held clutched, so that the helical sliding between their teeth becomes sufficient to turn the gear wheel and hence the adjuster screw. The manual override adjusters serve the threefold purpose of initially setting the shoes to drum clearance, retracting the shoes to facilitate drum removal and retracting the expanders when fitting new shoes.

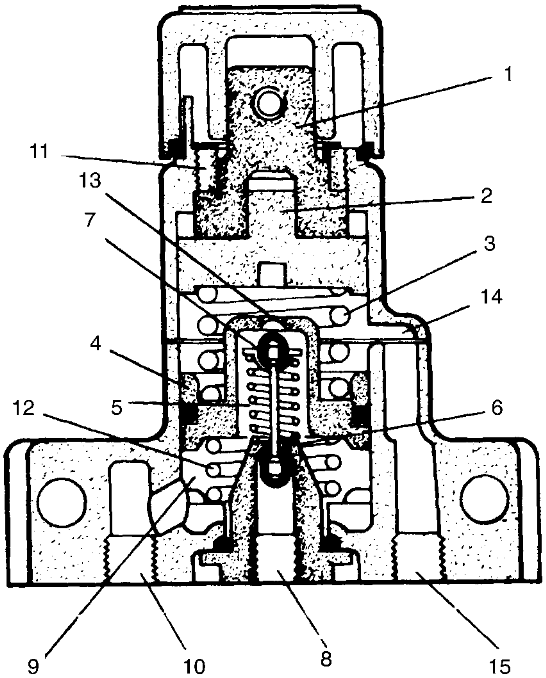


Figure 28.21 Section of Bendix Westinghouse hand control valve (*Seddon Atinson*)

1 cam follower
2 cam
3 spring
4 piston
5 inlet/exhaust valve
6 inlet
7 exhaust
8 inlet port
9 valve chamber
10 delivery port
11 adjusting ring
12 piston return spring
13 piston vent
14 valve chamber
15 exhaust port

28.5 HAND-OPERATED BRAKE VALVES AND OTHER EQUIPMENT

Hand-operated brake valves

There are several types of hand-operated brake valves used in heavy-vehicle air brake systems, their detail design varying in accordance with system requirements, but usually they are associated with the secondary and parking functions of spring brake actuators. Although the secondary brake system is only used if there is a failure in the foot-operated service brake system, it is still desirable that the hand-operated brake valve should exert a progressive controlling action over the spring brakes. In the language of the air brake engineer, this means that such a valve must provide a graduable inverse air pressure, since in applying the spring brakes for secondary braking we are concerned with gradually reducing air pressure. For the parking function all the air pressure is exhausted from the spring brakes, the handle of the valve being made self-locking in the parked position.

In construction a hand-operated brake valve typically comprises a control piston that is acted upon by a lower return spring and an upper control spring. The amount of compression in the latter is varied by a rotary cam arrangement that is positioned by movement of the valve handle, which has a friction-damped motion for sensitive control. When the spring brakes are released the control spring pressure is greatest and it is least when they are applied. Spring loaded against the inlet port of the valve body and towards the exhaust port of the control piston is again the familiar combination inlet and exhaust valve (Figure 28.21). With the hand lever in the off position, the increase in control spring pressure causes the piston to bear down on the combination inlet and exhaust valve, which uncovers the inlet port and covers the exhaust port. This allows air under pressure from the secondary/park reservoir to pass through the valve and on to the spring brake actuators. Their delivery air pressure is governed by the preset load exerted by the control spring, this pressure being sufficient to hold off the brakes for normal driving.

When the hand lever is moved from the off position (spring brakes released) and held towards the secondary/park position (spring brakes being applied), the rotary cam allows the control springs to extend and thus reduce the load on the piston. This permits the piston to rise under the influence of the air pressure and return spring force beneath it, until the spring-loaded combination valve covers the body inlet port and uncovers the piston exhaust port. Excess air then escapes through the piston into the chamber above, and from there is released to the atmosphere via the exhaust port of the valve body itself. The reduction in air pressure and therefore application of the spring brakes continues until the control spring moves the piston down sufficiently to cover both inlet and exhaust ports, following which the forces above and below the valve become equal and it assumes a balanced or lapped condition. A constant braking effect is then obtained until there is further movement of the hand lever. The operating characteristics of this type of valve are therefore similar to those of a foot-operated brake valve, but without the same sense of feel, although this feature is present in some designs of hand-operated brake valve.

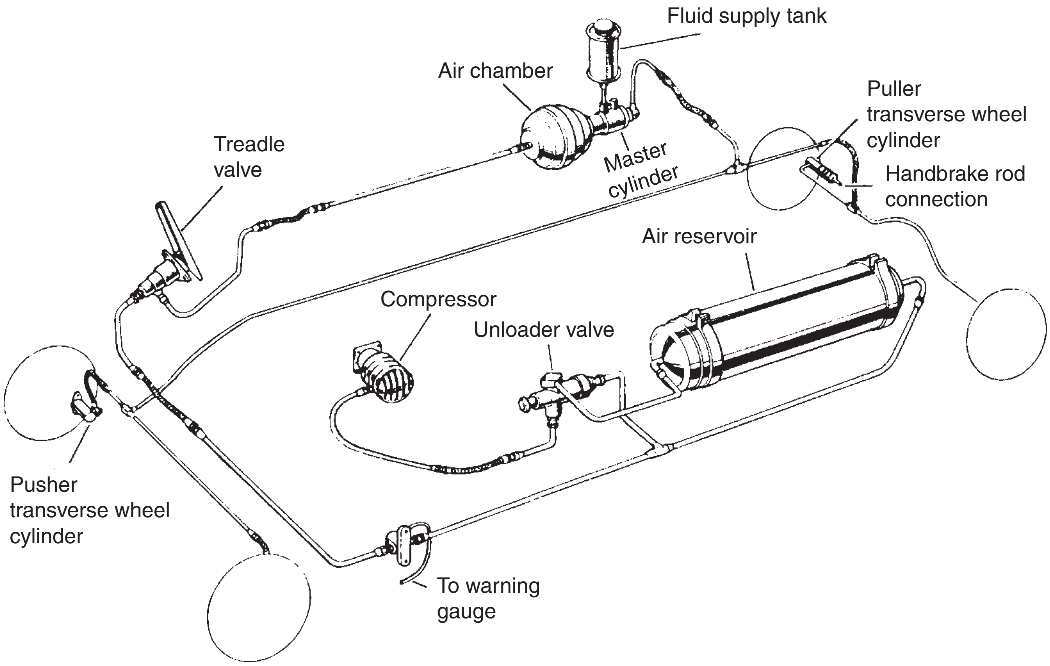


Figure 28.22 Basic layout of an air-over-hydraulic braking system (*Girling*)

Trailer connecting hoses

Since the early 1960s, precoiled nylon air lines, known as 'Susie' hoses, have generally replaced rubber air lines between tractive units and either their semi-trailers or drawbar trailers. The quick release end couplings for the hoses may be of the earlier male and female bayonet type or the later internationally used palm type, these also being known aptly as gladhand couplings in America because of their resemblance to clasped hands. Automatic shut-off valves may be incorporated in both types of coupling, so that air pressure is retained when the feed line to the trailer reservoir is disconnected. The end couplings of the hoses are designed so that they cannot be wrongly connected and the hoses themselves are colour coded for ready identification.

The colours normally used are as follows:

Yellow Service or control line, which supplies a signal air pressure to the relay valve on the trailer for normal operation of the trailer brakes.
Blue Secondary or auxiliary line, which supplies direct air pressure for secondary operation of the trailer brakes in the event of the service brake failing.
Red Feed or emergency line, which supplies air under pressure to the trailer reservoir through a relay emergency valve.
Caution It is very important that the correct procedure for safely connecting and disconnecting these brake line hoses is carefully followed, and this forms part of the driver training for goods vehicles. Reference should be made to The Official Goods Vehicle Driving Manual, published by The Stationery Office for the Driving Standards Agency.

Air-over-hydraulic brakes

For heavy vehicles in the medium weight range, hydraulic actuation of the brakes may be employed in conjunction with a compressed air source of power operation. Systems of this type are therefore known as air-over-hydraulic brakes, and can more readily generate the higher operating forces at the hydraulic wheel cylinders, which otherwise would demand unacceptably large vacuum servo units. Furthermore, the smaller compressed air chamber together with hydraulic master cylinder can be located wherever convenient on the chassis frame.

In a typical air-over-hydraulic brake system (Figure 28.22) an engine-driven compressor supplies air that is stored in the usual reservoir. When the brake pedal or treadle is depressed to operate the control valve, pressure builds up in the air chamber and acts against a diaphragm. Movement of the diaphragm and its associated push-rod then directly forces the piston of the hydraulic master cylinder to displace fluid at equal pressure to the wheel cylinders and thus apply the brakes.

28.6 AIR DISC BRAKES

General background

Following the successful adoption of hydraulically operated disc brakes for passenger cars that began in the mid 1950s, uprated versions of this type of brake were later developed and initially installed on public service vehicles, a notable example of their use being on the motorway coaches of the Midland Red Bus Company in Britain in the early 1960s. Operator experience at the time had tended to show that, so far as public service vehicles were concerned, the disc brake was best suited for long-distance hauls where brake temperatures were lower than those encountered in city running. Hence it was soon recognized that the key to successful operation of disc brakes on commercial vehicles was ensuring adequate dissipation of heat from their discs, otherwise both pad and disc life would become unacceptably short. To meet this requirement in practice meant that the disc had to be of ventilated construction, and designed with the maximum outer and minimum inner diameters together with the maximum thickness which could be installed within the existing wheel space, there being greater difficulties posed by the axle arrangements of commercial vehicles than had previously been the case for passenger cars. It also meant that further research and development had to be undertaken on disc pad materials to provide an increased life. For these reasons it was not until the early 1970s, following a renewed interest in their development in America, that manufacturers elsewhere began to introduce improved designs of disc brake which, together with more suitable friction materials, has led to their gradual acceptance on light to medium commercial vehicles.

The introduction of air-operated disc brakes for heavy vehicles is of more recent origin, because in their case additional design requirements had to be taken into account, not the least of these being the need for a force multiplying mechanism between the disc pads and the air brake actuator chamber. This requirement arises from the relatively low operating pressures of an air brake system, as compared with those of a hydraulic one, which would otherwise demand unrealistic increases in the size of the caliper piston and the amount of air to be moved. That is, the caliper could not accommodate the size of piston required and the brake force build-up time would be unacceptable. Since the presence of a force multiplying mechanism implies a large increase in brake actuator stroke for a small wear reduction in pad thickness, there is an essential requirement for an automatic adjuster that must necessarily be of the mechanical type.

However, the modern air disc brake can offer important advantages over those of an equivalent drum brake, which in general terms are greater structural rigidity, more consistent brake torque, better fade resistance and easier access for maintenance.

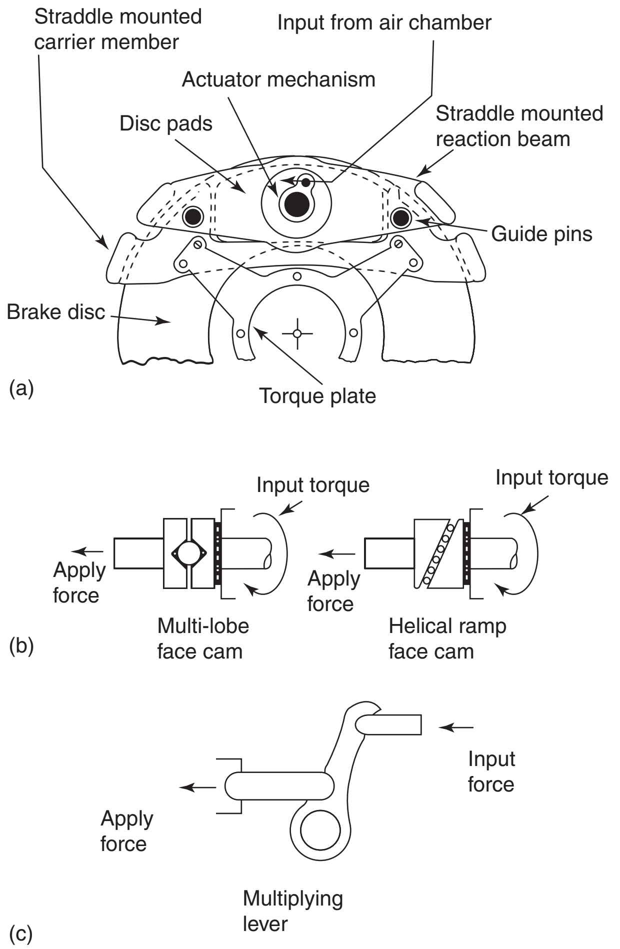


Figure 28.23 Basic arrangement of an air-operated disc brake: (a) straddle-mounted caliper (b) and (c) force multiplying actuators

Basic construction of air disc brakes

Air disc brakes for heavy vehicles have a basically similar clamping action to later versions of the floating caliper and cylinder, or first-type, hydraulic disc brake used on passenger cars (Section 27.2). However, the construction of the sliding caliper is modified to incorporate a more robust straddle-mounted reaction beam or bridge, instead of a simple claw, to provide an abutment for the outboard pad. The reaction beam is extended chordally across the face of the disc and straddles it at each end, where it unites with the caliper housing (Figure 28.23a). This structural change to the caliper ensures that the outboard disc pad receives a centrally applied clamping load. The backing plates for the outboard and inboard disc pads therefore reside against the caliper reaction beam or bridge and an opposite load spreader plate, which is urged towards the disc by the caliper actuating mechanism when the brake is applied. Equality of clamping loads is achieved by virtue of the friction drag forces on both disc pads being reacted against locating lugs, which are formed integral with a straddle-mounted carrier member (Figure 28.24). This is bolted either directly or indirectly to an axle flange, an intermediate torque reaction plate being used in the latter case. The carrier member also incorporates the sealed low-friction guide or slide pins, which support the freely floating caliper assembly (Figure 28.24). Quadruple slide pins are employed in the established Haldex air disc brake, these minimize any tendency for brake drag while improving the durability of the sliding components. The avoidance of brake drag with its adverse effects on both fuel economy and brake wear, is an important consideration to the transport engineer.

Actuators and automatic adjusters for air disc brakes

The actuator housing for the force multiplying mechanism is either integrated with, or separately bolted to, the inboard

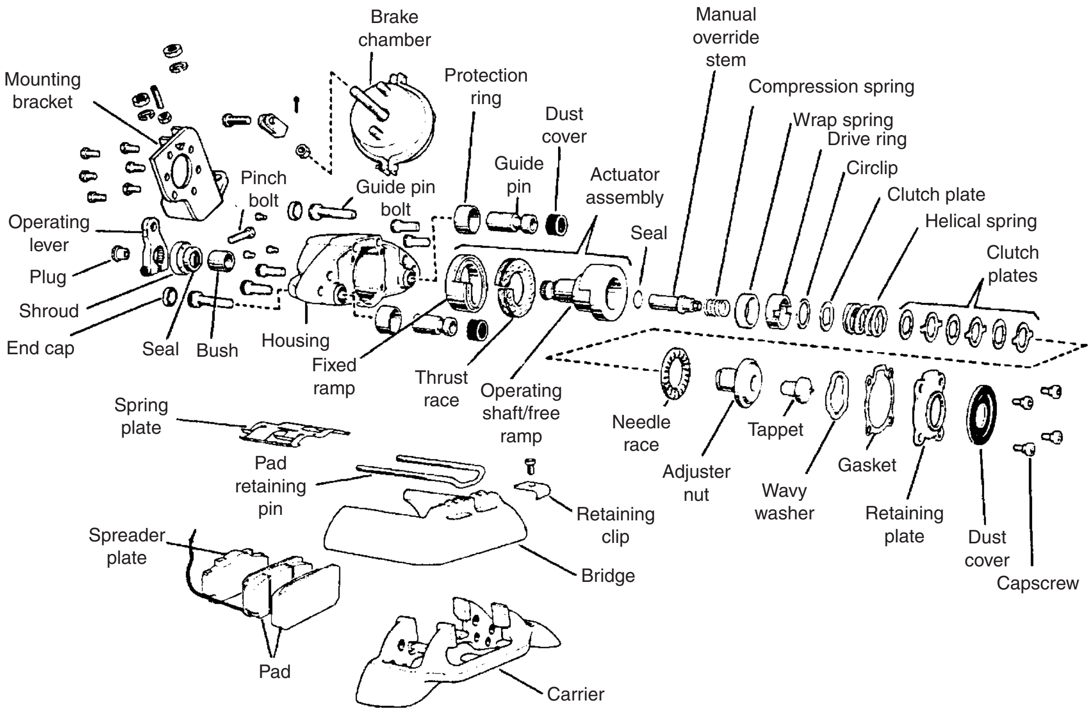


Figure 28.24 Exploded view of Girling reaction beam caliper brake (*Lucas Industries*)

side of the floating caliper and also provides a mounting for the air chamber (Figures 28.24 and 28.25). This is positioned either parallel with, or normal to, the disc face according to the type of force multiplying mechanism used. The former position may be likened to that used for operating rotary cam air drum brakes (Section 28.4), and similarly requires the intervention of a clevis joint where the air chamber push-rod connects to the brake operating lever, the cross-shaft of which transmits a torque input to the force multiplying mechanism (Figure 28.24). This therefore serves not only to convert the partial rotation of the input shaft into a linear motion for the caliper tappet (corresponding to the piston in a hydraulic disc brake), but also to multiply further the actuating force supplied by the air chamber, so that the clamping load imposed on the disc pads generates the required brake torque.

The force multiplying mechanism itself may comprise a pair of opposing face cams with either multi-lobes, or helical ramps, and rolling bearings interposed between their thrust surfaces (Figure 28.23b). An equivalent mechanism used in an American design of brake features a multi-start screw thread on the cross-shaft that engages an externally splined sliding nut. Hence, when the cross-shaft is turned relative to the rotationally fixed cam of the pair, or the sliding nut, the resulting linear movement creates an end-thrust on the load spreader plate for the inboard disc pad. Then by virtue of the caliper being free to slide, this end thrust is also transmitted to the outboard disc pad, so that both pads engage the disc for braking.

The alternative construction for air disc brakes, where the air chamber is mounted normal to the disc with a lever type force multiplying mechanism (Figure 28.23c) may similarly be compared to air drum brakes with wedge instead of cam expanders for the shoes (Section 28.4). Likewise, the more direct actuation of the disc pads can make for a compact design of brake with a reduction in unsprung weight and a minimum of wearing parts. A modern example of this type of construction is the previously mentioned Haldex DB20 air disc brake (Figure 28.25). Before describing the force multiplying mechanism of this heavy vehicle brake, it may be of interest to quote a few items of dimensional data to put its size into perspective, as follows:

Road wheel size	22.5 in
External diameter of brake disc	430 mm (16.9 in)
Thickness of brake disc (new)	45 mm (1.8 in)
Effective radius of brake disc	172.6 mm (6.8 in)
Swept area of brake disc	1808 cm^2 (280 in^2)
Lining area of each pad (2)	187 cm^2 (29 in^2)

The air chamber is offset above the centre-line of the disc pads, so that its spherically-ended push-rod can engage an upstanding lever. This acts eccentrically upon a wide cross bar, to confer a 15.8:1 mechanical advantage for the apply force at the disc pads. To reduce internal friction partially caged needle roller bearings are used at the pivot points for the straddle-mounted lever and its associated cross bar (Figure 28.25). The force multiplying mechanism is designed to accept a maximum brake chamber force of 13.9 kN (3127 lbf). A spring brake chamber is, of course, used in cases where the disc brake has a parking function.

Brake application (Figure 28.26) The upstanding lever (44) is actuated by the air pressure in the brake chamber (25/26). Since the external and internal radii of the lever (44)

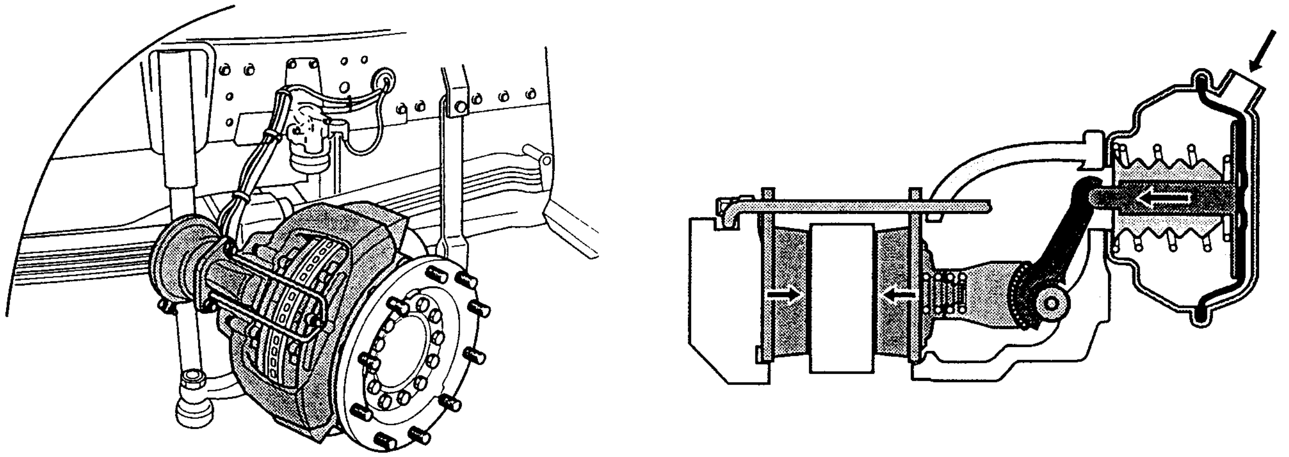


Figure 28.25 Installation and general arrangement of Haldex DB20 air-operated disc brake (*Haldex Brake Products AB, Sweden*)

do not have a common centre, their eccentricity means that the cross bar (41) is forced to move axially in the direction of the brake disc (A). This force is then transmitted from the cross bar (41), which also incorporates twin adjusting screws for setting the predetermined clearance between pads and disc, to the caliper tappets (28) for the inner pad (5). Once this pad (5) comes into contact with the brake disc (A), the caliper (2) is moved on its slide pins so that the outer pad (3) likewise comes into contact with the brake disc.

Brake release (Figure 28.26) The return spring (38) forces the cross bar (41) back into its rest position, so that the design clearance between the pads (5) and the brake disk (A) is restored.

In principle there are two types of automatic adjuster, which may be classified as ‘stroke sensing’ and ‘clearance sensing’. The former type ignores the clearance existing between the friction elements, and adjusts the brake when the air chamber stroke exceeds a predetermined limit. In contrast, the latter type adjusts the brake when the actual clearance between the friction elements exceeds a predetermined limit. Hence, it follows that the possibility of the brakes being over-, or under-, adjusted is less likely to occur where a clearance sensing system is used.

The Haldex DB20 air disc brake operates according to the clearance sensing principle. Similar to identifying the angular movements of their automatic slack adjuster for air drum brakes (Figure 28.20), the braking sequence is divided into three phases (Figure 28.26). Namely, Design clearance ‘C’, Excess clearance ‘Ce’ (which is to be adjusted out) and Elasticity ‘E’. Basically, the housing of the adjuster (62) is held in position against the internal radius of the upstanding lever (44) by a guide pin (47). This compels the adjuster housing (62) to follow the movement of the lever (44). The rotational motion is transferred from housing (62) to adjustment spring (63), which in turn, after passing the design clearance, transmits the motion to the companion sleeve (65), friction spring (66) and hub (67). If excess clearance is present, the rotation of the hub actuates a synchronized bevel gear drive for the twin adjustment screws incorporated in the cross bar (41), which then restore the correct clearance between the pads and disc.

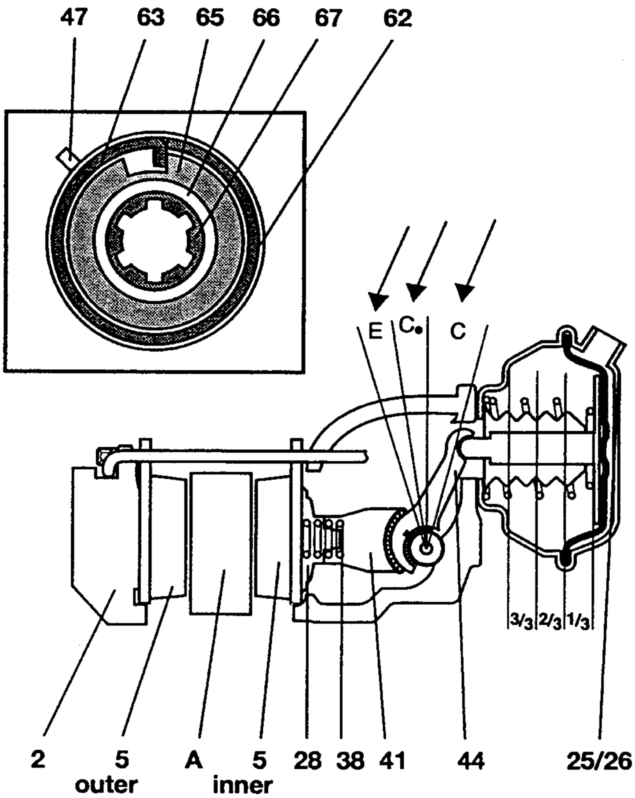


Figure 28.26 Application, release and automatic adjustment of Haldex DB20 disc brake (*Haldex Brake Products AB, Sweden*)

Caution It is, vitally important to consult the brake and vehicle manufacturers’ instructions when servicing the air disc brakes of heavy vehicles, which should only be performed by trained personnel.

28.7 ENDURANCE BRAKE SYSTEMS

General background

This is a relatively new term that has been introduced by European legislation into heavy vehicle braking technology, and embraces the use of what have long been more familiarly known as 'auxiliary retarders'. An auxiliary retarder as its name suggests may be defined as a vehicle-slowing, rather than a vehicle-stopping, device and is not therefore a substitute for the service braking system.

Reference has earlier been made to the heat fade resistance of brake friction materials (Section 27.3). In general this is of a high standard, but if extremely high temperatures are reached during prolonged spells of braking then the loss of friction can seriously reduce the effectiveness of the brakes and result in brake fade. This can pose very real safety problems in the case of heavy vehicles operating under exceptionally hilly conditions and not least those carrying hazardous goods, since the diesel engine lacks intake manifold depression to create a retarding effect on the overrun. In any event the additional work now required of the service brakes has been heightened still further by changing operating conditions. The ZF Company of Germany have identified these as follows:

Higher vehicle gross weights
Cargo volume maximized by fitting smaller diameter wheels with smaller braking friction areas, thus offering reduced service braking efficiency
Increased driveline efficiency
Improved Cd-value and, therefore, reduced aerodynamic drag (Section 30.8)
Better tyres affording reduced rolling resistance
Greater traffic volumes and irregular traffic flows have led to more frequent braking

Types of auxiliary brake retarder

Various types of auxiliary retarder for augmenting the friction brakes of heavy vehicles have therefore been devised, beginning with the Swiss-designed Oetiker exhaust brake that first appeared in 1925, and they may be classified as follows:

Exhaust brake retarder
Compression brake retarder
Hydraulic retarder
Electromagnetic retarder
Friction retarder

In modern practice auxiliary retarders generally have to be compatible with anti-lock air brake systems, their retarding effect being interrupted when ABS is in operation.

Exhaust brake retarder

The exhaust brake represents a widely used, relatively inexpensive and compact form of retarder. It essentially comprises a shut-off valve in the exhaust system (Figure 28.27) which enables the engine of an overruning vehicle to act as a low-pressure air compressor, so that the work done by the pistons in compressing the air exerts an additional retarding torque on the crankshaft and hence a braking effect on the transmission and wheels.

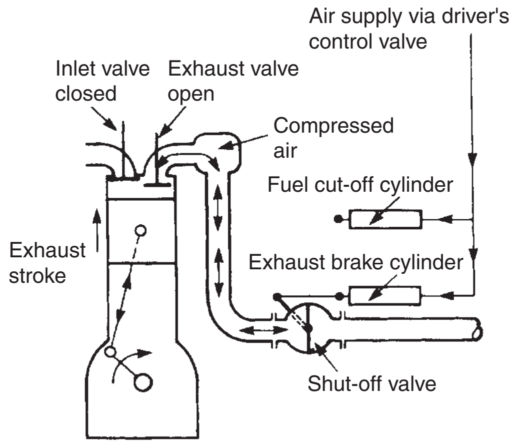


Figure 28.27 Schematic arrangement of an exhaust brake retarder

Rotary butterfly or sliding gate types of shut-off valve are used in the exhaust brake and must be carefully designed to avoid seizure, and also not to impede the normal flow of exhaust gases when in the open position. They are installed in the exhaust system between the manifold and silencer or, in the case of turbocharged engines, between the manifold and turbocharger. The shut-off valve may be operated either manually or automatically and in both cases provision must be made to cut off the fuel injection before the exhaust brake is applied, so that pure air only is compressed in the exhaust system. This occurs on what would normally be the exhaust strokes of the engine pistons and continues to the point when the inlet valves begin to open. These then allow the air compressed in the exhaust system at about 280–350 kN/m^2 (40–50 lbf/in^2) to escape via the cylinders into the intake manifold and thus equalize pressure before the cycle is repeated. An outstanding advantage of this type of retarder is that changing down to a lower gear will multiply the retarding torque imposed on the road wheels, because this increases with engine speed.

Apart from an increases in the life of the brake linings and drums that is to be expected from the use of any retarder, other advantages usually claimed for exhaust brake operation are those of reducing the danger of engine overspeeding on the overrun, and better maintaining engine temperature under these conditions because in compressing the air the engine is converting mechanical energy into heat. Possible disadvantages are related to any leakages developing in the exhaust brake installation and that maximum retarding torques are limited by engine displacement. A further consideration is that pulsations may be created in the engine intake system, which arise from a combination of cylinder pressures and intake valves opening during exhaust braking and can induce dirt to migrate through the air cleaner element, particularly if this is of the dry type. The use of a suitably tuned wave suppressor installed between the engine and air cleaner may therefore be required to dampen these pulsations,

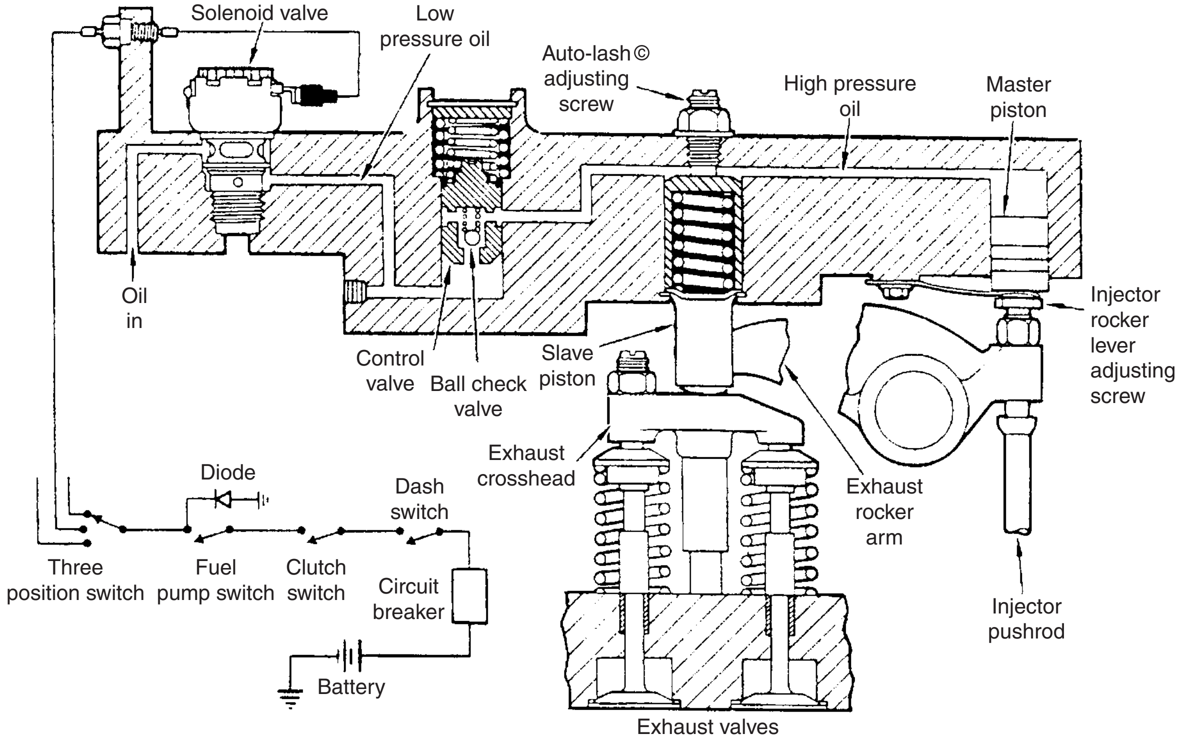


Figure 28.28 Schematic arrangement of Jacobs Engine Brake (*Jacobs Europe*)

this normally being a decision that is taken by the engine manufacturer.

Compression brake retarder

The engine itself has to receive some modification for installing this type of auxiliary retarder, which when operated, temporarily converts the power-producing diesel engine into a power-absorbing air compressor. A widely used retarder of this type is the Jacobs Engine Brake, which was introduced in 1961 and is universally known to transport engineers as the 'Jake Brake' (this being in fact a registered trademark of the Jacobs Vehicle Equipment Company in America). The background of the Jake Brake is worth a brief mention, because it was originally developed by C.L. Cummins, the American automotive diesel engine pioneer referred to earlier (Section 8.8). History records that he narrowly avoided a serious accident at a freight train crossing on a mountainous downgrade in California, when the brakes of an early diesel-engined truck he was demonstrating failed to hold. It was this experience which convinced him of the need for an independent engine brake, so that less reliance had to be placed on the brake linings and prayer under such circumstances.

Considering first the effect of engine compression braking without a Jake Brake, when the vehicle is driving the engine on the overrun, energy is transferred from the driving wheels into performing the engine compression strokes, which thereby hinders rotation of the wheels and creates a retarding force on the vehicle. However, this beneficial braking effect is negated on what would normally be the ensuing engine power strokes (the fuel injection being cut off), because when the pistons pass their TDC positions and begin their downward strokes, the energy stored in the compressed and heated air in the cylinders is returned to the pistons and hence helps, rather than hinders, the rotation of the driving wheels. In other words we have a swings and roundabouts situation where essentially no energy is absorbed and no net retarding work is done on the vehicle.

Considering next the effect of engine compression braking with a Jake Brake, the ingenuity of this device resides in its ability to release the air pressure in the cylinders in what is termed 'exhaust blowdown', before completion of their compression strokes. Essentially no stored energy is therefore returned to the pistons on what would normally be their power strokes, which otherwise would help rotation of the driving wheels. Stated another way, there is a positive loss of energy from the compressed and heated air to the engine exhaust and cooling systems, which represents the retarding work done on the vehicle.

At the heart of the Jake Brake is an electrically-controlled hydromechanical actuating system (Figure 28.28) which opens the exhaust valves in each cylinder as the piston approaches the end of its compression stroke, thereby releasing the compressed air by exhaust blowdown. A transfer of motion is therefore required between either the injector valve rocker adjusting screw of a common-rail unit injector (Section 8.8 and which betrays the American origins of the Jake Brake) or, less commonly, an exhaust valve rocker adjusting screw of an alternate cylinder. Either arrangement provides the necessary synchronizing link between engine camshaft motion and compression brake timing for the engine cylinders.

When the Jake Brake is operated the all-important exhaust blowdown process occurs as follows:

1 Energizing the solenoid valve allows engine lubricating oil to flow under pressure into the passageway connecting both the master piston and the slave piston.
2 The oil pressure acting on the master piston causes it to move downwards, until it comes into contact with the injector valve rocker adjusting screw.
3 When the injector valve rocker is activated as in a normal injection cycle, its rising adjuster screw forces upwards the master piston. This has the effect of directing oil under increased pressure to the slave piston, because the ball check in the control valve closes and traps the oil for motion transfer.
4 The high-pressure oil therefore compels the spring-returned slave piston to move downwards and intercept the exhaust valves cross-head (Section 2.6). Hence, the exhaust valves are momentarily opened, which releases the compressed air in the cylinder to the exhaust system and the atmosphere.
5 Since this blowdown of compressed air to atmospheric pressure precludes the return of energy to the engine piston on what would normally be its power stroke, the work done in compressing the air to provide auxiliary retardation is not returned during the expansion process and therefore cannot detract from the braking effect.

The manufacturer's safety precautions should be strictly observed in relation to the installation, operation and maintenance of the Jake Brake.

Hydraulic retarder

Of the various retarders used, this type, together with its associated cooling equipment, generally tends to represent the most expensive and often the heaviest installation. The hydraulic retarder operates on the principle of a fluid coupling that is being deliberately kept stalled, so that the vortex motion of the circulating fluid converts into heat some of the mechanical energy being transmitted by the propeller shaft and therefore exerts a braking effect on the road wheels. In construction the hydraulic retarder differs from that of a conventional transmission fluid coupling, as described earlier, in comprising a propeller-shaft-driven double-sided impeller, which is rotated between two inwardly facing and permanently fixed turbine members (Figure 28.29). In hydraulic retarder terminology, the impeller and turbine members are referred to as the rotor and stators respectively. The purpose of this duplex arrangement with a double-sided rotor and twin stators is to increase the maximum braking effect of the retarder.

During normal driving when braking is not required the retarder is empty of oil, but when it is brought into operation, typically by initial depression of the normal brake pedal, the braking effect can be closely controlled according to the amount of oil admitted to the retarder via a pneumatically actuated valve. To dissipate the heat being generated by the retarder, arrangements are made for the fluid in circulation to be passed through either a heat exchanger in the engine cooling system, or a separate oil-to-air heat exchanger in the case of an air-cooled engine. The main advantage of the hydraulic retarder is that it develops a much greater braking effect than other types, and especially from high speeds. In common with other drive line retarders it still remains effective in the event of an engine failure or a missed downward gear change.

This type of drive line retarder can be installed as either a 'remote' or a 'close-coupled' unit. In the former case it may conveniently replace the intermediate bearing of a divided propeller shaft, while in the latter it can be integrated with the transmission gearbox as in the modern ZF-Intarder and driven through step-up gearing to gain high retarding torques even at low propeller shaft speeds (Figure 28.30). An incidental advantage of this particular arrangement is that there can be a shared oil circulation and heat exchanger system between the transmission gearbox and the Intarder, which reduces oil ageing by virtue of continuous cooling.

Electromagnetic retarder

In principle, the electromagnetic retarder imposes a braking effect on the vehicle drive line and hence the road wheels that is derived from magnetic drag forces. These are created

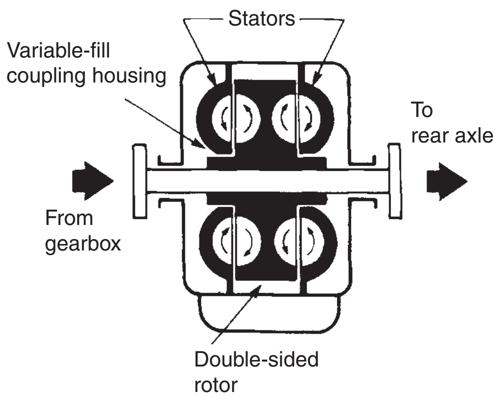


Figure 28.29 Schematic arrangement of a hydraulic retarder

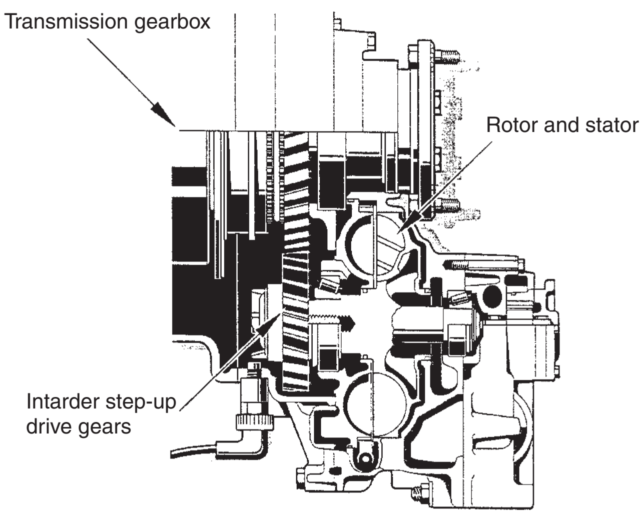


Figure 28.30 Cutaway view of integral ZF-Intarder (*ZF GB*)

by what is called an eddy current effect, the eddy currents being induced in either a single or a pair of rotors by a stator member (Figure 28.31) which comprises a series of electromagnetic pole pieces and is energized from the vehicle electrical system. Since the eddy currents may be regarded as short-circuited electrical currents, they likewise produce a heating effect in the rotor(s), so that some of the mechanical energy being transmitted by the drive line is converted into heat as part of the retardation process. This heat is dissipated to the air by virtue of the propeller-shaft-driven rotor(s) being provided with impeller vanes and therefore acting also as a cooling fan. During normal driving, when braking is not required, no current flows through the electromagnetic pole pieces and the unit remains inactive.

A long-established example of this type of auxiliary retarder is the Telma unit that consists of only two basic components.

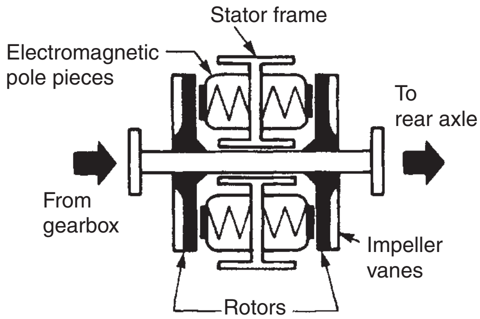


Figure 28.31 Schematic arrangement of an electromagnetic retarder

These are the fixed stator assembly, which can be either mounted directly to the gearbox or to the driving axle, or mid-mounted independently between the chassis frame members (Figure 28.32), and the rotor assembly that is permanently coupled to the propeller shaft. There is, of course, no physical contact between these two assemblies, so that a friction-free operation is ensured. Among the advantages of the Telma retarder are that it is relatively easy to retrofit and by its nature offers optimum integration with ABS systems. It is also self-cooling and self-regulating, its retarding ability being independent of engine speed or gear selected. Telma retarders for buses are automatically controlled through the footbrake and so are used in every brake application.

Friction retarder

Less commonly, a friction-type retarder can be used that takes the form of a wet multiplate friction clutch, which in some cases may be incorporated in the vehicle gearbox (Figure 16.5). For the installation shown the multiplate clutch is conveniently arranged so that it can either be fully engaged to operate reverse gear, or partially engaged via the brake pedal control to act as a friction retarder in the transmission system. This particular arrangement therefore contributes to a compact installation with little increase in weight for the retarder. The stationary plates of the clutch are produced from cast iron and the rotating ones from sintered bronze. A separate lubrication system is used in conjunction with a heat exchanger to ensure adequate cooling for the retarder. Advantages of the friction type of auxiliary brake retarder are those of providing a rapid response and maintaining an effective control until the vehicle is brought to a standstill.

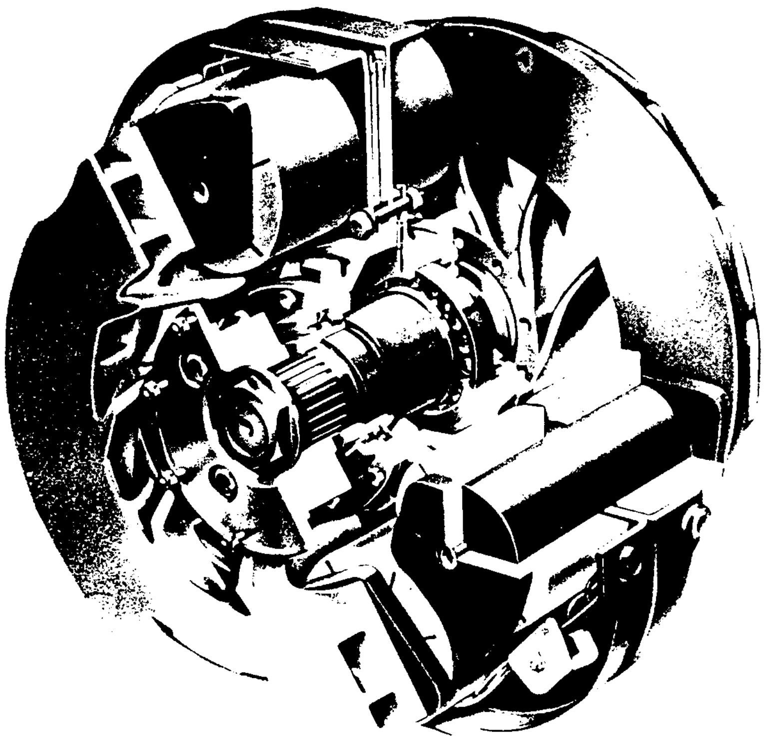


Figure 28.32 Telma electromagnetic retarder for mid-mounting in drive line (*Telma Retarder*)

29 Anti-lock brakes and traction control

29.1 BACKGROUND TO ANTI-LOCK BRAKING

One of the most serious hazards of driving and a significant contributor to the accident toll is the locking of wheels under heavy braking on slippery surfaces. The consequent loss of braking efficiency will greatly increase the braking distance and, coupled with this, will be loss of steering response if the front wheels lock and even worse a loss of directional stability if the rear wheels lock. Furthermore, the vehicle will exhibit an unwelcome sensitivity to road surface irregularities and camber, so that even small disturbing forces can cause it to slew sideways, making control even more difficult. In other words, if a wheel is braked to the point of locking its sideways grip is reduced to zero.

It was mentioned in Chapter 27 that to achieve the shortest emergency distance, all wheels need to be almost on the point of locking. This is because the rolling friction developed by a tyre that is still exerting a rotational grip against the road surface is always greater than the locked wheel friction developed by a tyre that is simply sliding over the road surface, especially when this is wet and slippery. Experienced drivers have long recognized that emergency stopping under these conditions is best accomplished by what is known as cadence braking, the efficacy of which can readily be verified on a skid pan. This technique involves rapid pumping of the brake pedal, so that steering control is retained with each momentary release of the brakes, whilst stopping distance is being shortened as much as possible.

Anti-lock braking systems seek automatically to reproduce a similar cycle of events by means of wheel speed sensors. These detect the point at which a wheel is about to lock and signal the intention to a control mechanism, which momentarily either holds constant or actually reduces the braking force to the wheel concerned, the off-on braking cycle being repeated as long as the wheel is about to lock. Ideally then, an anti-lock braking system should be even more swiftly acting than an experienced driver, not only in minimizing straight-line stopping distances on slippery surfaces, but also and more importantly in allowing the driver to avoid obstacles by retaining steering control where otherwise this would be lost with locked wheels.

It is perhaps of interest to recall that the basic idea of what was then called an anti-skid control was first employed for railway braking systems by the Westinghouse Air Brake Company in America, but it was in the further development of this idea and its application to the disc brakes of aircraft that this same company found the widest demands in the late 1940s. By the early 1950s similar anti-skid controls, such as the Dunlop Maxaret, were also being developed for the aircraft industry. So it was as a result of the experience gained in this field that the possibility of adapting the Maxaret system for use on motor vehicles was investigated by the Dunlop company and an installation tested by the Road Research Labortory. At this juncture it must be appreciated that the objectives sought in applying the wheel anti-lock principle to aircraft and motor vehicle braking systems were somewhat different. With an aircraft the objectives are to prevent at least expensive tyre damage and at worst a potentially dangerous tyre burst, by avoiding wheel locking however briefly it may occur during maximum braking through the landing run. In comparison with an aircraft tyre, the tyre of a motor vehicle is relatively lightly loaded and any damage caused to it by wheel locking is far less significant, but what is really important is any loss of directional control or stability as already mentioned.

Early thinking on anti-lock brake systems for motor vehicles was therefore directed not only towards passenger car applications, but also for commercial vehicles with the jack-knifing problem on articulated ones especially in mind (Section 30.5). Although the Dunlop Maxaret anti-lock principle was eventually adapted to the braking system of the specialist Jensen FF four-wheel-drive car in 1966, it was not until the advent of electronic control systems and electromagnetically sensed wheel speeds, beginning with the introduction of the Anti Blockier System (ABS) in 1978 by Robert Bosch working in conjunction with Mercedes-Benz, that the development of anti-lock braking systems started in earnest and has led to their increasing adoption. Indeed, since 2004 they have become standard equipment on passenger cars in Europe and their operating requirements are defined by regulation.

Finally, it should be mentioned that not all authorities agree with the designation 'anti-lock brake system'. The Society of Automotive Engineers in America originally preferred the term 'wheel slip control system', defining this as 'a system which automatically controls rotational wheel slip during braking'.

29.2 BASIC COMPONENTS OF ANTI-LOCK BRAKE SYSTEMS

The various components used in anti-lock braking systems and their purpose may be summarized as follows.

Wheel speed sensors

As their name suggests, these constantly measure wheel speeds and signal this information either electronically or, less commonly, mechanically to the anti-lock control system.

An electromagnetic sensor comprises a coil winding with a permanent magnet core and pole pin, the latter being accurately aligned in a radial or an axial position with a toothed sensor ring, but separated from its teeth by a small although critical air gap of typically 1 mm (0.040 in). A shrunk-on mounting for the sensor ring can be furnished on the hub flange of a non-driven wheel, or on the neck of a constant-velocity joint housing for a driven wheel, while in some cases it may be mounted from the pinion shaft of the final drive to serve both rear wheels. It cannot be mounted from parts directly subject to brake heating as this would affect the magnetic properties of its metal. The teeth of the sensor ring therefore act as a signal generator, so that when each passing tooth faces the pole pin of the sensor the changing magnetic field induces a voltage across the winding. Above walking pace this produces a frequency of signal that varies in proportion to road wheel speed and thus supplies the required input to the control system.

A mechanical sensor depends on the rotational inertia of a small flywheel to create a signal of imminent wheel lock during braking. Each freely mounted flywheel receives its drive through a spring-loaded ball-and-ramp clutch mechanism, the input shaft for which can be conveniently belt driven from a transmission drive shaft so that wheel speed can be sensed. In operation the flywheel will tend to overrun the shaft driving it when the rotational deceleration of its associated road wheel exceeds a predetermined amount. The inertia of the flywheel will then compel the balls to roll up their ramps, thereby creating an axial component of force within the clutch mechanism. This movement of separation is directly utilized by the control system as a signal of imminent wheel locking.

Electronic control unit

Except in the case of hydromechanical anti-lock braking systems, a microprocessor-based electronic control unit is required to interpret the signals received from the wheel sensors and issue the necessary commands to the system modulator. In simple terms its purpose is to detect any significant slow-down of wheel speed relative to vehicle speed, which could result in the braking force at any wheel exceeding the available tyre to road adhesion force. The rapidity and accuracy with which the electronic control unit can function is such that the braking force can typically be modulated from three to twelve times a second, according to road surface conditions, thereby conferring smooth controlled braking. With anti-lock air brakes for heavy vehicles, the braking force modulation is limited to three or four times a second so as to minimize air consumption.

Apart from computing brake anti-lock requirements, another important function of the electronic control unit is that of monitoring not only its own state of health but also that of other related components, so as to ensure safe operation of the system. Should there be any malfunction it must therefore evaluate whether only part, or the whole, of the system must be switched out of action and then retain the nature of the fault in its memory for service investigation. A warning light informs the driver if a malfunction in the system has resulted in the electronic control unit being automatically switched off. In this event the system is failsafe, because normal braking control without anti-lock intervention remains available to the driver.

System modulator

This component represents the final link in the operating chain of an anti-lock braking system, since in all except hydromechanical versions it receives the commands from the electronic control unit and translates them, independently of driver action, into valve-based sequences that modulate either hydraulic or pneumatic pressure in the individual brake lines. As earlier indicated, the process of modulation can simply mean holding the line pressure constant or, if further correction is required, momentarily relieving and then restoring it. The modulator therefore requires its own source of supply pressure, which in electrohydraulic versions usually takes the form of an electrically driven pump capable of generating high pressure in the region of 14 to 17.5 MN/m^2 (2000 to 2500 lbf/in^2). This pump is sometimes arranged to operate in conjunction with a hydraulic accumulator for storage purposes.

An ABS in which hydraulic servo brakes (Section 27.8) and their modulator share a common hydraulic power supply is known as an ‘Integrated ABS’. This is distinct from a system with vacuum servo-assisted brakes and an independent hydraulic power supply for their modulator, which is sometimes referred to as an ‘Add-on ABS’ even though the units may be installed together.

With a hydromechanical anti-lock braking system, the hydraulic pump is mechanically driven from the same source that drives the flywheel wheel speed sensors. In the case of electropneumatic modulators for anti-lock air brakes on heavy vehicles, the continuous supply of compressed air made available by the compression and storage facilities of the braking system proper eliminates the need for an independent source of supply pressure.

It therefore follows that it is really the functional differences between the various types of modulator which determine the particular operating principle of an anti-lock braking system.

29.3 TYPES OF ANTI-LOCK BRAKE SYSTEM

The modulating principles of anti-lock hydraulic braking systems may be classified as follows:

Plunger return
Power recharge
Dynamic inflow

Plunger return

This principle of modulation was early developed to superimpose an anti-lock facility on a conventional hydraulic brake system, and interest in it has recently been revived. An essential feature of this type of system is the introduction of a cylinder and plunger modulator into each brake line between the master cylinder and the wheel cylinders, so that by moving the plunger up and down within its cylinder it becomes possible to vary the volume of a particular brake line and therefore modulate the line pressure and hence braking force generated at that wheel. Except under conditions of imminent wheel locking, the plunger is maintained at its lowest position in the cylinder by fluid at high pressure directed above it from an independently driven pump and storage accumulator, while a downward projecting pin from the plunger holds open an isolating valve that allows the free passage of fluid between the master cylinder and wheel

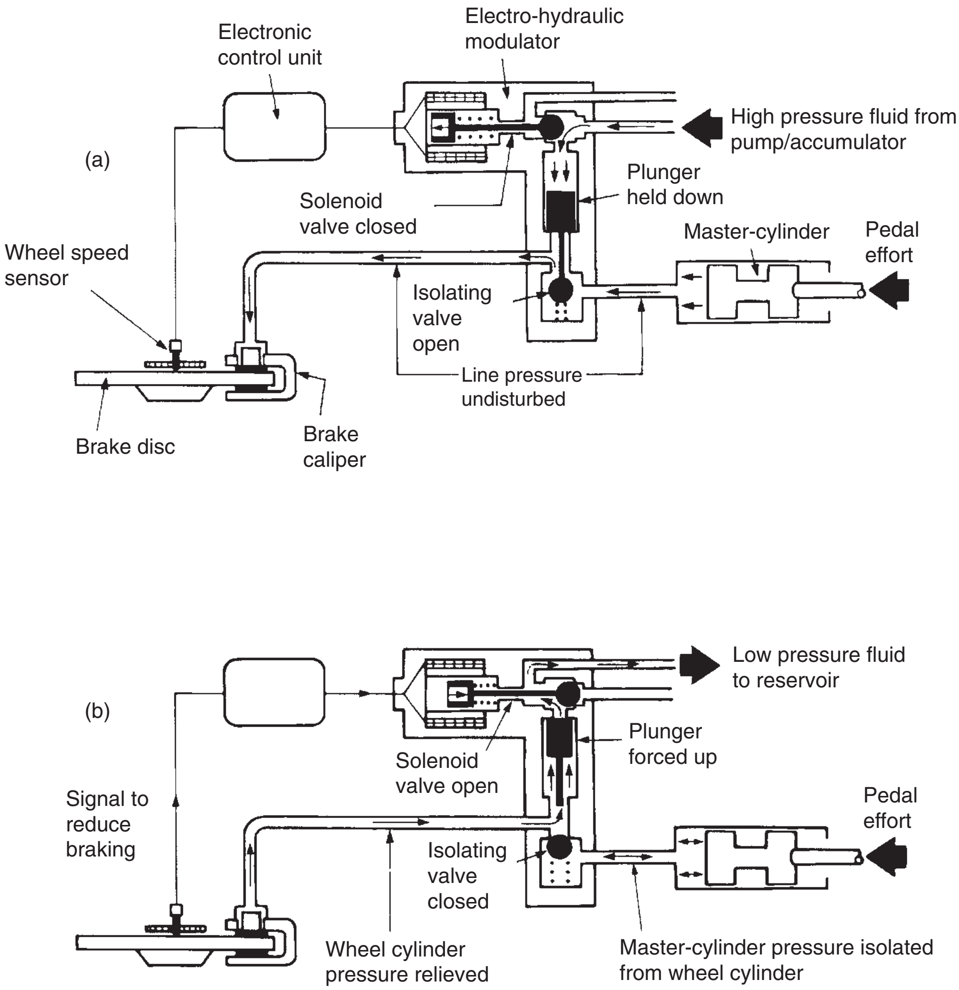


Figure 29.1 Plunger return principle of anti-lock brake modulation

cylinder (Figure 29.1a). Normal braking up to maximum pressure is therefore accomplished with the least volume of brake line, because there is always sufficient pressure acting above the plunger to restrain it from rising against master cylinder pressure acting below it.

However, if a wheel begins to lock the electronic control unit commands the appropriate solenoid valve in the module to shut off the fluid supplied under pressure to the plunger. As a result the plunger rises in its cylinder, the effect of which is twofold: first, it allows the isolating valve for the master cylinder to close, so that no further increase in wheel cylinder pressure is possible no matter how much effort the driver exerts on the pedal; and second, it increases the volume and therefore relieves the pressure in the line to the wheel cylinder, which momentarily releases the braked wheel (Figure 29.1b). Line pressure is of course restored to the wheel cylinder when the electronic control unit commands the solenoid valve to open, so that fluid at a higher pressure can force the plunger down its cylinder to reduce the volume of the brake line and re-establish a connection between the master cylinder and the wheel cylinder, via the now open isolating valve.

A modern application of the plunger return principle of modulation may be found in the Lucas Girling stop control system (SCS). Each modulator in this hydro-mechanical anti-lock brake system comprises an integrated wheel speed sensor, mechanically driven pump and brake pressure controller (Figure 29.2). Comparing this arrangement with that showing the general principles of a plunger return system of modulation (Figure 29.1), it will be recognized that what are termed the deboost piston, cut-off valve and dump valve correspond to the plunger, isolating and solenoid valves respectively.

The advantages generally claimed for the plunger return principle of modulation are that it avoids the need to open a conventional hydraulic brake system for the release of brake fluid, when line pressure is reduced for anti-lock control; the large open area of the isolating valve offers minimum restriction to the build-up of master cylinder pressure during normal braking; and the system is less costly than others because it does not involve the use of highly developed valve assemblies.

Power recharge

This has been the most widely used principle of modulation in anti-lock brake systems. Here again it is superimposed

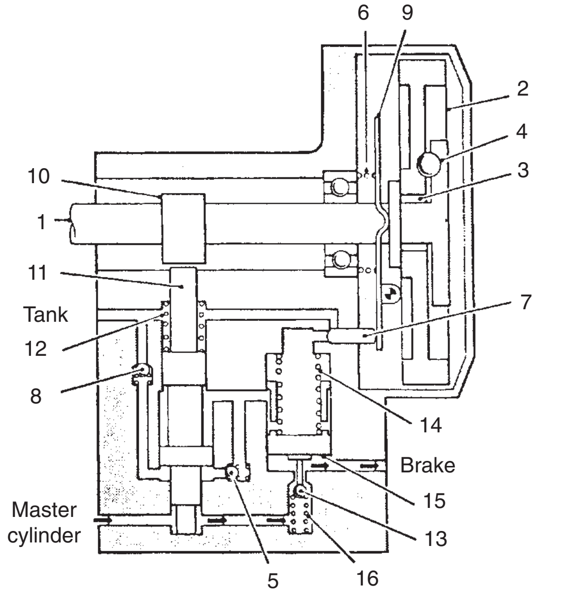


Figure 29.2 Lucas Girling SCS modulator unit (*Lucas Girling*)
1 drive shaft
2 flywheel
3 flywheel bearing
4 ball-and-ramp drive and clutch
5 pump outlet valve
6 flywheel spring
7 dump valve
8 pump inlet valve
9 dump valve lever
10 eccentric cam
11 pump piston
12 piston spring
13 cut-off valve
14 deboost piston spring
15 deboost piston
16 cut-off valve spring

upon a conventional hydraulic brake system, but with the essential difference that a brake line has to be opened in order to reduce pressure for anti-lock control, rather than simply increasing its volume as in plunger return modulation.

In principle a solenoid-operated multifunctional hydraulic shuttle valve serves to modulate the brake pressure between the master cylinder and either a single or a pair of wheel cylinders. During normal braking up to maximum pressure the spring-returned solenoid valve assembly is not energized, so that its upper valve remains open while the lower valve is held closed. In this position the valve assembly provides a free passage for the flow of brake fluid between the master cylinder and wheel cylinder (Figure 29.3a). If a wheel begins to lock during braking, the electronic control unit initially commands the modulator to energize the appropriate solenoid with one-half maximum current, which raises the valve assembly to an intermediate position with both its upper and lower valves held closed. This has the effect of not only isolating the wheel cylinder from any further rise in master cylinder pressure, no matter how much effort the driver exerts on the pedal, but also holding constant the wheel cylinder pressure (Figure 29.3b).

If this action of simply preventing any further rise in wheel cylinder pressure is not sufficient to correct the imminent locking of the wheel, then the electronic control unit commands the modulator to energize the solenoid with maximum current and therefore raise the valve assembly to its fullest extent. This results in the upper valve remaining closed to isolate master cylinder pressure, but the lower valve being opened to bleed off brake fluid to the return pump via the accumulator, thereby relieving wheel cylinder pressure to free the wheel (Figure 29.3c). At the same time the modulator is commanded to switch on the return pump, so that the fluid bled from the wheel cylinder line can be directed under pressure back to the master cylinder line and the inlet side of the solenoid valve assembly ready to recharge the system for the next cycle. This occurs when the modulator is commanded to switch off the pump and de-energize the solenoid, which allows the valve assembly to drop to the normal braking position. Each time a wheel cylinder line is depressurized, the recharging pressure in the master line cases a slight raising and pulsating at the brake pedal. It should nevertheless be held firmly applied, since any sympathetic pumping of the pedal by the driver will simply increase the time taken to stop.

The power recharge principle of modulation has long been incorporated in certain versions of the Bosch ABS and has been developed to provide a high level of vehicle control under difficult braking conditions commensurate with the higher cost of this type of system.

Dynamic inflow

This principle of modulation was an essential feature of a new concept in anti-lock brake systems introduced by the German company of Alfred Teves (ATE) in the mid 1980s, wherein brake operation, servo amplification and anti-lock modulation were combined in a fully integrated system. An additional low-cost version of this widely used system incorporates a further development of the dynamic inflow principle of modulation and serves to illustrate its application (Figure 29.4).

If the electronic control unit detects a locking tendency at any wheel, it commands modulation of the brake line pressure through solenoid-controlled inlet and outlet valves in the conventional manner. However, when it is necessary to reduce line pressure by opening an outlet valve, the excess volume of brake fluid is conveyed directly to the supply tank. At the same time an electrically driven pump is switched on and supplies fluid under pressure to both brake circuits served by the tandem master cylinder. This has the effect of slowly forcing the master cylinder pistons back to the point at which their central spring-loaded ball valves are unseated by axial stop-pins and limit the pressure (Figure 29.5). The significance of these central valves in the dynamic inflow principle of modulation is that they ensure the pressure generated by the pump remains proportional to pedal effort. They also allow the excess volume of brake fluid not required for modulation replacement to be returned to the supply tank, thereby avoiding the need for a hydraulic accumulator in the system.

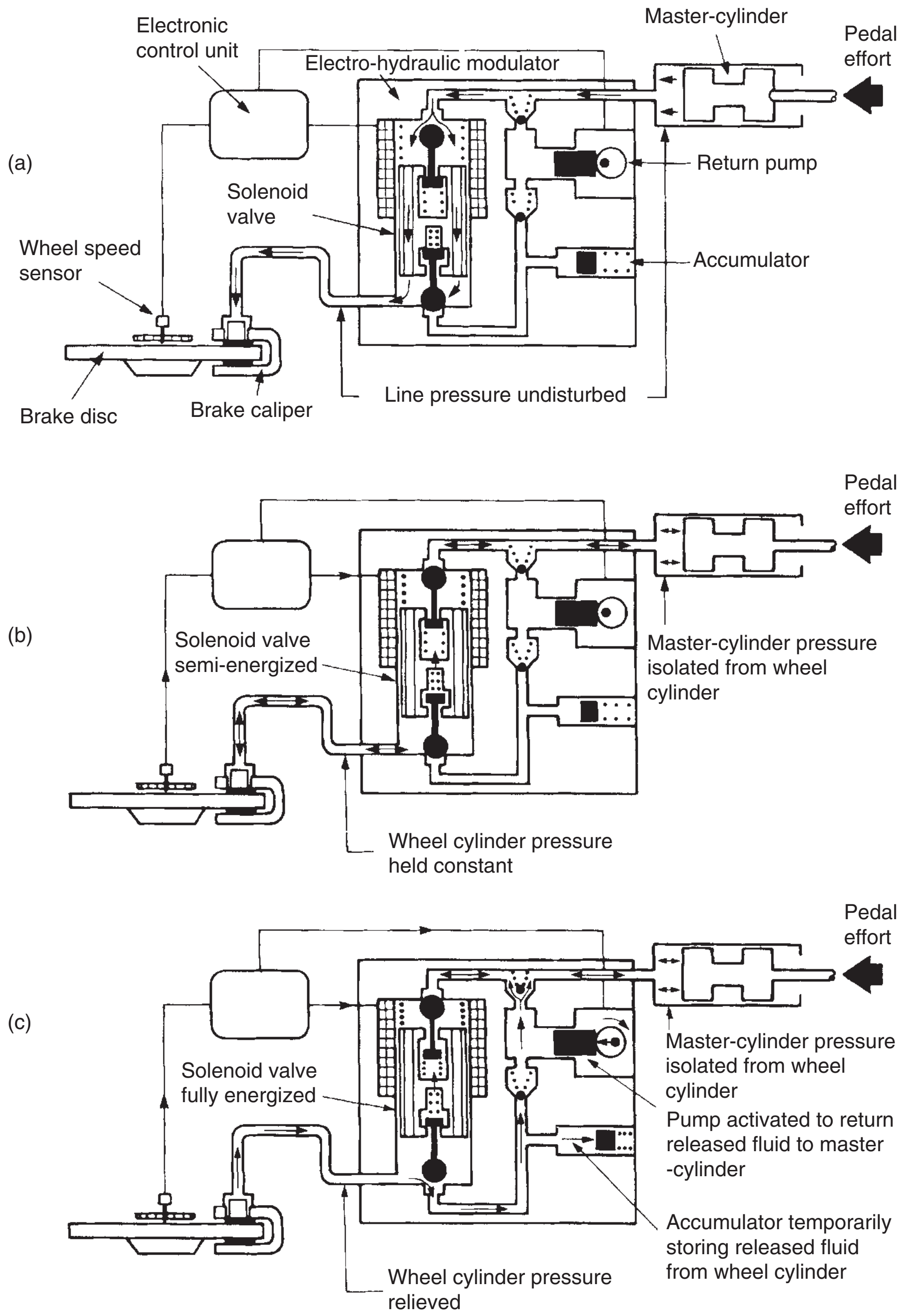


Figure 29.3 Pressure recharge principle of anti-lock brake modulation

29.4 OUTPUT CONTROL CHANNELS FOR ANTI-LOCK BRAKES

The extent to which each wheel is independently controlled with anti-lock brakes determines the number of output channels that need to be provided by the modulator system. Anti-lock brakes were originally introduced for rear-wheel-drive cars having a divided-line brake circuit with a front/rear split (Section 27.4). On these it was found that a four-channel anti-lock control, where each wheel had its own speed sensor and each wheel cylinder could have its apply pressure either held constant or relieved, was not necessarily the ideal arrangement. Although such a system behaved well in terms of stopping distance and steering response when braking on ordinary slippery surfaces, it could be at a disadvantage if the nature of the road surface offered more grip to the wheels on one side of the car than on the other, or what is sometimes grandly referred to as an asymmetrical split grip roadway.

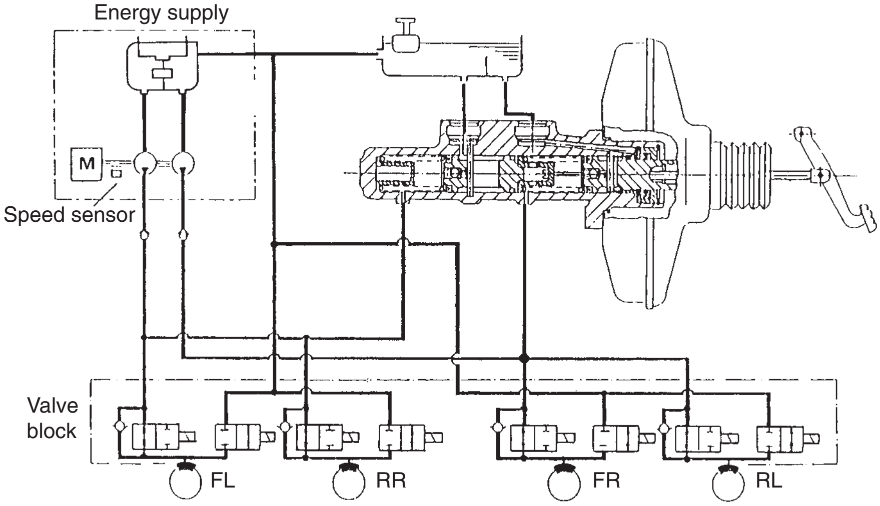


Figure 29.4 Anti-lock brakes with X-split circuit (*Alfred Teves*)

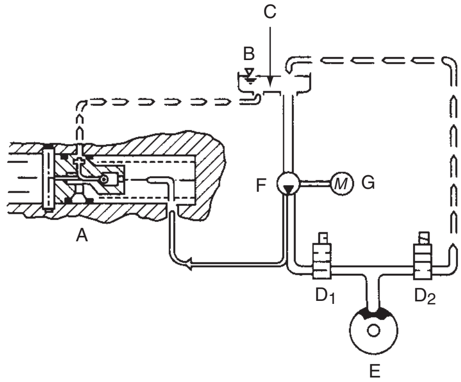


Figure 29.5 Dynamic inflow principle of anti-lock brake modulation (*Alfred Teves*)

A Master cylinder
B Fluid level sensor
C Reservoir
D_1 Solenoid valve (inlet)
D_2 Solenoid valve (outlet)
E Disc brake
F Non-return valve
G Electrically driven pump

This resulted in the side-to-side braking forces being unequal for the front and rear wheels, which could impose an unstable yawing movement or sideways turning on the car.

To avoid this condition three-channel control systems were developed, so that the front wheels could be speed sensed and controlled individually, while the rear wheels could be speed sensed but controlled as a pair by a floating balance or ‘select-low’ valve. This means that the maximum braking effort that can be applied to the wheel on the high-grip side of the car is limited to the maximum that can be applied to the other wheel on the low-grip side, so that neither wheel is allowed to lock and compromise directional stability.

For modern front-wheel-drive cars that conventionally have a divided-line brake circuit with an X-split (Section 27.4), a three-channel control with four sensors can similarly be used (Figure 29.4). Alternatively a less costly two-channel control system with fewer components may be employed. Such a system takes advantage of international legislation that governs the design of passenger vehicle braking systems, which decrees that under all conditions of loading the front wheels must ultimately lock before the rear ones to maintain directional stability. It is, of course, for this reason that the lightly laden rear wheels of front-wheel-drive cars are provided with brake apportioning valves to prevent them from locking (Section 27.4). From this it follows that if a two-channel two-sensor system imposes an anti-lock control on the front wheels only, then through their diagonal hydraulic connections to the rear wheels these too will benefit from a fair degree of control, since in any event they should be prevented from locking by their apportioning valves. A clearly identifiable example of a two-channel control system for front-wheel-drive cars with an X-split brake circuit is the earlier mentioned Lucas Girling SCS, which incorporates a separate anti-lock modulator with integrated speed sensor for each front wheel (Figure 29.6).

In the case of a four-wheel drive vehicle, the anti-lock brake system used on the earlier mentioned Mitsubishi Galant car (Section 21.4) features two diagonal output channels, together with speed sensors at all four wheels that operate in conjunction with a ‘G’ sensor, which monitors the overall amount of vehicle deceleration (Figure 29.7). A microprocessor-based electronic control unit receives signals from each sensor on independent input channels. It then uses this information to modulate the hydraulic line pressure for the brakes of the right front and left rear wheels on one output channel, and the left front and right rear wheels on the other channel, in the event of incipient wheel locking under braking. A select-low valve is also incorporated in the system, the function of which as earlier explained is to ensure that the maximum braking effort that can be applied to the wheel on the high-grip side of the car is limited to the maximum that can be utilized by the other wheel on the low-grip side. Another point of interest is that the viscous coupling

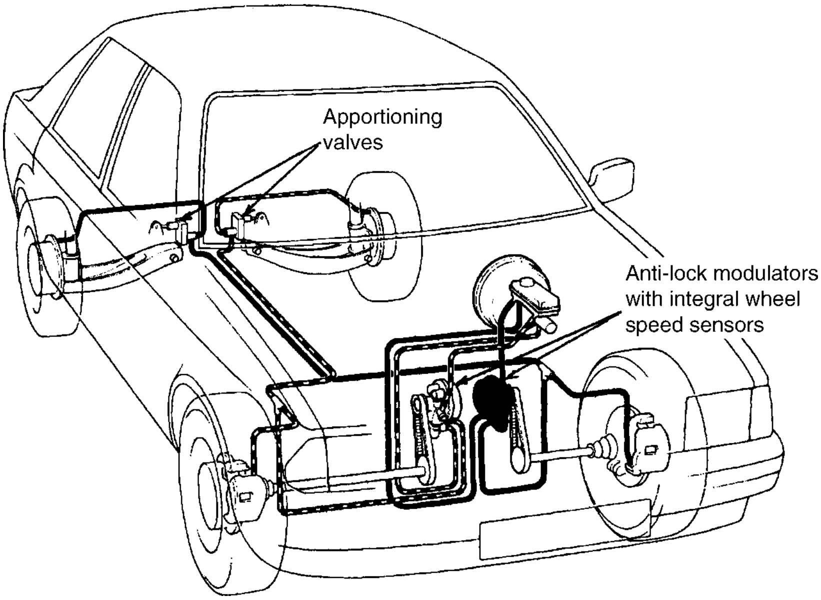


Figure 29.6 Lucas Girling SCS system with two-channel control (*Lucas Girling*)

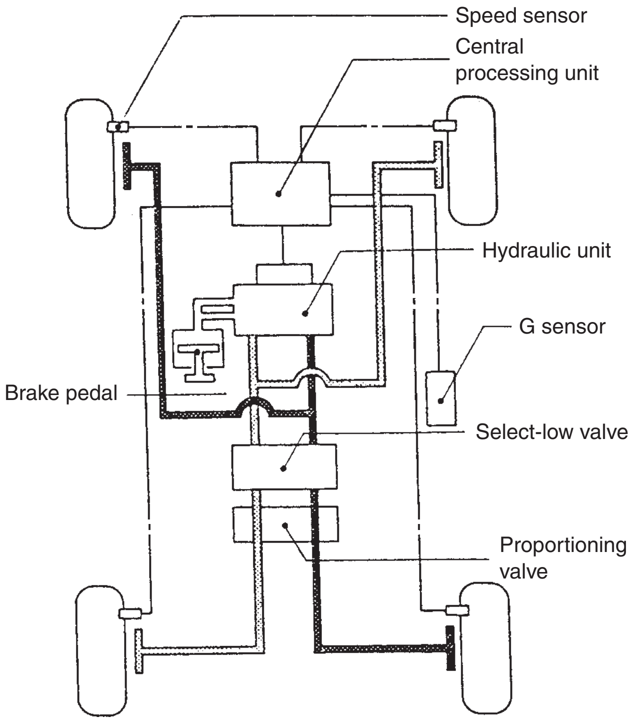


Figure 29.7 Four-wheel drive with anti-lock braking system (*Colt-Mitsubishi*)

unit (VCU) of this 4WD system (Section 21.4) also behaves as an anti-lock brake modulator. By virtue of acting as a limited slip coupling between the front and rear wheels, it automatically inhibits any one pair of wheels from locking up on a sudden application of the brakes.

29.5 ANTI-LOCK AIR BRAKES FOR HEAVY VEHICLES

There is clearly no less a requirement for anti-lock braking on heavy vehicles than there is for passenger cars and, in particular, the hazards of over-braking and skidding with articulated vehicles are widely acknowledged. In fact the gradual adoption of ABS for heavy vehicles, trailers and buses is becoming mandatory under EC braking legislation.

The principles of anti-lock braking as earlier described for hydraulic systems broadly apply to the air brakes of heavy vehicles, but there is a greater multiplicity of components and, as would be expected, basic differences in those that modulate the air pressure in the brake lines.

In a typical installation for a 6×4 vehicle the wheel speed sensing equipment is mounted within the drum brake assemblies and comprises six toothed rings, which act in conjunction with axially disposed electro-magnetic sensors. For anti-lock control three separate electronic control and modulator units are mounted from the vehicle chassis frame, their locations being adjacent to the front and tandem rear axles. Each modulator is provided with electropneumatic control valves and intervenes in an appropriate brake line between a system relay valve and a pair of axle brake actuators or chambers. The modulator control valves have three operational modes and can be commanded by the electronic control unit either to hold constant, relieve or restore air pressure directed via the relay valve from the foot-operated brake valve to the wheel brake actuators. Diaphragm controlled ‘hold’ and ‘exhaust’ valves are used in the modulator, their respective closing and opening being obtained by admitting system air pressure during braking into their operating chambers through solenoid valves, which are energized from the electronic control unit (Figure 29.8). Unlike an anti-lock hydraulic brake system, where it is necessary to conserve the fluid supply, the

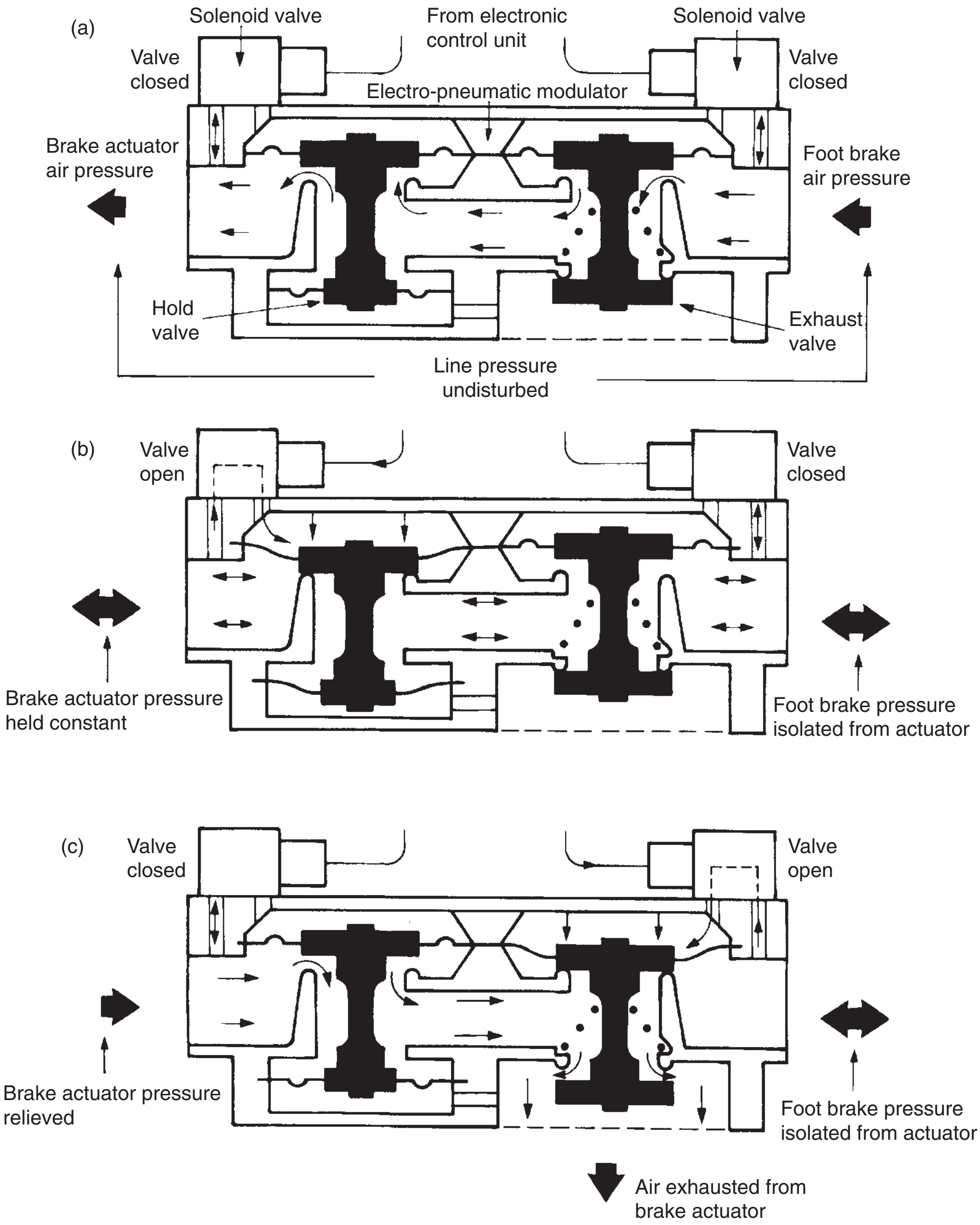


Figure 29.8 Modulation principle for anti-lock air brakes

air exhausted from the modulator is simply dumped to the atmosphere when line pressure is relieved to prevent wheel locking. Similar to an anti-lock hydraulic brake system, however, the electronic control unit possesses a comprehensive self-diagnostic capability and is failsafe in operation.

Anti-lock air brakes can also be installed on multi-axle trailers, a separate electronic control and modulator unit being used with wheel speed sensing being taken from the middle axle of a tri-axle trailer.

Anti-lock brake systems for both light and heavy vehicles are now generally accepted as being reliable in extended service. However, unless any malfunctioning of the system can be attributed to readily identifiable faults, such as defective electrical connections, broken cables, incorrectly gapped wheel speed sensors or damaged sensors; then resort has to be made to specialist diagnostic test equipment, which is used in conjunction with the comprehensive test schedules issued by the vehicle manufacturer.

29.6 TRACTION CONTROL SYSTEMS

General background

During moving off and acceleration, a traction control system performs a similar safety function to anti-lock braking by preventing the driving wheels from slipping, which therefore helps both to maintain directional control and to improve traction under adverse driving conditions, especially on road surfaces that are slippery on one side of the vehicle only. Since a traction control system may be regarded as a logical but inverse development of anti-lock brakes and can therefore utilize some of the same technology, it is perhaps to be expected that this type of system was pioneered by Robert Bosch working in conjunction with Mercedes-Benz, their anti-slip regulation system being introduced in 1987 and known as Antriebs-Schlupf-Regelung (ASR). Other manufacturers have since introduced traction control systems that may similarly be integrated with anti-lock hydraulic and now air brake systems.

Basic operating features of a traction control system

The two essential functions that must be performed by a traction control system are the automatic braking of a single driving wheel that is about to spin and the automatic throttling back of the engine if both driving wheels are about to spin.

Operating the wheel brake independently of the driver in this way not only is the most effective method of rapidly preventing a driving wheel from spinning, but also can arrest the momentum of a driving wheel whose grip suddenly changes to slip and begins to spin as the load on the power train is removed. It will be appreciated that, owing to the effect of the final drive differential, the application of a braking torque on a single driving wheel that is about to slip will result in an equivalent driving torque being applied to the other wheel that is gripping. Traction can therefore be maintained because the system is then acting in the manner of a differential lock. The final link in the system is that of imposing a drive torque control on the power train, so the available torque at the driving wheels is limited to that required for effective traction without slip.

Brake torque control of a traction control system is based on the existing components of an anti-lock braking system. It is therefore managed by an electronic control unit that combines the functions of both systems. The electrohydraulic modulator system used for passenger car anti-lock braking is extended to include additional valves for switching from braking to traction control modes, pressure modulation and pressure limitation. For smooth operation and, in the case of front-wheel-drive cars, minimum steering interference there is a slower build-up of hydraulic pressure for brake torque control as compared with that required for anti-lock braking. As the driving wheels must be braked without any action from the driver, the hydraulic energy is supplied from a high-pressure accumulator that is charged by an electrically driven pump. When signals received from the wheel speed sensors indicate that intervention by the traction control system is required, the electronic control unit commands the modulator switch-over valve to connect the hydraulic accumulator with the appropriate brake cylinder of the wheel beginning to spin. As a modulator valve is provided for each driving wheel, it is thus possible to brake either wheel independently of the other. The modulator valve holds the brake line pressure constant as soon as the wheel speed stops rising and then relieves it when the wheel slip has been brought under control.

Engine torque control is obtained by replacing the conventional mechanical linkage between the accelerator pedal and engine with an electronically based accelerator controller, through which the driver indirectly controls either the throttle valve of a petrol engine or the metering valve of a diesel engine. Based on signals received from the wheel speed sensors, the system therefore allows commands from the traction electronic unit to take precedence over driver action at the accelerator pedal, so that if necessary the engine can be automatically throttled back for traction control even though the driver may unwisely be urging greater acceleration. Such a system involves the sensing of accelerator position by a potentiometer, whose signals are then transmitted to the electronic control unit, which in turn commands an electrically driven actuator to operate the appropriate engine controls. A limited-movement mechanical connection can be retained between the accelerator pedal and engine controls, so that in the event of a fault developing in the electronic control system the vehicle can still be driven. The traction control system is, of course, switched off instantly when the driver applies the brake in the normal manner.

The general layout of the Teves integrated anti-lock braking and traction control system, as applied to a front-wheel-drive car, is shown in Figure 29.9.

29.7 VEHICLE DYNAMICS CONTROL

Whereas anti-lock brake and traction control systems are intended to function only when critical driving conditions are reached in the essentially straight-ahead direction, that is they prevent either a braked wheel from locking or a driven wheel from spinning, a vehicle dynamics control (VDC) or electronic stability programme (ESP) system extends this function to stabilize the vehicle when cornering, or in other words it influences the distribution of wheel lateral forces before critical conditions are reached. Such a system has been developed by Robert Bosch of Germany, again in conjunction with Mercedes-Benz, which constantly compares the direction steered by the driver with the actual direction of the vehicle. If there is a significant deviation from the former, the electronic control unit recognizes this and automatically intervenes to resolve the conflict by either actuating the brakes or adjusting the power from the engine, thereby redistributing the lateral forces acting at the wheels and stabilizing the vehicle.

Since determining the actual course of the vehicle is a relatively complicated business, a vehicle dynamics control system naturally requires components additional to those already associated with ABS/ASR systems (Figure 29.10). In particular a lateral accelerometer or G sensor is required that provides a sensitive response to the forces generated during cornering, and a yaw rate sensor that measures the speed at which the car rotates about its vertical axis. The latter sensor may be regarded as being at the heart of the VDC system, and is in fact a complex instrument that has been derived from aviation practice and adapted to the extreme environmental conditions of the motor vehicle. Two further sensors are also involved, these being a steering angle sensor that signals the intended course steered by the driver, and a braking pressure sensor.

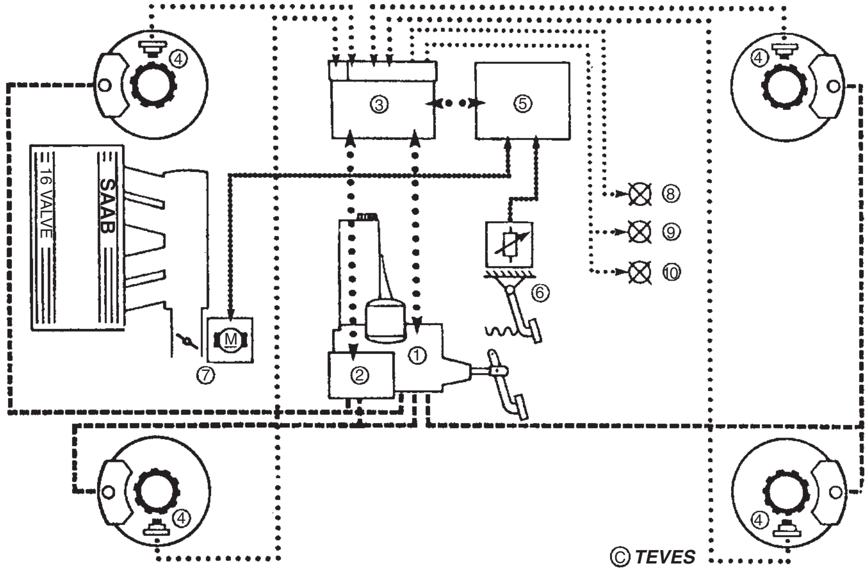


Figure 29.9 Layout of an advanced integrated anti-lock brake and traction control system (*Alfred Teves*)

1 Hydraulic unit with integrated ABS function
2 TCS valve block
3 Electronic ABS/TCS controller
4 ABS sensors (front axle ABS/TCS)
5 Electronic throttle controller } Electronic accelerator
6 Accelerator pedal sensor
7 Throttle actuator
8 ABS warning lamp
9 TCS warning lamp
10 TCS function tell-tale lamp

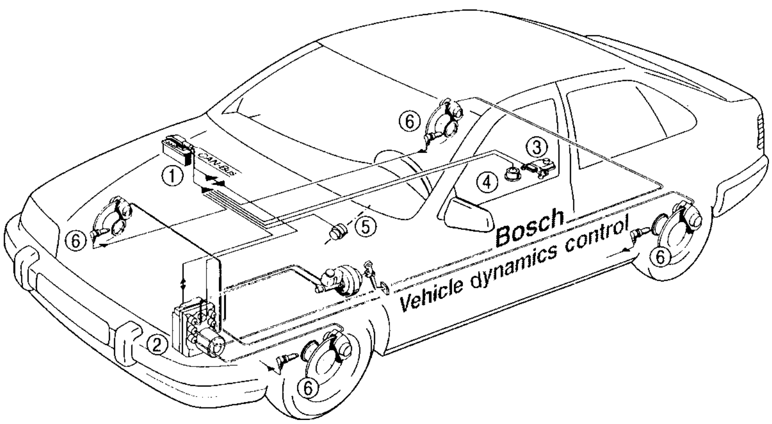


Figure 29.10 Layout of Bosch Vehicle Dynamics Control (VDC) system (‘*Photo:Bosch*’)

1 Electronic Control Unit (ABS/ASR/VDC) with CAN link to Vehicle Powertrain Control Module
2 Hydraulic Modulator with Pressure Sensor
3 Yaw Rate Sensor
4 Lateral Accelerometer
5 Steering Angle Sensor
6 Wheel Speed Sensors

For an understanding of how the VDC system responds to actual driving situations, it is first necessary to recall how the relative slip angles developed by the tyres as the vehicle is steered round a bend, determine whether or not it will understeer or oversteer (Figures 22.4 and 22.15). Another relevant factor is that the cornering ability of a tyre is reduced and therefore contributes to that end of the vehicle running wide, when either the tractive or braking forces generated between its tread and the road surface are increased. If therefore a car understeers and swerves off course when negotiating a bend, the VDC system compensates for the reduced cornering ability of the front wheels by automatically braking the inner rear wheel to impose a restoring yaw moment (Figure 29.11). Furthermore, the speed of the car is automatically reduced to

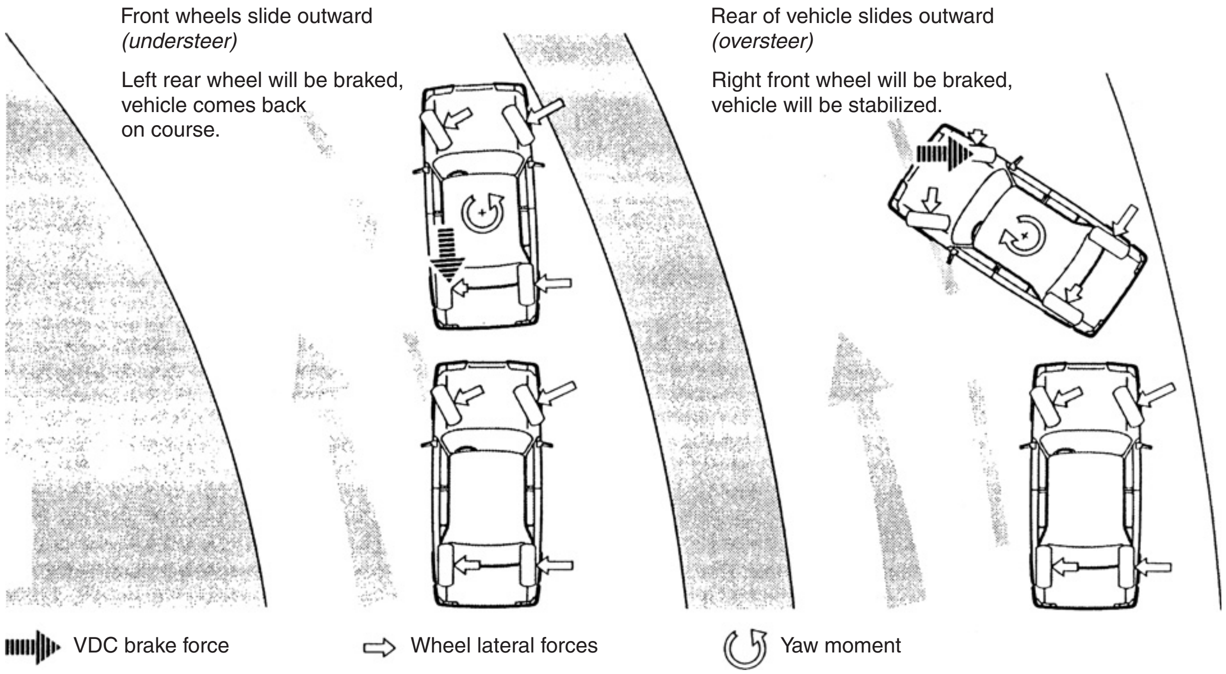


Figure 29.11 Action of Bosch Vehicle Dynamics Control (VDC) system ('*Photo:Bosch*')

an appropriate level, either by throttling the engine or by intervention of the braking system on selected wheels, or both. Conversely, if an oversteering condition is encountered, then the VDC system compensates for the reduced cornering ability of the rear wheels by automatically braking the outer front wheel to impose a restoring yaw moment, but this time in the opposite sense (Figure 29.11). The effectiveness of the VDC system in modifying the lateral stabilizing forces acting on a vehicle is not confined solely to cornering, but also to achieving optimum driving stability during rapid lane-change manoeuvres. However, it must be stressed that neither anti-lock brakes, traction control, nor vehicle dynamics control systems can defy the basic laws of physics and cater for what is sometimes referred to by development engineers as 'super-lunatic' driving behaviour!

30 Vehicle structure and aerodynamics

30.1 INTEGRAL BODY CONSTRUCTION

General background

The integration of frame and body functions into an all-embracing shell is variously known as integral, unitary, chassisless and originally by the French as monocoque construction. A significant step towards integral construction was the introduction of the all-steel welded body, a concept that was developed in America by the Budd Manufacturing Company for Dodge cars in the late 1920s. Although a separate chassis frame was still used, it no longer functioned as the total load-carrying member.

Integral construction proper was introduced by Citroën in France during the mid 1930s and was evolved in collaboration with the previously mentioned Budd Company. It was based upon the principle of stressing the outer skin of the body shell to perform a load-carrying function, which contributed to the strength and rigidity of the inner structural members, thereby eliminating the need for a separate chassis frame. Perhaps it may be of interest to mention that the monocoque method of providing a more efficient structure was also well known to aircraft engineers and featured in such historic World War II aircraft as Blenheims and Spitfires.

Finally we come to modern practice, where the principle just described of pure integral construction is often modified in various ways, notably in combining it with separate subframes for mounting various parts of the vehicle mechanism. The aim here is chiefly to suppress noise, vibration and harshness from the body interior by the extensive use of rubber isolators and also to simplify the assembly of the mechanical units to the body structure. This particular form of integral or semi-integral construction was pioneered by Daimler-Benz of Germany in the early 1950s. Figure 30.1 illustrates the main parts of a modern integral construction body.

The basic structure

In conventional practice the passenger car body is produced from overlapping sheet metal, fastened by a multiplicity of spot welds often robotically applied. For example, the steel panels that form the body structure of the modern Citroën C5 car are united by no less than 4200 spot welds. The body framework is constructed by joining together thin-walled closed-section members, and where these are subject to particularly large torsional and bending stresses, such as in the case of the body sills, they may be reinforced by the inclusion of longitudinal partitions (Figure 30.4). Similarly the frame joints may require to be locally reinforced by additional flanged members, since much depends on their structural integrity to maintain stiffness of the body and hence preserve correct alignment of the various units it supports. A further requirement is to avoid resonant vibration of the body structure, which demands a high degree of torsional stiffness or resistance to twisting. It generally follows that a body structure that has the required degree of torsional stiffness also possesses acceptable beam stiffness or resistance to bending. The body is subject to beam loading that arises from its own weight and that of its occupants, together with the weight of the engine and transmission units. It is therefore stressed in compression along its upper body and in tension along its underbody. In modern practice this has led to the selective use of high-strength steel panels in the upper part of the body and ultra high-strength panels in its lower part. As compared to standard steels, these specialized alloy steels confer greater strength so that the thickness of the panels can be reduced to save weight. Furthermore, production methods can now be such that any given panel may be varied in thickness, thereby matching the particular stress levels imposed upon it. That is, an increased wall thickness can be concentrated in areas subject to greatest loading.

The individual members comprising what is termed the ‘body less doors’ are most conveniently identified in relation to the following assemblies: underframe, side frames, and roof frame. With regard to the all-important underframe assembly, which forms the entire lower portion of the body structure, this is now more often referred to as the ‘platform’. Its importance in terms of the considerable research and development costs involved is such that since the mid-1990s platform sharing between different models of high volume manufacturers has become increasingly common to spread these costs. A notable pioneer in this respect was Fiat Auto, who introduced a common floorpan for their Fiat 127 and 128 models in 1970.

Underframe assembly

This may be regarded as the backbone of the modern integral construction body, since it forms the lower portion of the car and provides the major part of its strength (Figure 30.2). Its middle section comprises a floor ribbed for stiffness and flanked on each side by the inner sill members, and usually it has a central tunnel which houses either the propeller shaft, remote gear control linkage or exhaust system, according to the particular layout of the car. The floor panel terminates at its front end in the toe board and dash panel subassembly and at its rear end in the heel board and rear seat pan subassembly. This typically incorporates the boot floor in one large pressing. Box-section cross-members are provided beneath the front seat and at the leading edge of the rear seat pan. A pair of box-section longitudinal members or longerons, running beneath and either partly or wholly the length of the floor panel inboard of the sills, are extended forwards and rearwards to the body extremities. If the

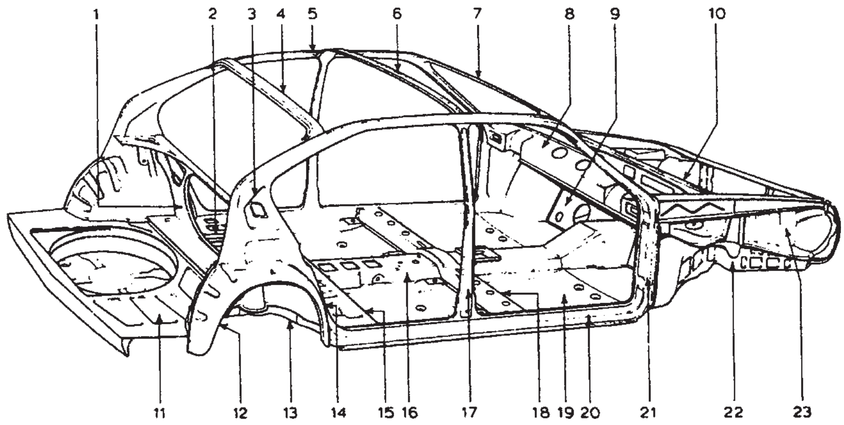


Figure 30.1 Main parts of a modern integral construction body (*Alfa-Romeo*)

1 rear seat squab bulkhead
2 rear seat pan
3 quarter panel
4 rear window (backlight) reinforcement panel
5 cantrails
6 roof bow member
7 windscreen header panel
8 dash panel
9 toe board
10 service compartment bulkhead
11 boot floor
12 wheel arch
13 longerons
14 D post
15 heel board
16 cenral tunnel
17 B-C post
18 cross-member
19 floor panel
20 sills
21 A post
22 longerons
23 wing valances

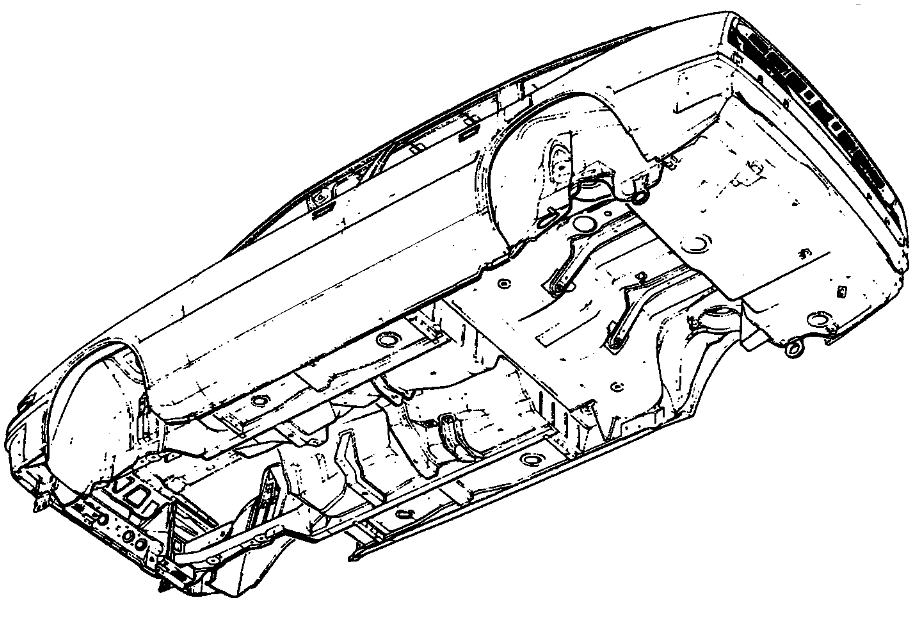


Figure 30.2 Underside view of integral construction body for front engine rear-wheel drive car (*Pressed Steel Fisher*)

longerons are interrupted along their length, the front pair extend inwards as far as the cross-member beneath the front seats, but the rear pair extend no further inwards than the heel board cross-member. Full-length longerons are used for sports utility vehicles (SUV) where integral body construction is employed. The underframe assembly is completed by the front wing valances and wheel arches, which are united with the dash panel and front pair of longerons.

Side frame assemblies

Each of these comprises an outer sill member, door posts, quarter panel including rear wing, and roof cantrails. With a four-door body the front, middle and rear door posts are designated A, B-C and D posts respectively. In all cases they can pose complex joining conditions with their surrounding structure. Their upper ends are united by what is termed a cantrail member, which is also of box-section construction. A side frame is completed by the rear wing valance and wheel arch, this assembly then being integrated with the rear seat pan and boot floor. Each front wing is attached separately both to its underframe valance and wheel arch and side frame door A post.

Roof frame assembly

This serves to close the essentially box-like structure of the conventional car. It comprises, of course, the roof panel flanked on each side by the cantrails mentioned previously and may be cross-braced by roof bow members. The roof panel joins the windshield header panel at its front end and the reinforcement panel for either a rear window or tailgate at its rear end.

Subframes

A subframe is a rigid, detachable assembly that is connected to the lower body structure through rubber isolation mounts. At the front end of the body it supports an independent suspension system, steering mechanism and the power unit. Acknowledging the latter function it is sometimes referred to as an 'engine cradle'. Since the suspension system and power unit are themselves connected to the subframe through isolation bushings and mounts, it will be evident that there is a double filtering effect against the transmission of vibration and noise to the car body interior. Another requirement to be met by the front subframe in modern practice is that it must safely redistribute the forces of impact in the event of a head-on collision. A four-point mounting is a typical arrangement for the front subframe. At the rear end of the body a subframe supports an independent suspension system, either wholly or partly according to its layout, and the final drive unit. Similar to the application of a front subframe, the suspension system and final drive unit are likewise connected to the rear subframe through isolation bushings and mounts. It necessarily follows that front and rear subframes, which are pre-assembled with the units they support, readily lend themselves to high volume production methods.

Front and rear subframes were traditionally fabricated from several steel stampings welded together and locally reinforced. Since Ford of America introduced a 'hydroformed' front subframe in the mid-1990s, this method of manufacture is now widely used as indeed it is for various other automotive components. Hydroforming or tubular hydroforming is basically a process in which a tube is inserted in the closed cavity of a die, which is formed to correspond to the shape of the finished component. The tube is then partially sealed and filled under high pressure with a hydraulic fluid, so that the increasing pressure forces the tube to assume the internal shape of the die. Manufacturing advantages of hydroforming include more economic assembly arising from fewer parts, the absence of welded flanges and hence a saving in weight, and from a design point of view the ability to accommodate changes in cross-section area. An alternative to the steel subframe is one fabricated from aluminium alloy, using for example extruded sections as introduced in 1999 by General Motors for their Chevrolet Impala, or hydroformed tubing that has recently featured in German practice. An aluminium alloy subframe can be advantageous in terms of high rigidity and low weight.

Advantages of integral body construction

As compared with the once conventional separate frame and body for car build, the following advantages are usually attributed to integral body construction in its various forms:

Stiffer structure for the same weight
Improved passenger accommodation
Safer in a collision
More economical to manufacture.

Stiffer structure for the same weight

This derives mainly from the elimination of the body-to-frame flexible mountings and provides the opportunity for increasing all-round car performance, without incurring a weight penalty. In particular the body is less prone to shake, which helps to improve both ride and handling qualities. The greater stiffness also contributes to improved durability of the structure, because of its better resistance to the weakening effects of vibration. From this it follows that there should be less tendency for body squeaks and rattles to develop. Another important consideration is that the greater stiffness or rigidity of the structure minimizes distortion at the body apertures.

Improved passenger accommodation

The desire to achieve this has always been one of the main reasons for a manufacturer adopting integral construction, and it is gained in two ways. First, the structure is inherently better adapted to the established modern styling requirement for a wide body and hence contributes to greater seating space. Second, it enables the space between minimum ground clearance and the roof to be more efficiently utilized, thereby improving headroom despite a low roof line. This is because the absence of chassis frame side members allows a lower floor relative to the body sills, while the simple cross-members that lend stability to the body floor may be buried under the seats.

Safer in a collision

For some years now increasing attention has been directed by road safety research organizations and manufacturers towards what is called the passive safety of a car (Figure 30.3a). This term refers to the better protection against injury for the car occupants once an accident has occurred. In fact international legislation is now in force which, among many other passive safety requirements, seeks effectively to limit the amount of deformation that the passenger compartment of a car may suffer in relatively low-speed accident situations including front, side and rear impacts.

From our knowledge of basic physics we may recall that the second of Sir Isaac Newton's three laws of motion tells us that when a force acts upon a body, it produces an acceleration which is proportional to the magnitude of the force and inversely proportional to the mass of the body, or in perhaps more familiar terms:

$$\text{force} = \text{mass} \times \text{acceleration (or deceleration)}$$

From this it should be evident that it is the rapidity with which the vehicle is stopped that influences the force imposed upon the occupants. Therefore, the very sudden stopping of a vehicle in a crash can accelerate the occupants, with respect to the vehicle, such the force of impact may either injure or kill them. In simulated crash situations human volunteers (suitably restrained by energy-absorbing harness) have withstood peak acceleration forces of just over $20\,g$ or, stated another way, just over 20 times the gravitational force acting on their own body. On the other hand, it is generally recognized that it would be difficult to sustain life with peak accelerations of $60\,g$ or more.

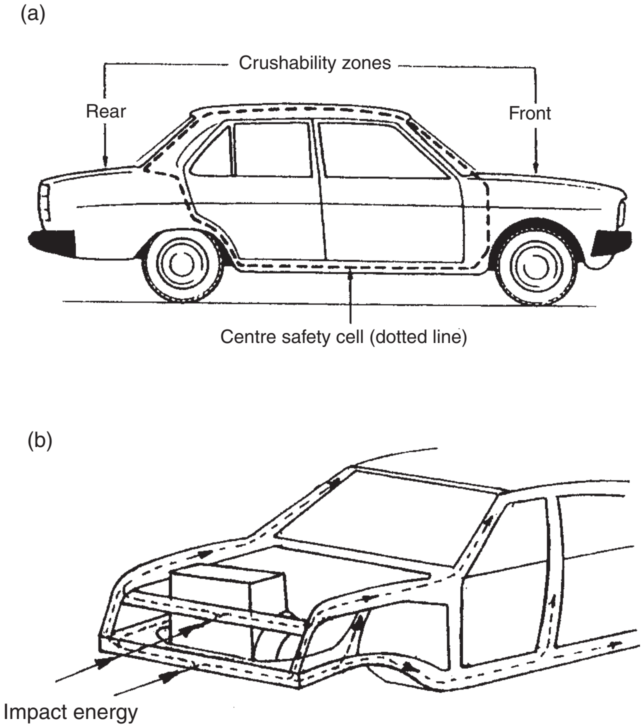


Figure 30.3 Passive safety of integral body construction: (a) crushability zones (b) frontal impact energy dispersion

One method of minimizing the peak deceleration of a crashing vehicle and therefore the acceleration of its occupants is to absorb the energy of impact over a fractionally longer period, by allowing the extremities of the body structure to collapse in a progressive manner. In general terms, this can mean that if a vehicle travelling at a speed in the range of 25 to 45 mile/h (40 to 72 km/h) collides head on with a rigid barrier, then provided that the front end structure of the body can become crushed to a distance of about 0.6 m (2 ft) the peak acceleration suffered by the occupants should not be greater than 40 g.

The greater protection afforded by an integral construction body is achieved mainly by designing the structure so that, unhampered by a separate rigid chassis frame, the front and rear portions of the body may collapse progressively under impact and preserve more nearly intact the middle passenger compartment. Front impact forces are therefore distributed partly through the lower main longerons into the stiffened door sills and floor pan, and partly through the upper rails of the engine compartment into the stiffened A posts and roof panel (Figure 30.3b). Rear impact forces are similarly distributed via the boot compartment into the body structure. A transverse engine installation, especially with in-line cylinders, and a spacious boot compartment both contribute to ample front and rear crush zones. In other words, the front and rear portions of the body are deliberately made more crushable than the occupied middle cell. This particular feature of safety body design, involving controlled crushability zones, was pioneered by Daimler-Benz in the early 1950s. Later research by this company further showed that the most severe and also most common type of frontal impact occurs when two approaching vehicles collide with an overlap, rather than fully head-on. Since this implies that the forces of impact must be contained mainly by one side of the body structure, it has led to the use of forked longitudinal members, which better distribute the forces of impact into the appropriate load-bearing sections of the rigid passenger compartment and provide increased protection for the footwell (Figure 30.3b).

Other safety features that may be directly applied to the passenger compartment or ‘safety cell’ as it is sometimes termed, include steel stiffening beams that span the roof cantrails, which in conjunction with reinforced door posts provide better protection in roll-over accidents; a steel stiffening beam that spans the A posts at scuttle height to lend additional support for the steering column, which not only reduces steering-wheel incursion in the event of frontal impact, but also allows effective air bag deployment; and steel stiffening beams installed across the interior of each door, these being termed ‘side intrusion bars’ and are attached at their ends to reinforced hinges and catches, so that side impacts can be more readily absorbed by the central body structure (Figure 30.4). To ensure a better controlled collapse upon impact for more recent lightened front-end body structures, the forward portions of the longerons and upper rails may be provided with a series of either grooves across their faces or notches along their edges (Figures 30.4 and 30.5). These V-shaped indentations are known as ‘fold initiators’ and serve to stabilize the axial collapse of the members by encouraging them to deform in a convoluted manner. In low-speed frontal impacts the yielding of these members therefore protects against more serious distortion of the central body structure and has the incidental advantage of reducing accident damage repair costs.

Another established safety feature, which complements the controlled crushability of the modern car, is the collapsible steering column to prevent the driver becoming impaled in a severe accident. To limit the rearward displacement of the steering column into the passenger compartment, some designs of energy-absorbing column incorporate either a convoluted metal section that folds like a Japanese lantern, or plastic pins that shear, when the column is subjected to a telescopic impact load. Other designs of anti-penetration column have deliberately misaligned and universally jointed sections that afford a similar protection for the driver (Figure 23.37).

More economical to manufacture

Although the initial tooling costs of integral body construction can reach staggering proportions, the long-term and necessarily high-volume manufacturing costs are reduced for several reasons. These include the more efficient use of materials (especially the saving that results from the elimination of a separate chassis frame and its associated body mountings), the reduction in the number of parts handled, and the automation of production build techniques.

A more recent design concept for the integral construction body is the pre-assembled modular front end, which can then be assembled to the main body structure as a single unit and therefore reduce assembly time and hence production costs. This system favours an ‘open’ front end construction, where

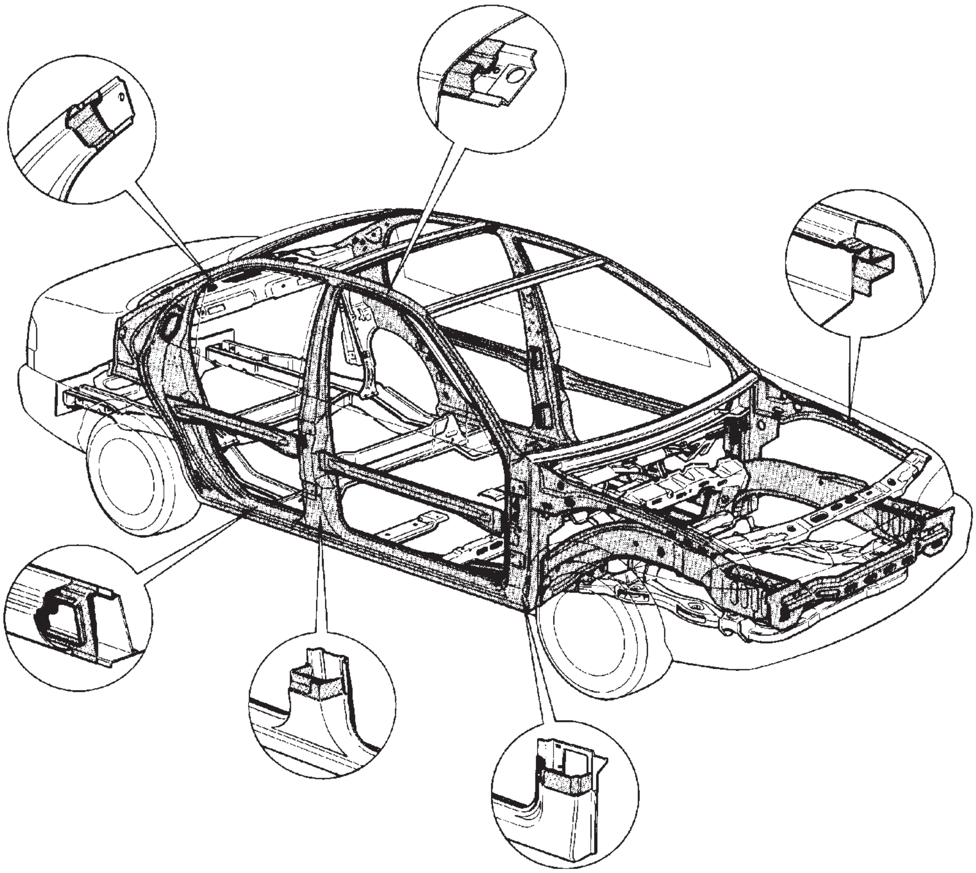


Figure 30.4 Integral construction body sections for stiffness and safety (*Ford*)

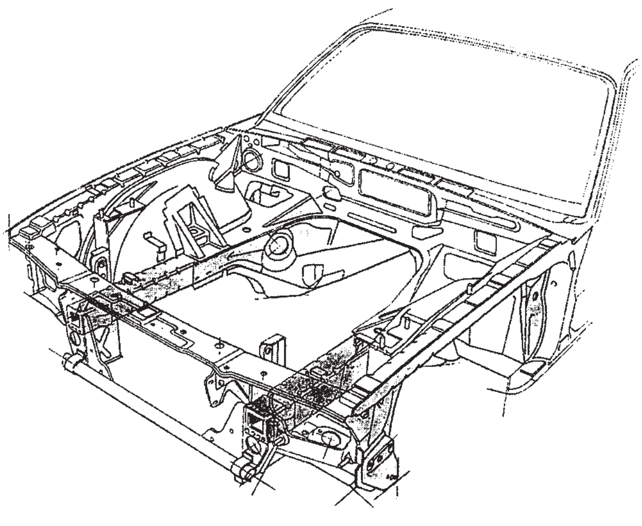


Figure 30.5 Energy absorbing front-end body structure (*Jaguar*)

a transverse impact beam is attached to the lower rails of the engine compartment after the body has been assembled, rather than a 'closed' front end which already has the transverse member welded to it during body assembly.

Less favourable considerations

These are usually concerned with the following aspects of vehicle operation: increase in noise transmission, and earlier deterioration from corrosion.

Increase in noise transmission

The integral construction body tends to be more affected by panel drumming and road rumble noises, as compared with the separate frame and body. Therefore, resort was made early to such devices as the swaging of floor and other body panels, so as to stiffen them and raise their natural frequency of vibration; the application of sound-deadening treatments on panels; and the liberal use of sound-absorbing materials in the offending areas of the body. As mentioned earlier, the semi-integral type of body construction better lends itself to the suppression of noise, vibration and harshness from the body interior.

Earlier deterioration from corrosion

The reason for this is that any corrosive attack on the body panels tends to have a greater weakening effect on the structure as a whole, than would be the case if a separate chassis frame of relatively thicker material was used to support the body. An integral construction body with its panels pressed from sheet metal in thicknesses typically ranging from 1.2 mm (0.048 in) for mild steel down to 0.7 mm (0.028 in) for modern high-strength but also ductile steels, the latter being first used on Japanese cars in the interests of weight saving, is therefore by its nature more prone to deterioration from corrosion. Protection from corrosion involves, in the first instance, the careful design of structural members to avoid wherever possible joints exposed to the entry of road splash and to ensure that those which are exposed are suitably sealed against it. Second, the body structure as a whole must be subjected to selective and thoroughly applied anti-corrosion chemical treatments. In some body constructions the steel pressings are galvanized in areas especially prone to corrosion, while the pillars may, for example, be wax flooded and injected with silicone foam to repel moisture. Another feature protecting against corrosion is the use of anti-chipping screens, which for the Citroën C5 car are used on the underside of the body, sump and rear bumper.

Stiffness requirements of integral construction

An integral construction body must possess considerable stiffness or rigidity for several reasons. The greater its rigidity the higher will be the natural frequency of vibration and hence the lower the amplitude of this vibration. If the former should be too low and unhappily correspond to the frequency of excitation caused by vibration of the unsprung mass of the wheels and suspension, then resonant vibration and body shake can occur. Clearly the body must also be rigid enough to preserve correct alignment of the mounting points either for the front and rear suspension systems direct, or for their intermediate subframes. Similarly, the body shell must be able to resist deflection at its apertures for the doors, so that metal-to-metal clearances and optimum sealing conditions are maintained. The build quality of the modern car is often perceived by what are termed its 'shutlines'. That is, the consistency and size of gap existing between its doors and their surrounding body panels, a gap now in the region of 4 mm (0.15 in) compares to one of about twice this amount acceptable in earlier body build. Figure 30.4 illustrates various sections contributing to stiffness in an integral construction body.

In resisting the various deflections, the integral construction body is subject to a combination of loads that may be summarized as being mainly the following:

Vertical bending loads imposed by the gravitational force acting through the centre of gravity of the car and reacted against at the mounting points for the front and rear suspension systems.

Horizontal bending loads resulting from cornering inertia force acting through the centre of gravity of the car and again reacted against at the mounting points for the front and rear suspension systems.

Torsional loads created by the twisting effect on the body of a single wheel climbing an obstacle and reacted against across the mounting points for the front and rear suspension systems. The maximum torsional load condition would occur if the car were supported on only three of its four wheels.

Stiffening effects of body members

Scuttle

This term has long been used to identify that portion of a car connecting the bonnet and the passenger compartment of the body. Such a clear line of demarcation is now hardly discernible, because the curved and steeply raked windscreen used with modern styling has a lower edge that extends almost to the dash panel. It is, therefore, the latter which really defines the scuttle portion of the modern car. Structurally, the scuttle portion contributes to the torsional stiffness by acting as a forward cross-tie between the underframe and the side frames. A similar role is performed towards the rear of the body by any transverse panel or rear bulkhead that may be used behind the rear seat squab and which spans the rear wing valances.

Underbody assembly

From the description given earlier of the underframe it should be fairly evident that the longitudinal sills, central tunnel and cross-members promote considerable resistance to bending over the length and width respectively of the floor area. For this reason these strengthening members of the body structure are produced from thicker steel plate than that used for the body panels, typically in the region of 2–2.5 mm (0.078–0.097 in). A tunnel bracing member may also be added to the floorpan, to provide increased resistance to side impact collisions. The stiffening effect of the suspension supports, as related to the underbody as a whole, is mentioned later.

Roof bow assembly

In early designs of integral construction the roof bow assembly was generally thicker in section and of greater curvature than is found in modern practice, so that it now contributes rather less to the bending and torsional stiffness of the body, although as mentioned earlier stiffening beams are now increasingly used in the roof structure for safety considerations (Figure 30.4).

Suspension supports

These generally serve to extend the resistance to bending over the entire length of the underbody assembly. Since the longeron members which they normally comprise are integrated with the floor panel and the inswept front and rear wing valances, the supports also contribute to the torsional stiffness of the body.

Bulkhead brace

The term 'bulkhead' originally referred to the vertical flat partition of the scuttle assembly that divided the engine and passenger compartments. In modern practice an often less obvious bulkhead is formed by the complex-shaped pressing that comprises the dash panel. A bulk-head brace serves as a rigid triangulating member, in both side and plan views, between the front longerons and the dash panel where it joins the A posts of the side frames. It may comprise a box-section strut that is disposed diagonally across the sloping inner face of each wing valance, or the bracing effect may be implicit in a sturdy box-like construction for the combination of front wing and valance.

Cantrails
Since the modern styled car has much larger windows or, in the language of the body engineer, greater glass areas, there is less space available for structural members in the upper part of the body than was formerly the case. The cantrails (or roof rails) must therefore be of carefully contrived box sections so that they provide the necessary resistance to bending at the joints between the door pillars and the roof. This bending effect is imposed when, in plan view, the roof tries to rotate relative to the floor as the body is loaded in torsion. That is, the cantrails would attempt to move fore and aft relative to the sills below.

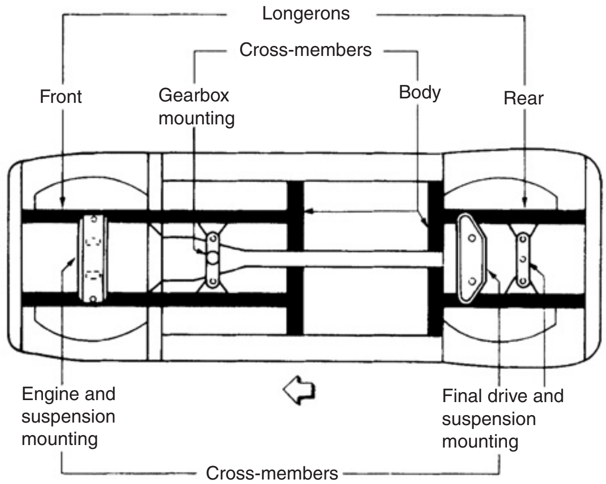


Figure 30.6 Mountings for mechanical units as viewed from underside of integral construction body

Supporting functions of the body

The attachment of the mechanical units to an integral construction body demands an approach different from that adopted for a separate frame and body, because of the relative difference in metal thickness of the load-bearing members. More specifically, it is necessary to spread the load to be supported over a greater area of the thinner sheet metal used with integral construction. (Figure 30.6). Otherwise problems could arise in service from localized distortion of the sheet metal sections, which at least could introduce misalignment and at worst result in failure of a particular mounting system.

Typical examples of the manner in which the abovementioned technique is applied are as follows.

Engine and gearbox assembly
With an engine and gearbox unit mounted fore-and-aft, the front engine mounts are conveniently carried by either a simple cross-member, or a more complex subframe, which is primarily used to mount the front suspension mechanism to the body structure (Figures 30.11a and c). In the former case, a shallow partially boxed cross-member can be rigidly attached towards the front ends of the longerons (Figure 30.7a).

Alternatively, a subframe may be flexibly attached to the longerons at more widely spaced locations along them (Figure 30.4). In the latter event, the front extremities of the longerons are integrated with a cross-beam to provide the necessary rigid foundation for mounting the sub-frame. The rear gearbox mount is carried by a short cross-member bolted to the underside of the transmission tunnel, this being locally reinforced for the purpose. Less commonly, the engine and gearbox may be carried in their entirety by a flexibly mounted open subframe that extends over the whole length and breadth of the front longerons.

It has long since become established practice to employ a three-point mounting system for the engine together with gearbox. This superseded the earlier four-point mounting system not only to relieve the engine unit of all strain due to deflections of the chassis frame then used, but also to absorb torque fluctuations more effectively by virtue of greater mounting flexibility. The torque fluctuations induced by the power impulses from each cylinder are a major source of engine vibration, their effect being most pronounced at idling speeds. They tend to rock the engine unit in opposition to crankshaft rotation, and thus establish the natural axis of oscillation or roll axis of the engine. Hence, it is about this axis that the support mounts must permit the greatest degree of freedom for engine movements. In practice, the roll axis slopes down towards the gearbox end of the engine unit, since the centre of gravity of the engine is higher than that of the gearbox. The actual arrangement of the support mounts may be contrived to produce either a ‘centre of gravity’ or a ‘centre of percussion’ system of suspension.

In a typical centre of gravity system, a pair of sandwich mounts supports either the front or, less commonly, the rear of the engine unit. They are arranged in V-formation so that, under the static load of the engine, their projected normals meet on the roll axis (Figure 30.8a). The third mount is of either the cylindrical or the sandwich type, which is arranged to intercept the roll axis at the other end of the engine unit (Figure 30.8b). Hence, the engine is supported about its roll axis which passes through the centres of gravity of both engine and gearbox, so that sideways shake of the engine as it rocks on its mounts is minimized. Furthermore, the main supporting pair of mounts resists the rocking motion of the engine as pure shear loading on their rubbers, thereby providing the most effective isolation of vibrations. To assist in controlling the fore-and-aft movements of the engine unit, either rubber buffer limit stops or a rubber-bushed stay rod may be required, especially where the support mounts are all of the single sandwich type.

With a centre of percussion mounting system, the engine is again supported about its roll axis, but the transverse mounting planes are so disposed that no interaction from disturbing forces occurs between the front and rear mounts. The system is based upon the physical principle that if a

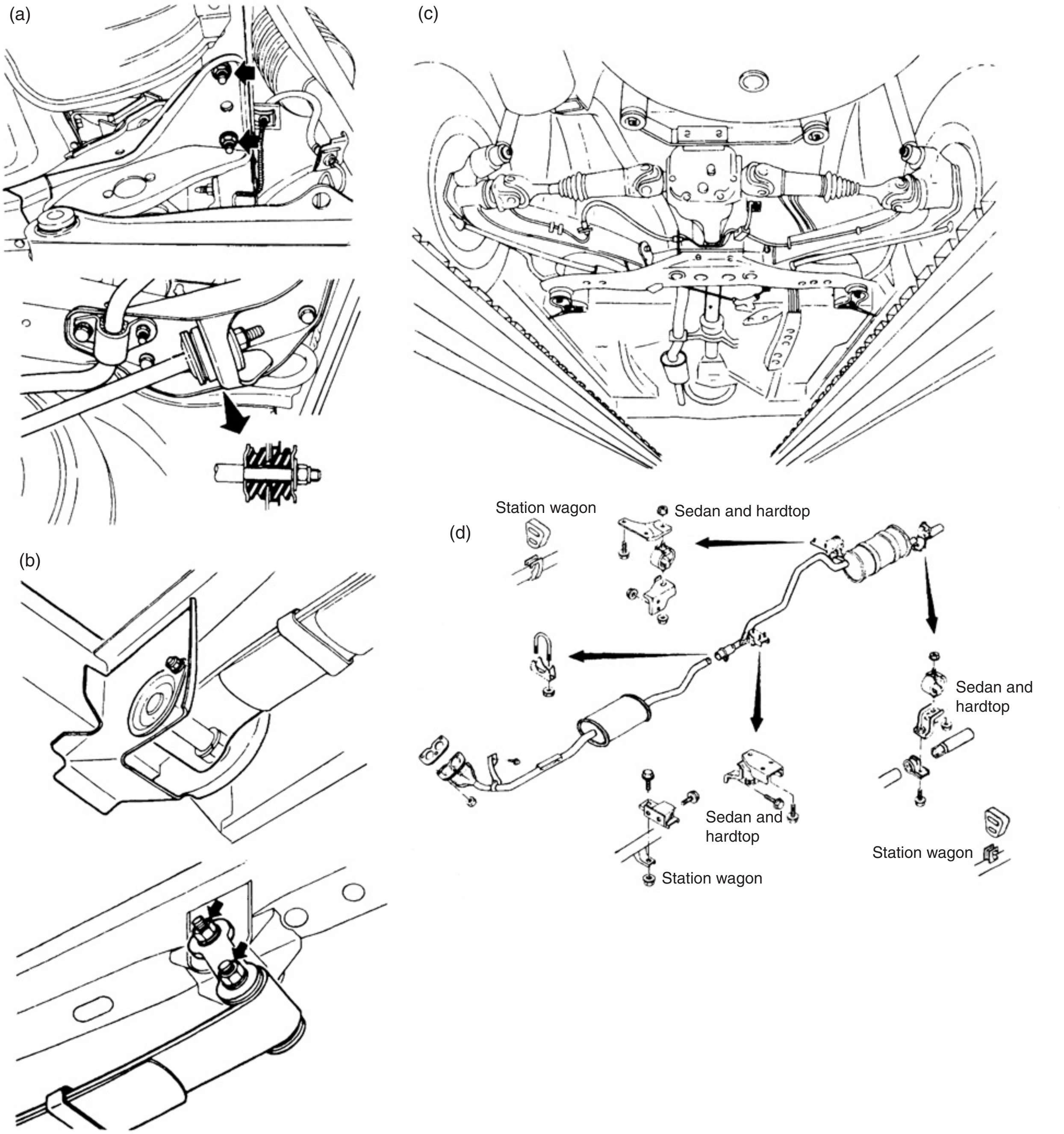


Figure 30.7 Unit mounting methods for integral construction body: (a) cross-member and mounting brackets for engine and front suspension (b) mounting brackets for rear suspension semi-elliptic leaf springs (c) cross-member and mounting brackets for IRS (d) mounting brackets for exhaust system (*Nissan*)

pivoted body is dealt a blow at its centre of percussion, no reaction is produced at the pivot, whereas if the blow is struck at any other point of the body a disturbing force is reacted at the pivot. In other words, the pivot 'doesn't know' when the body is struck at the centre of percussion. As applied to an engine mounting system, the rear mount is thus located at the centre of percussion for the disturbing forces acting upon the front mounts, and vice versa. This system of mounting is especially applicable to in-line four-cylinder engines, where the vertically acting unbalanced secondary forces may be entirely accommodated by the front support mounts. For this purpose, the latter are located on a horizontal lateral axis that intersects the roll axis in the plane of the unbalanced forces; that is, midway between the cylinders (Figure 30.9). The rear mount is then located at a suitable distance from the centre of percussion for these disturbing forces and is thus not affected by them.

In the case of a transversely mounted engine and gearbox where the support mounts must be able to accommodate the full torque applied to the road wheels, the three mounting points are oriented about the unit as follows. One mount is positioned beneath the unit at the juncture of the bell-housing and engine block and is supported by a detachable third longeron member running across the bottom of the engine compartment to a front cross-beam. A second mount is positioned at cylinder head level to support what would normally be the front of the engine, a pick-up point for this mounting being provided on the reinforced valance structure. The third mount takes the form of either a high-mounted stay rod or, as in later practice, a low-mounted torque arm. To control rocking movements of the engine and gearbox unit, the rubber-bushed ends of the stay rod connect to the rear face of the cylinder head and the upper dash panel structure. The alternative torque arm is rigidly attached at its front end to the bell-housing periphery and at its rear end enters a rubber bushing supported from the lower dash panel structure (Figure 30.10a). With both arrangements the mounting support areas are locally reinforced on the body. For improved vibration control during engine idling, another method of mounting a transverse engine and gearbox is to employ a four-point system. Here the right- and left-hand mounts are placed on the roll axis of the unit and the other two front and rear mounts are inclined to intercept this axis (Figure 30.10b).

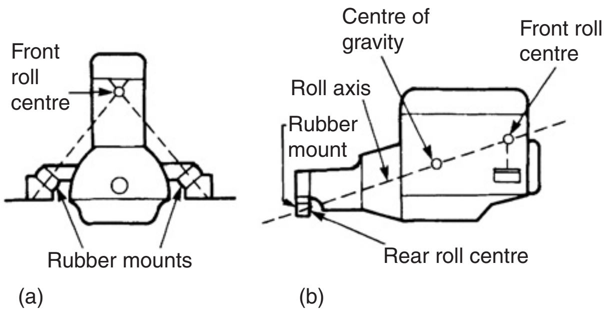


Figure 30.8 Centre of gravity engine mounting system

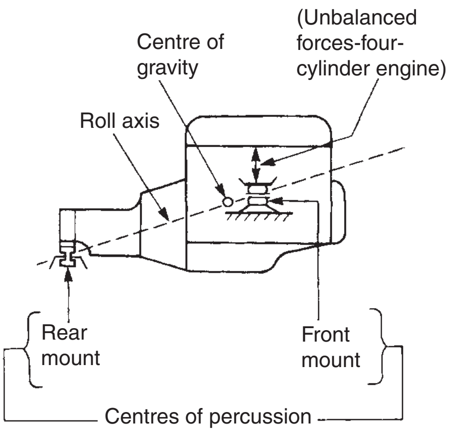


Figure 30.9 Centre of percussion engine mounting system

Final drive

If the rear wheels are the driven pair and are also independently sprung, it is necessary to provide mounting points for the final drive unit on the car body structure. In conventional practice, however, the final drive unit is supported either wholly, or partly, by the flexibly mounted subframe from which the suspension arms are pivoted (Figure 30.7c). The subframe is attached beneath the forward ends of the rear longerons. When an additional mounting is required for the rear end of the final drive unit, it is usually supported from a stiffening cross-beam integrated with the boot floor and the longerons.

Front suspension

As mentioned previously in connection with engine mounting, the front suspension mechanism is supported from the body structure either directly by means of a rigidly attached and simple cross-member, or indirectly through the medium of a flexibly mounted subframe.

We must now qualify this statement by observing that modern IFS systems of the MacPherson strut and link type require additional supporting features. In particular, a pair of forward anchorage brackets are required where diagonal links are used

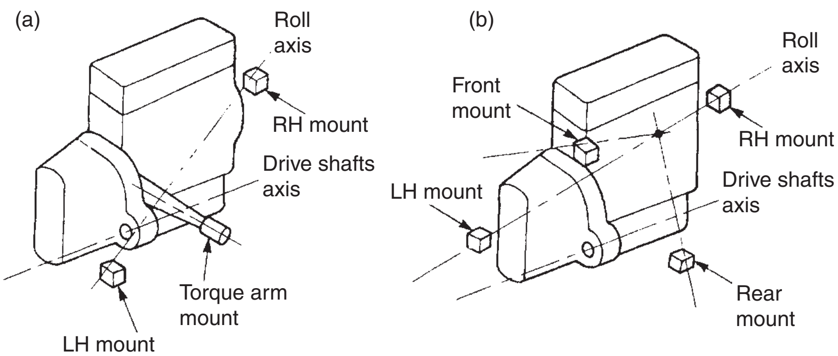


Figure 30.10 Transverse engine mounting systems

to transmit fore-and-aft forces from the track control arms to the body structure (Figure 30.7a), as described in Section 23.2. With this type of system an upper flexible mounting must also be provided for each strut, and, since the suspension spring load is reacted against at the same point, a very rigid and suitably braced MacPherson tower is an essential feature of the wing valance construction (Figure 23.13). Further to increase rigidity and improve car handling by reducing deflections at the strut anchorages, a tower-to-tower cross-brace may be additionally used. The widely spaced attachment points for this type of suspension system do, nevertheless, make a valuable contribution to reducing load concentration on the body front end structure, as indeed was the original design intention.

Rear suspension

In conventional practice the suspension mounting for driven rear wheels, which are independently sprung either by the semi-trailing arm, or by the increasingly popular multilink systems, is effected through the media of a subframe flexibly mounted from the rear longerons (Figures 30.7c and 30.11b). An upper location for each suspension coil spring and shock damper is provided against the upswept portion of the longerons that may be increased in width to accommodate them, or alternatively by the equivalent of MacPherson towers integrated with the rear body structure.

In contrast, the increasingly popular method of suspending non-driven rear wheels by means of a modified strut and link system requires a substantial box-section cross-member from which the transverse links can pivot and against which each suspension coil spring can react if it is separately mounted from the strut (Figure 23.34). Otherwise, MacPherson towers are required in the rear body structure. The cross-member is rigidly attached towards the rear of the body sills, which in this case are extended under the boot floor to furnish a wide base mounting for the rear suspension system (Figure 23.34).

For live axle rear suspension systems the pivot points for flexibly mounting either semi-elliptic leaf springs (Figure 30.7b) or positive locating links are provided by external brackets and internal sleeves integrated with the rear longerons.

Exhaust system

The installation of the flexibly mounted exhaust system to the underframe of an integral construction body varies in detail according to the geographical layout of the system (Figure 30.7d). For a directly routed single system, the front and the rear silencer with tailpipe are supported from mounts that are typically located from rigid body sections towards the centre of the rear seat pan cross-member and the far end of the appropriate rear longeron. With a longer-wheelbase car an additional forward support for the system, taken say from the cross-member under the rear gearbox mount, is also likely to be employed. If any form of dual exhaust system is used then there must, of course, be some duplication of the mounting arrangements. Owing to the high operating temperatures of catalytic converters, their adjacent rubber mountings are silicone based.

Anti-vibration mountings

By the very nature of its rotating, reciprocating and oscillating mechanisms, the running of a car is attended by a certain amount of vibration. Even though such vibration is minimized at source in modern automotive practice, its transmission to the car body interior either simply as vibration or in the form of noise audible to the occupants, or both, is always undesirable. Indeed, its suppression is the declared mission in life of the noise, vibration and harshness (NVH) test engineer, a specialist activity first identified by the Ford Motor Company in the early 1970s when increasing attention began to be given to this problem.

The ideal, but albeit totally unrealistic, method of confining vibration to any particular mechanical unit would be to suspend it in space, so that it could vibrate without causing a disturbance in the car. In reality, the next best thing is to support the unit on flexible anti-vibration mountings in which rubber is generally employed as the spring medium. By using a flexible, as opposed to a rigid, mounting system the vibratory forces of a mechanical unit are reduced to the relatively small spring forces transmitted by the support mountings themselves.

At this point, we should perhaps distinguish between vibration isolation and insulation. For example, the engine rubber mounts not only isolate the car structure from outgoing vibrations of the engine, but also insulate the engine against incoming vibrations from the car structure, so that the overall effect is for vibration of the engine unit to be minimized under all running conditions.

An advantage of using rubber, rather than steel, as the spring medium for anti-vibration mountings is that the sound transmission through them is reduced. Basically this can be attributed to the absence of a metal-to-metal path for the sound to travel along, since rubber in itself is not particularly sound-absorbent owing to its natural frequency being small compared with that of sound. Another advantage is that the internal friction of rubber tends to dampen vibration. However, it is worth reflecting that no matter how much a car may cost, its acceptable functioning depends very much on the few blocks of rubber used to support its mechanical units. The various applications of anti-vibration rubber mountings can be readily identified from the illustrations in Figure 30.11, showing front and rear suspensions systems together with their subframes.

A more recent trend has been towards the use of ‘hydro-elastic’ engine rubber mountings with built-in hydraulic damping, this idea having been first developed by R.E. Rasmussen of General Motors in the early 1960s. In principle, an otherwise conventional conical rubber mountings is internally partitioned to accommodate upper and lower fluid-filled chambers. As the mounting responds to engine movements, the fluid is forced in either direction through an orifice in the partition, thereby exerting a controlled damping force on the spring action of the mounting (Figure 30.12a). This enables a better compromise of high damping at low frequencies and vice versa to be attained, which is especially beneficial in mounting the transverse engines of front-wheel-drive cars.

Since the late 1990s a further refinement in the design of hydraulically damped rubber mountings has been to provide

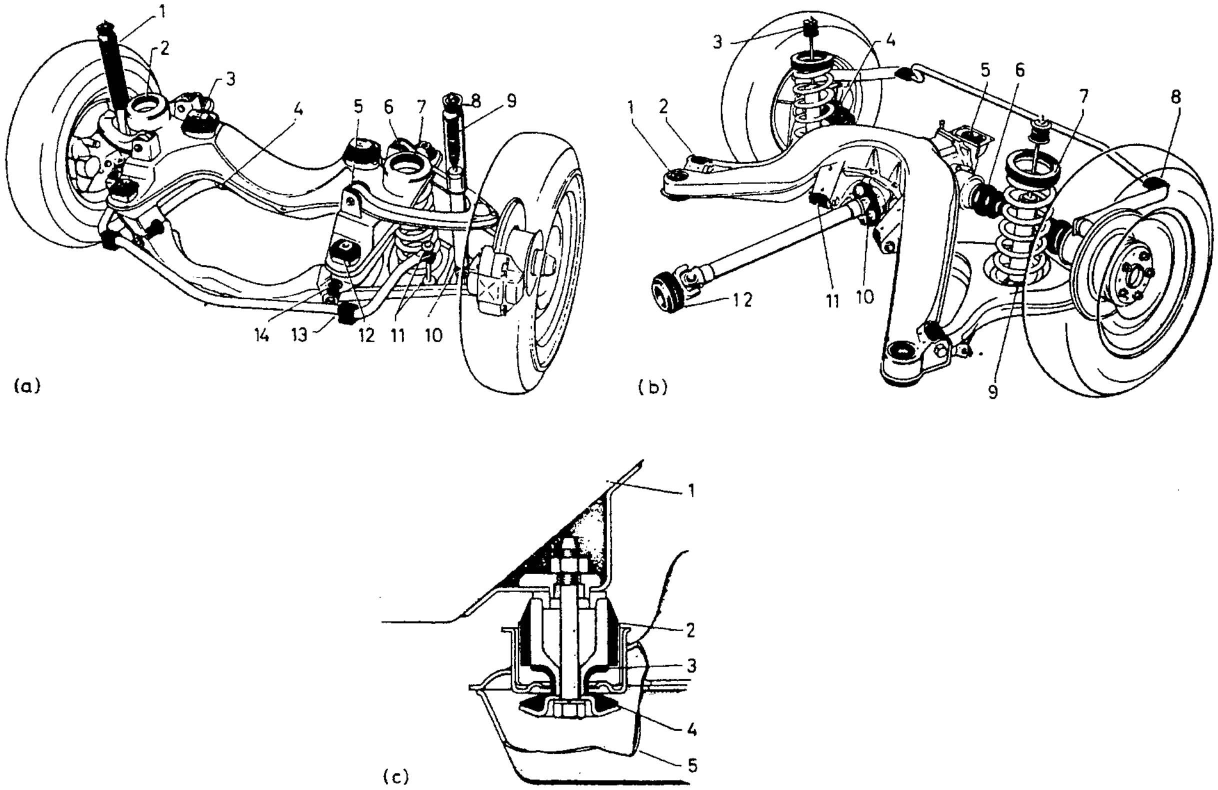


Figure 30.11 Reducing noise, vibration and harshness by the use of rubber parts on axles (*Mercedes-Benz*)

(a) *Front axle*
1 protecting sleeve
2 ring
3 engine mounting
4 bush of the lower lateral wheel fork arm
5/6 bush of the upper transverse wheel fork arm
7 front axle carrier mounting
8 shock damper rings
9 spring washers
10 shock damper mounting
11 pad
12 front axle carrier mounting
13 torsion bar bearing
14 bush of the lower lateral wheel fork arm

(b) *Rear axle*
1 rear axle carrier mounting
2 diagonal wheel fork arm bush
3 shock damper pad
4/6 half-axle gaiters
5 rear axle centre mounting
7 coil spring mounting
8 torsion bar bearing bush
9 shock damper mounting
10 propeller shaft coupling plate
11 diagonal wheel fork arm bush
12 propeller shaft bearing

(c) *Front axle carrier detail*
1 body
2 mounting
3 flange plate
4 bumper pad
5 front axle carrier

them with an adaptive control system. In basic principle this involves the use of a slide valve that is actuated by a linear solenoid, so that an additional orifice between the upper and lower chambers of the mount can either remain covered or be uncovered (Figure 30.12b). Clearly this action will influence the extent of fluid exchange between the two chambers, and hence the degree of damping imposed upon movements of the engine. The solenoid actuator receives its electrical signals via an electronic control system, which recognizes fluctuations in crankshaft rotation as an indication of engine vibration and modifies the damping accordingly.

All the thin metal panels of an integral construction body likely to vibrate or transmit noise are covered with anti-vibration materials of one type or another. They generally take the form of sound-deadening treatments, such as the application under heat of bituminized material direct on to the panels; and sound-absorbing materials in pad form, such as resin-bonded layers of natural and artificial fibres that are cemented to the panels (Figure 30.13). Also, the various box-section stiffening members of the body may be filled with foam products. In all cases the principle is the same: the greater the difference in sound resistance, as defined by the product of sound velocity and material density, between the materials in contact, the more effective is the sound damping.

Checking integral construction body alignment

To verify the accuracy of repair work carried out after accident damage to integral construction car bodies, various

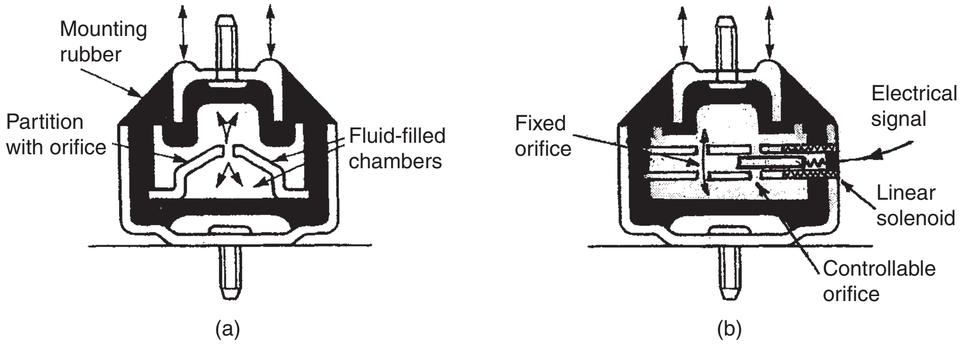


Figure 30.12 Schematic arrangements of hydraulically damped rubber mountings: (a) fixed orifice (b) fixed and controllable orifices

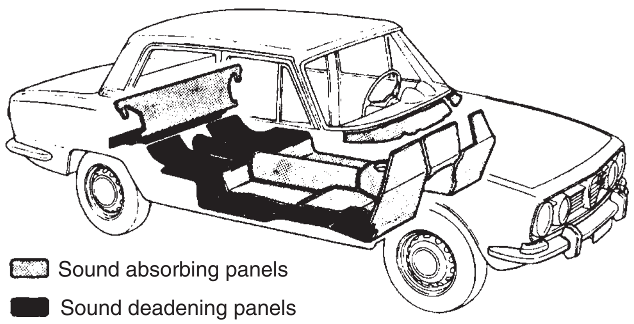


Figure 30.13 Sound-proofing of integral construction body (*Alfa-Romeo*)

forms of alignment checking benches or base panel gauges may be used, which establish the correct location of the mounting points for the suspension and steering assemblies. In the example of body alignment checking bench illustrated (Figure 30.14), all sliding dowels should match freely the respective holes in the body or, at the most, after a slight pressure is applied by hand. The diameter of each dowel is significantly less than the bore of the corresponding hole in the jig, this clearance directly relating to the maximum tolerance allowed for the positioning of the hole in the body.

Body alignment checking equipment must be treated with care, since any loss of accuracy resulting from damage can be reflected in car handling and tyre wear problems.

30.2 ALUMINIUM BODY CONSTRUCTION

General background

The designers of early motor cars soon began to recognize that merely building a more powerful engine was not necessarily the most effective way to improve car performance, if it was to be constrained by excessive weight of the bodywork. As early as 1903 we find for example the pioneer French car manufacturer, Panhard et Levassor, offering an optional lighter weight all-aluminium body albeit at an extra cost to the customer. For better quality cars therefore it increasingly became the practice to cover the wooden framework of their bodies with either pressed or hand-beaten, sheet aluminium panelling. However, this type of lighter weight body construction suffered a decline with the gradual demise of the separate steel chassis frame on which it was built and the widespread adoption of the all-steel integral body construction. There have nevertheless been sporadic attempts to make much more extensive use of aluminium alloy for the structural elements of the car, but with rather less commercial than technical success. This situation may now change with the introduction by Audi in 1994 of the 'Audi Space Frame' (ASF) form of construction for their luxury high-performance A8 model, which is intended to be comparable in terms of both weight and fuel economy to a mid-range saloon of conventional steel construction.

Audi space frame construction

This type of space frame construction should not be confused with that sometimes employed for specialist sports cars, where a web-like structure of tubular steel members is panelled in aluminium, since it more closely resembles a conventional integral body construction. It nevertheless differs from the latter in that instead of utilizing pressed-steel assemblies, an aluminium-silicon alloy is variously deployed in the forms of extruded sections, cast components and sheet panelling. Extruded box-section, straight and curved members comprise the basic structure of the body frame, because their use avoids any loss of rigidity associated with spot-welded seams and they can be produced with variable wall thickness according to the duty they perform, thereby allowing the optimum use of material. Vacuum cast components are employed to confer the required rigidity at the intersecting welded joints for the extruded members, since this method of die-casting in which the die is evacuated before injection of the molten aluminium contributes to a high-strength casting. The sheet panelling for the outer skin of the body is formed from thin-gauge aluminium alloy stampings, these being either riveted or bonded to the frame members.

To enhance passive safety of the ASF body structure in frontal impacts, the longerons beneath the passenger compartment are branched into separate sections, so as to render the entire footwell area extremely resistant to deformation. The forward extending longerons are designed to crumple in specific steps, so that the front end structure can deform

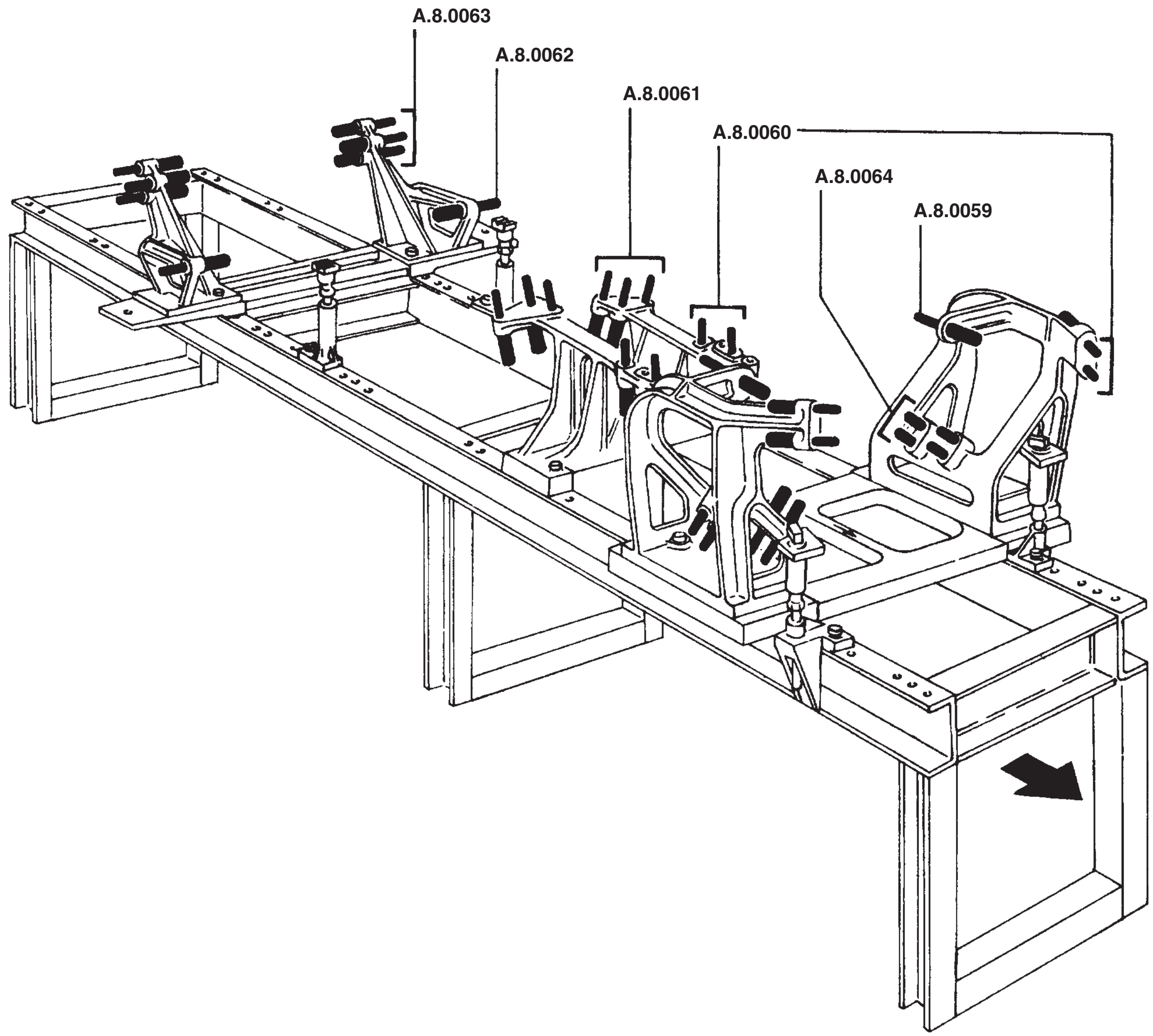


Figure 30.14 An example of a body alignment checking bench: the dowels as fitted are the same on both sides and symmetrical about the longitudinal centreline (*Alfa-Romeo*)

progressively and assist the absorption of energy by the main body structure. Similar considerations have also dictated the design of the rear end structure of the body. To contain sideways impacts the side structures of the body with their broad based pillar-to-sill connections are cross-braced, the sills themselves intersecting generous size crossbars spanning the front and rear of the passenger compartment. Additionally, the doors are provided with reinforcing impact members.

The advantages associated with this new form of aluminium alloy integral body construction may therefore be summarized as follows:

1 It contributes to an overall reduction in the weight of the car, because the engine can be smaller and lighter, the suspension mechanism can be lighter and the fuel tank can be smaller as less fuel is needed to propel a lighter car.
2 The improved power-to-weight ratio of the car allows a better performance for less power output, so that fuel economy and emissions reduction will both benefit.
3 From consideration of 1 and 2 the car will handle better on the road and will also accelerate faster in response to traffic conditions for safer driving.
4 The structural stability of the body helps to reduce vibrations and improves noise control.
5 From consideration of 4 the body also confers better energy absorption thereby enhancing passive safety for the occupants.
6 The greater use of aluminium alloy can be of benefit to the environment, because it has recognized potential in recycling.

Later developments

The 2003 Jaguar luxury car represents a further development in aluminium body construction. It differs from the space frame design in more closely resembling a conventional steel body construction. This is achieved by using pressings of dent-resistant aluminium sheet for the body panels; aluminium extrusions for the roof, doors and safety areas; and aluminium castings where increased strength is required for the door hinge pillars, bolt-on front end, and to mount the engine, transmission and running gear. Other points of interest concern the use of aerospace methods of joining the structural elements. These methods involve using self-piercing rivets that can accommodate up to four layers of material without breaking through the final layer, and structural bonding with epoxy adhesive to provide a stiff connection and a uniform force distribution between the parts to be joined.

From a service point of view any major repairs to an aluminium body are likely to demand specialist treatment, while a generally higher level of skill may also be required for any minor repairs where aluminium welding is involved.

30.3 MULTI-PURPOSE VEHICLES

General background

These represent a relatively new category of vehicle, which has continued to gain in popularity since Chrysler introduced their successful Voyager model in 1983. A multi-purpose vehicle, usually now abbreviated to MPV and earlier known as a 'people carrier', basically gives the impression of a saloon passenger car that has been enlarged both in length and height to accommodate three rows of seats, the seats having a swivelling facility in some designs. There may also be provision to convert seats into tables and for the ready removal of seats to increase cargo space, thus emphasizing the versatility of the MPV. It is typically of box-like form with pronounced sloping of the nose and windscreen, utilizes either a front engine, front-wheel drive or, less commonly, a front engine, rear-wheel drive layout for the mechanical elements, and is intended to ride and handle as nearly as possible like a conventional passenger car. A further development has been the application of a part-time four-wheel drive system to this type of vehicle.

MPV body construction

This generally takes the form of an all-steel integral construction body including sills, which is welded to a ladder-type underframe that extends from the nose to an extreme rear cross-member, so that the whole structure combines strength with lightness and also provides a flat platform for the passenger compartment. The underframe comprises two generously-proportioned longerons which, according to the length of the vehicle, are braced by seven to ten cross-members with their sectional widths increased where appropriate, such as for supporting the engine and transmission units. To complement the front end structure the forward ends of the longerons are supplemented by upper side-members of smaller section, these extending forwards from the A posts at the lower level of the windscreen.

As earlier described for conventional passenger cars, the ends of the longerons and side-members incorporate fold initiators to control deformation in the event of a frontal impact. Front, intermediate and rear stiffening beams are similarly integrated into the roof structure to confer better protection in roll-over accidents. For noise suppression the platform floor and the roof panel may be ribbed to stiffen them and raise their natural frequency of vibration. An MPV may have either conventionally hinged doors or front hinged and rear sliding doors, the object of the latter being not only to facilitate access but also to allow more convenient removal and refitting of the seats according to seating requirements. At the rear of the body a rigid frame is provided to mount the tailgate. The various sections that contribute to the stiffness in the body structure of the modern Nissan Serena MPV, which has a front engine, rear-wheel drive layout, are shown in Figure 30.15.

Sports utility vehicles

Another type of multi-purpose vehicle that has become increasingly popular is the 'sports utility vehicle', usually now abbreviated to SUV, which first appeared on the American market in the early 1990s. It is basically an amalgam of pickup truck and passenger car, but differs from the latter in having a noticeably increased ground clearance, a narrower wheel track and a higher centre of gravity. A further development of the SUV is the Chevrolet Avalanche, introduced by General Motors in 2002, and described by them as an Ultimate Utility Vehicle. In effect it can be converted from an SUV to a pickup truck, simply by removing and stowing the rear window assembly, folding forward the rear seat and lowering what is termed a 'midgate' (similar to a tailgate), thereby creating a generous cargo space. The attractions of an SUV are generally those of providing a more commanding view of the road by virtue of the higher seating position, and possessing a reasonable off-road driving capability, which is conferred by the increased ground clearance and typically enhanced by a four-wheel-drive transmission system. On the debit side safety considerations preclude making abrupt changes in direction and also require speed to be moderated when encountering gusty cross-winds, owing to the taller build and higher centre of gravity of the vehicle.

Structurally, this type of vehicle was originally more closely related to light trucks rather than passenger cars. That is, an often utilitarian body was mounted on a separate ladder chassis frame. This type of construction is now less used than formerly, an integral construction body based on passenger car practice being the preferred option, except where there is a requirement to cater for really arduous off-road driving. An integral body construction generally allows an SUV to be lower in height, while maintaining an acceptable ground clearance, and also to be less heavy. Both of these considerations can prove beneficial to fuel consumption.

30.4 COMMERCIAL VEHICLE CHASSIS FRAMES

General arrangement

The structural foundation of a commercial vehicle lies in its chassis frame, which in long-established practice has taken

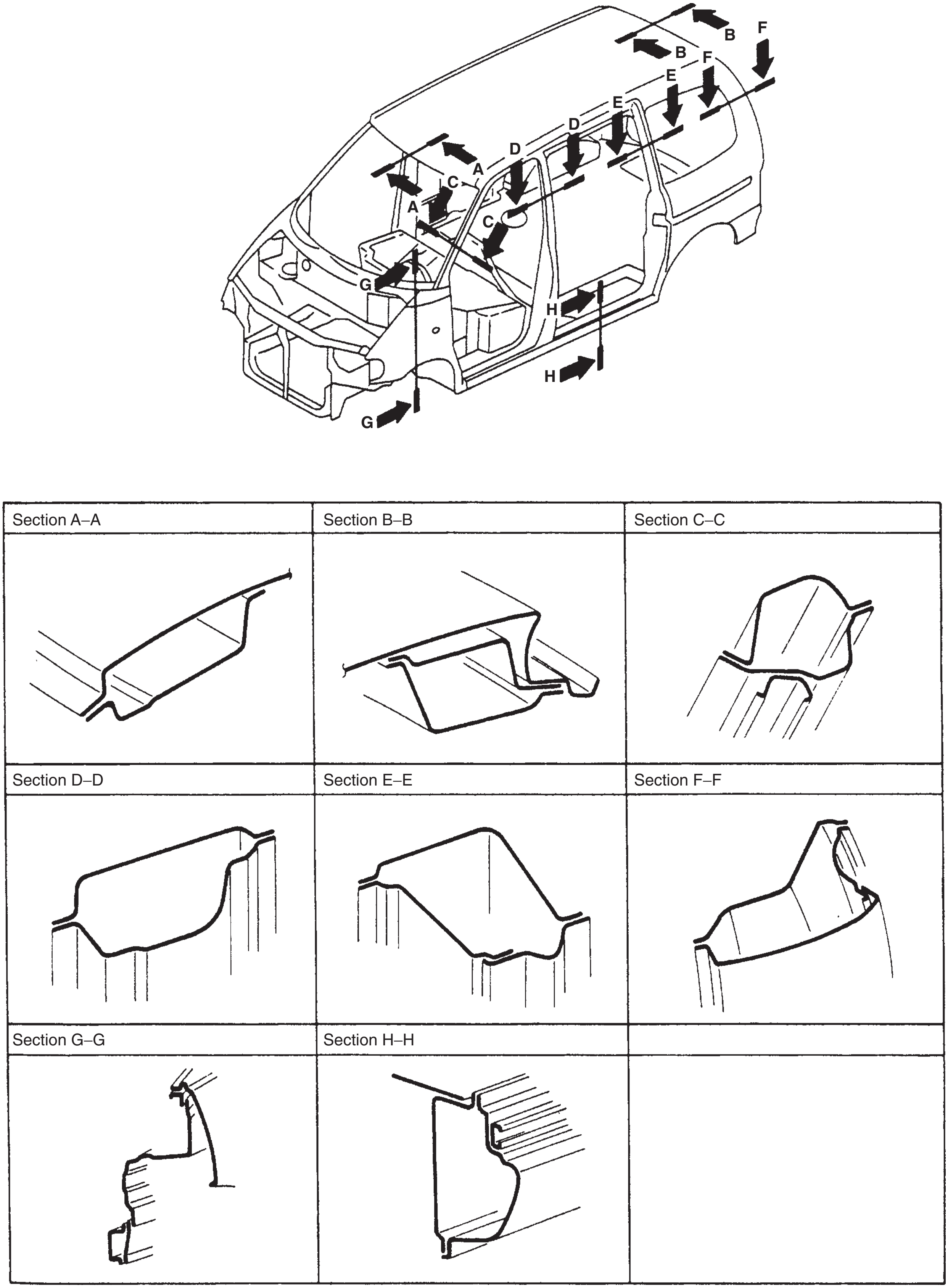


Figure 30.15 Body stiffening sections of Nissan Serena MPV (*Nissan Motor GB*)

an approximately rectangular form resembling a ladder. Indeed, this type of chassis frame is often referred to as a ladder frame, because basically it comprises two side members joined by a series of cross-members (Figure 30.16). Attached to these frame members are numerous mounting brackets for the vehicle mechanism and body (Figure 30.17) their arrangement being typically as follows:

Cab body This is supported from mounting brackets attached to the forward ends of the side members and may in turn incorporate stanchions (upright supports) to secure the radiator. Rubber cushion mountings are generally provided at the cab body mounting points to insulate the cab from vibrations and to minimize any racking strains imposed upon it by the flexibility of the chassis frame. A forward-tilting cab with hand-operated hydraulic tilt mechanism may be used to facilitate maintenance work on the engine unit of some commercial vehicles.

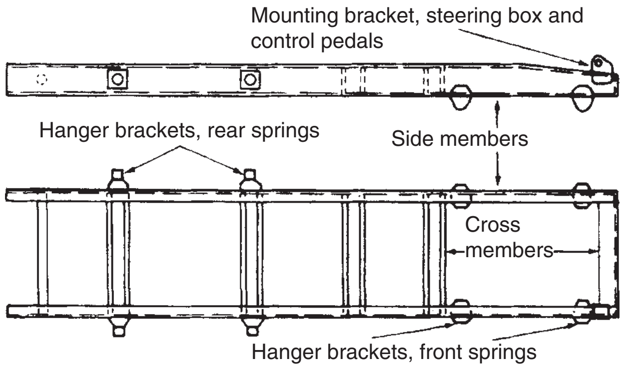


Figure 30.16 A commercial vehicle ladder chassis frame

Steering box An extremely rigid mounting bracket is attached to the front end of the side member which supports the steering box and, in some cases, the control pedals. Rigidity of mounting is usually assisted by the nearby attachment of the front cross-member.

Front springs Extending below the forward portions of the side members are two pairs of hanger brackets from which the front springs and axle are pivoted. To maintain front end alignment and distribute stresses evenly within the frame, the forward cross-members are often positioned to coincide, or nearly so, with the attachment points for the spring hangers. In the case of air sprung front axles, repositioning of the front cross-members together with additional mounting brackets is usually necessary (Figure 23.52).

Power unit This is supported either from the first and second cross-members, or from one of these cross-members and both side members. Suitable attachment brackets are provided at these points for mounting the combined engine, clutch and gearbox unit on rubber cushions. Alternatively, the power unit may be suspended from rubber-bushed links. These various systems of flexible mountings serve to isolate the frame and cab from vibrations caused by the idling diesel engine.

Propeller shaft For reasons that have been explained in Section 19.3, the long length of propeller shafting required on many commercial vehicles generally demands at least one intermediate support bearing. Again, this is rubber cushioned and it is mounted in a bracket underslung from one of the middle cross-members.

Rear springs These pivot from two pairs of outrigger brackets attached to the rear portions of the side members. The rearward cross-members are almost invariably positioned to coincide with these spring mountings and therefore maintain alignment of the sprung rear axle relative to the

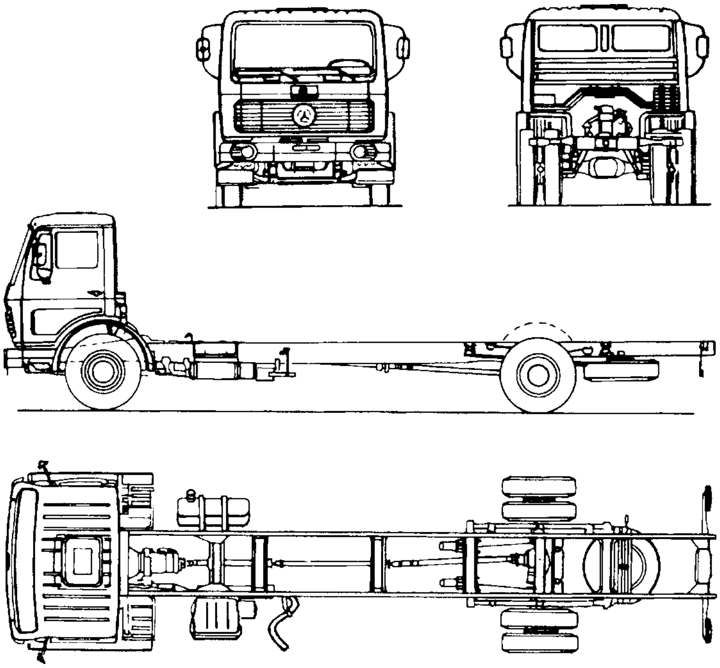


Figure 30.17 General arrangement of a four-wheel commercial vehicle (*Mercedes-Benz*)

chassis frame. In the case of air and rubber springing for rear axles and tandem axle bogies, the repositioning of cross-members together with additional mounting brackets again is usually necessary (Figures 23.40, 23.41 and 23.46).

Spare wheel This may be supported from brackets attached beneath the extreme rear end of the chassis frame, which is usually closed by a final cross-member.

Other services Various other brackets are provided on the frame, especially about its middle portion, chiefly to support the fuel tank on one side and the exhaust system on the other, as well as the reservoirs for the air-operated braking system.

In 1998 Mercedes-Benz introduced a 'two-piece' frame construction in which a common frame-head can be attached to differing lengths of rear ladder frame, according to vehicle application.

Chassis frame construction

So far the commercial vehicle ladder frame has been identified as comprising basically two side members joined by a series of cross-members. It is now necessary to examine this structure in a little more detail, especially with respect to the material sections used, the different methods of joining the frame members, and the means by which they may be reinforced for greater rigidity, as follows.

Material sections

Channel section steel pressings of constant web height are usually employed for the chassis frame side-members. Some examples of their dimensioning for the cold riveted frames used on a Volvo range of trucks are tabulated below:

Although the use of channel section provides adequate resistance to vertical bending loads, it is much less efficient in resisting sideways bending and torsional (twisting) loads (Figure 30.18). However, it does lend itself to the economic production of commercial vehicle frames for the numbers required, and also provides for easy attachment of the cross-members and the various other mounting brackets.

The cross-members must resist torsional deflection of the frame as a whole imposed by rocking movements of the sprung axles, and also prevent individual twisting of the side members arising from overhung loads such as the fuel tank. Various material sections are used for the cross-members, including I-section beams fabricated from back-to-back channel sections, top-hat and tubular-section beams (Figure 30.19). I-section cross-members are often used towards the front of the frame, because they can better resist bending loads in the horizontal and vertical directions, thereby maintaining front end alignment and supporting the power unit. Top-hat-section cross-members may be used towards the middle of the frame, since they more readily lend themselves to special profiles. These can be required to provide clearance for a component, such as an intermediate bearing for the propeller shaft. Tubular-section cross-members are normally found towards the extreme rear end of the frame, where they offer good resistance against torsional loads.

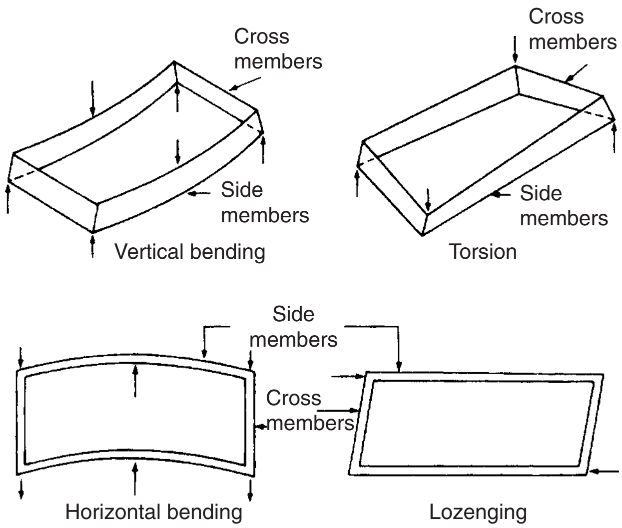


Figure 30.18 Modes of chassis frame deflection (exaggerated for clarity)

Table 30.1 Chassis frame details of Volvo trucks

Model	*GVW (kg)*	*Frame width (mm)*	*Web height (mm)*	*Flange width (mm)*	*Web thickness (mm)*	*Flange thickness (mm)*
FL608 4×2 Rigid	7500	860	220	75	5	5
FL612 4×2 Rigid	12000	862	230	75	6	6
FL618 4×2 Rigid	18000	864	275	90	7	7
FL7 6×4 Rigid	26500	800	275	90	8	10
FL7 8×4 Rigid	34200	800	275	90	8	10
FL10 4×2 Tractor	19500	800	275	85	6	6
FL10 6×4 Tractor	25500	800	275	90	7	7

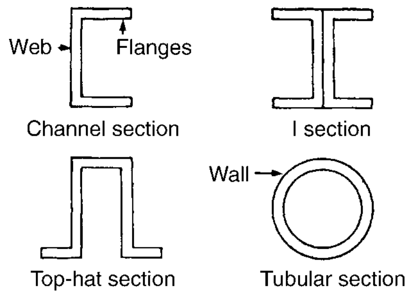


Figure 30.19 Frame member sections

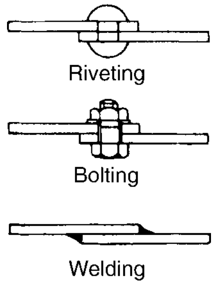


Figure 30.20 Methods of joining the frame members

Joining the frame members

The processes that may be employed to join the side and cross-members are welding, riveting and bolting (Figure 30.20).

Welding gives the most rigid joint, but this method has so far been generally uneconomic for the production numbers involved, especially since machine welding would be required to guarantee the necessary weld uniformity. Welding is therefore usually confined to the fabrication of individual cross-members.

Riveting is widely used for joining the frame members, and it may be done cold rather than hot so as to avoid the rivets contracting during cooling and developing clearances in their holes.

Bolting is also popular and can be more effective than riveting as regards rigidity because the tightness with which the frame members are clamped together can be accurately controlled. A possible disadvantage is that screwed connections may loosen if not properly installed.

A combination of riveting and bolting is sometimes employed; for example, a bolted connection may be confined to the front cross-member so that it can be readily removed to facilitate engine replacement.

Reinforcing the frame joints

To ensure that the inherent rigidity of I- and top-hat-section cross-members is effectively transmitted to the side members, both their mode of attachment and method of reinforcement are given careful attention. Each cross-member is usually attached to the flanges of the side members rather than to the more flexible web. The joints are then reinforced by gusset plates interposed between the top and bottom faces and their mating side member flanges in the case of I-section cross-members, and angled between the front and rear faces and the side member webs where top-hat-section cross-members are used (Figure 30.21).

Apart from reducing stress concentration at the frame joints, these gussets play an important part in resisting sideways bending loads on the frame and also what is known as lozenging of the frame (Figure 30.18). The latter refers to any tendency of the frame side members to move longitudinally relative to each other, which may arise from unequal driving or braking forces being transmitted to the frame.

In long-established practice the chassis frame members of heavy commercial vehicles are produced from carbon-manganese steel pressings.

Chassis cab construction

The cabs mounted on commercial vehicle chassis frames now have to comply with ECE safety standards. In the case of the Volvo trucks mentioned earlier, which also have to meet Swedish safety standards, their cabs are manufactured from sheet steel panels hot-dip galvanized to protect the vehicle from corrosion, whilst the entire cab is treated with an anti-rust compound. Key external panels such as the front grille and step-well area are finished in plastics to protect the cab from road debris. Plastics panels and trim material are labelled and identified to ease recycling.

A cab-over-engine (as opposed to the American cab-behind-engine) chassis layout is generally favoured by European heavy vehicle manufacturers, chiefly because it provides maximum cargo space for a given overall length of vehicle, and also confers better visibility for the driver (Figure 30.17). To facilitate engine maintenance a hydraulically operated tilting mechanism may be fitted to the cab, so that it can be tilted forward about its front anchorage points. According to system design it can accommodate either full over-centre tilting of the cab, or optional partial lifting and full tilting facilities. Such a system basically comprises a pivoting hydraulic ram that is mounted between the chassis frame and the cab, which connects through flexible hoses to a hand-operated, double-acting, pump with integral fluid reservoir. A slow controlled descent of the cab, once it has passed either way over-centre, is imposed by bleed valves in the ram.

Caution The manufacturers' instructions for its safe operation must be strictly observed.

Checking chassis frame alignment

The time-honoured method of checking chassis frame alignment is by measurement of diagonals, using a plumb-bob and a chalked string. To carry out this check the vehicle should be placed on level ground with a clean surface and the handbrake applied. Initially, the positions of the forward attachment points of the front springs and the rearward attachment points of the rear springs should be transferred to the ground. This is done by chalking an area beneath each position of the plumb-bob and then drawing a cross with a pencil or scriber such that the centre of the cross represents the plumb line. The four crosses on the ground should now accurately represent the extreme corners of the frame, so that if the vehicle is

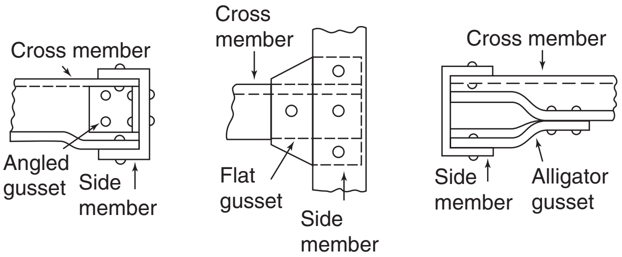


Figure 30.21 Methods of reinforcing the frame joints

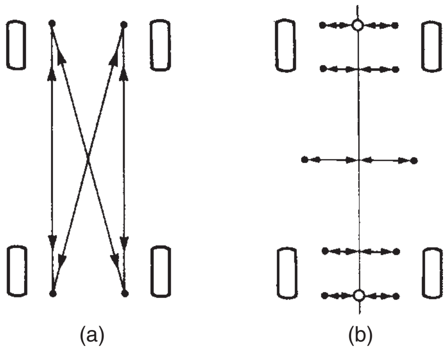


Figure 30.22 Checking chassis frame alignment

moved away the dimensions of the diagonals and lengths of the side members can be determined exactly (Figure 30.22a). If opposite dimensions are within about 6 mm (0.25 in) the frame would generally be regarded as satisfactory, unless the vehicle has received a side impact.

In this case, further points are taken on each side of the frame and plumb lines dropped to the ground. Suitable points are at the rearward attachments of the front springs and the forward attachments of the rear springs, and also two corresponding points on each side of the centre of the frame. There will now be five crosses on the ground on each side of the frame; therefore if the vehicle is moved away again, an exact copy of the essential points of the frame are pictured on the ground. The points at each end of the frame can next be joined by lines and these lines bisected and a line representing the centre of the frame can be marked by stretching a chalked string between them, carefully lifting the centre of the string and allowing it to snap back to the ground so that a white line is marked thereon. From this centreline all the points marked can be checked for correct relative separation (Figure 30.22b), which will indicate whether the frame is bent or not.

Centralized chassis lubrication systems

It is worth recalling that the chassis lubrication requirements of the modern car are exceedingly modest as compared with past practice. For example, in a Ford Popular model of the 1930s one was required to 'lubricate thoroughly with a grease-gun' no fewer than 19 chassis grease nipples every 1000 miles. In the case of larger cars with more elaborate running gear, the demands could be even greater; an example here is the famous Rolls-Royce Silver Ghost of an earlier period, with 25 chassis lubrication points that required attention weekly or every 500 miles. In return for this dedicated attention, one was advised that 'properly lubricated, a Rolls-Royce car will run indefinitely'.

To avoid the chore of hand lubricating the chassis bearings, the original concept of a centralized chassis lubrication system comprising a central pump discharging oil through common supply lines was introduced (perhaps as might be expected) by the American motor industry in the mid 1920s and was first used on Packard cars. Known as the Bijur lubricating system, it was later adopted in Britain, where it continued to find applications on high-grade motor cars, notably Daimler and Rolls-Royce, until the late 1950s. In contrast to these earlier chassis lubrication requirements, the modern car with its rubber-bushed suspension components and 'lubricated-for-life' suspension and steering ball joints generally does not have any chassis lubrication points that require routine attention.

Nowadays, centralized chassis lubrication systems are confined to commercial vehicle applications, which can still feature numerous metal-to-metal bearings throughout their chassis. In a sophisticated modern example of such an installation, up to 60 chassis bearings can be automatically grease lubricated from a central, pneumatically operated, pump assembly. This embodies individual pumping units that feed a multiline distribution system, each bearing being connected to its respective pumping unit by a length of nylon tubing, and receiving precisely metered quantities of grease on an electronically controlled time cycle.

Similar installations are designed to use a transmission grade oil as the chassis lubricant (Figure 30.23).

The automatic chassis lubrication system of a commercial vehicle typically supplies the following points;

Road spring pivot and shackle bearings
Tandem axle balance beam bearings
Front axle king-pins
Track rod ball joints
Drag link ball joints
Power-assisted steering cylinder pivots
Brake camshaft bearings
Brake slack adjusters
Clutch operating shaft bearings
Gear lever linkage
Pedal linkages
Accelerator cross-shaft bearings
Tipping body hinges
Fifth-wheel coupling pivot and jaws.

It must, of course, be added that a centralized chassis lubrication system will only continue to function in an automatic mode, as long as its lubricant reservoir is replenished at intervals specified by the manufacturer. Its pipework and fittings must also be periodically inspected for any accidental damage that could restrict lubricant delivery to any point.

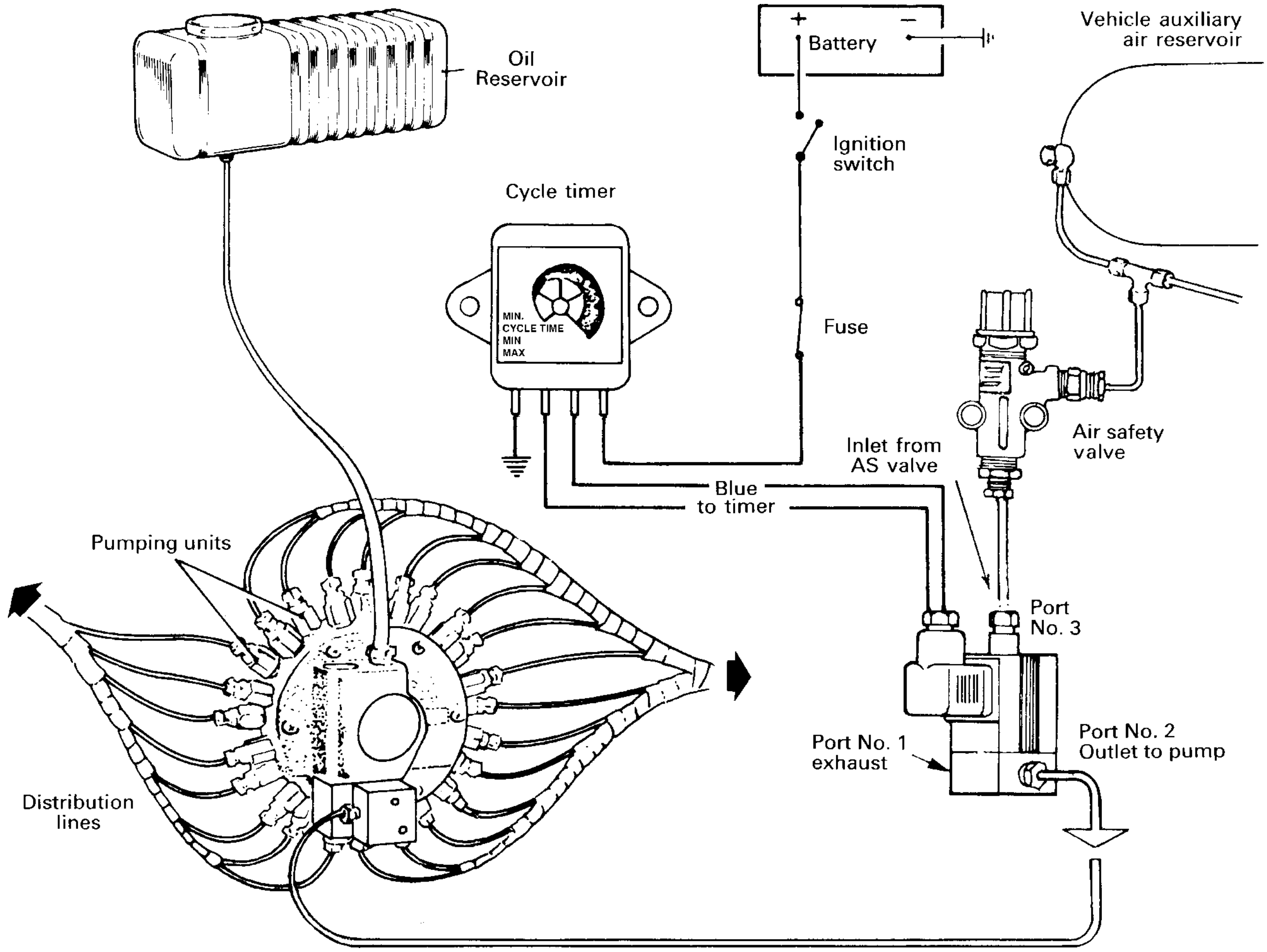


Figure 30.23 Airdromic timed system of automatic chassis lubrication (*Interlube*)

30.5 TRAILER AND CARAVAN COUPLINGS

Trailer couplings

The simplest types of trailer coupling are those used between a four-wheeled, or less commonly six- or eight-wheeled, rigid vehicle towing a drawbar trailer. This type of trailer is identified as having axles at both front and rear, the front axle being able to swivel in the manner of a turntable and connected to the rear of the towing vehicle by means of a cross-pivoted drawbar. The actual forms of coupling used between the rear of the vehicle and the apex of the trailer drawbar are basically the 'pintle hook' and the 'drop pin' (Figure 30.24a and b). Either of these is mounted from the rear cross-member of the vehicle frame and engages the drawbar eye of the trailer. A cushioning rubber sandwich may be incorporated in the coupling mounting, so as to reduce shock transference between the vehicle and trailer. Safe operation of these couplings is provided by swing-down latches for pintle hooks and internal locking for lever retracted drop pins. For more convenient operation of drop pin couplings on heavy vehicles, remote retraction of their pins may be conferred by pneumatic actuators.

More comprehensive forms of coupling are required in the case of articulated vehicles, which comprise a towing vehicle or tractive unit and a semi-trailer. A four- or six-wheeled vehicle may be used as the tractive unit, and the semi-trailer is identified as having from one to three axles at the rear end only. Its front end rests on and is coupled to the tractive unit, which supports 20 per cent or more of the weight of the trailer when it is uniformly loaded. Landing gear legs are provided to support the front end of the trailer when it is uncoupled from the tractive unit.

An early development of the semi-trailer coupling was the ingenious but relatively complicated automatic coupling. This was introduced in the early 1930s by Scammel Lorries Ltd in England for their three-wheeled Mechanical Horse tractive unit and semi-trailer, a combination that remained popular for local haulage over many years. The automatic coupling was later developed for application to more conventional tractive units and semi-trailers, but the limitations placed on its carrying capacity were such that it was generally superseded by the more rugged fifth-wheel type of coupling described later. In brief, the automatic coupling required the rear of the tractive unit to be equipped with a pair of ramps,

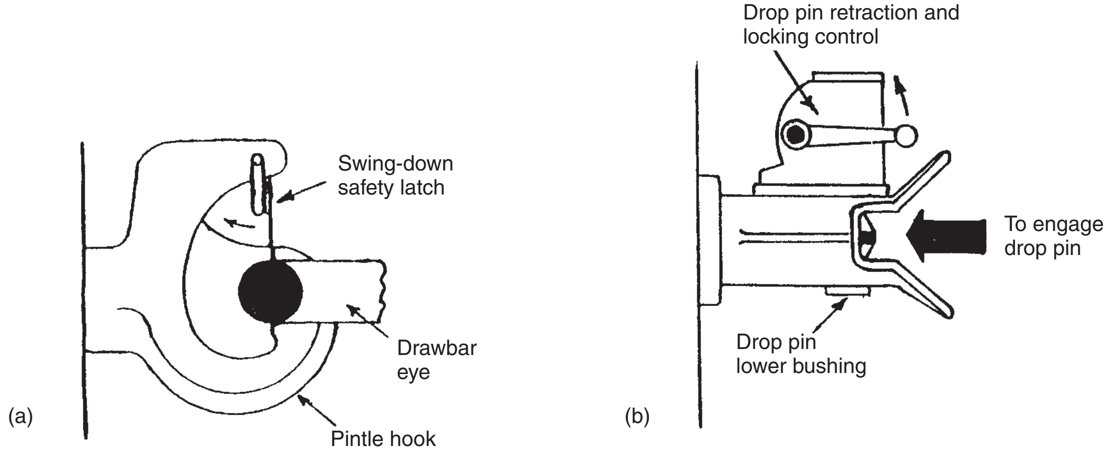


Figure 30.24 Drawbar trailer couplings: (a) pintle hook (b) drop pin

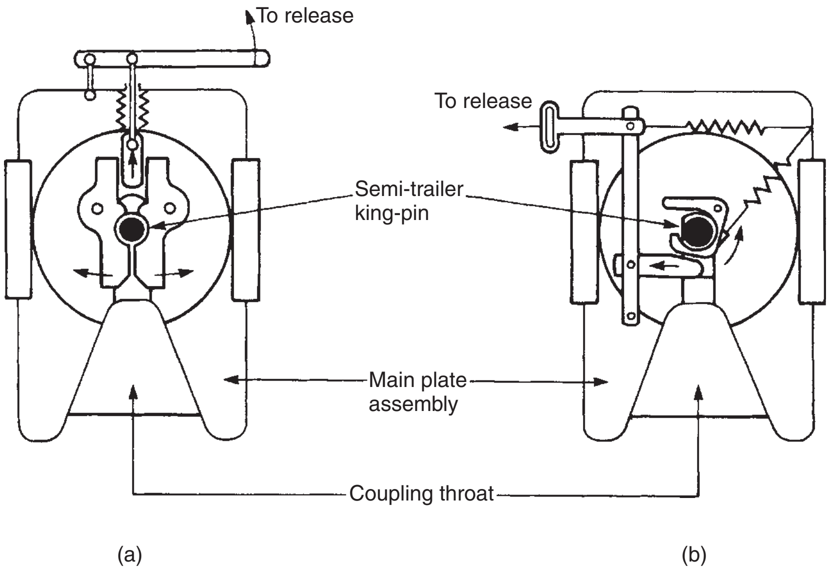


Figure 30.25 Fifth-wheel couplings for semi-trailers: (a) twin jaw (b) hook and wedge

their purpose being to lift the front end of the semi-trailer together with its retractable two-wheeled undercarriage off the ground when the tractive unit was reversed under the trailer. The undercarriage then released and folded away as a pair of flanged rollers mounted on the trailer moved up the ramps until the trailer turntable automatically locked into position on the tractive unit. Weekly greasing of the undercarriage joints was essential to ensure their freedom of movement and safety in operation.

As earlier inferred, the fifth-wheel coupling has long since become established practice for hitching a semi-trailer to its tractive unit. Those parts of the fifth-wheel coupling that are attached to the tractive unit comprise a tilting main plate assembly, which forms the mating lower half of the turntable and is mounted from a support plate that is secured rigidly to the rear of the vehicle frame. The main plate assembly is provided with grease retaining areas on its turntable surface and a V-shaped coupling throat. This terminates in either a twin-jaw or a hook-and-wedge locking mechanism (Figures 30.25a and b). To complete the fifth-wheel coupling a rubbing plate is secured to the underside of the trailer and, together with its central down-pointing king-pin, forms the mating upper half of the turntable.

The coupling process, or locking up as it is sometimes called, is effected when the main plate lock approaches and then grips the king-pin, as the tractive unit is reversed slowly under the front end of the braked semi-trailer. Before this operation is performed the unlocking mechanism of the coupling main plate must be correctly set, which basically involves withdrawing a locking plunger of a twin-jaw lock (Figure 30.26), or the wedge of a hook-and-wedge lock. After coupling, the unlocking mechanism is allowed to spring return to its locked position. In some modern installations of fifth-wheel coupling the unlocking mechanism is pneumatically assisted. It is important that the main plate and locking mechanism of a fifth-wheel coupling are periodically cleaned, inspected for wear and re-lubricated in accordance with manufacturer's instructions. However, lubrication of the main plate is not required where this is fitted with a grease-free, low friction liner made from a special composite material. The full procedure for the safe coupling and uncoupling of a semi-trailer from its tractive unit is, of course, all part of the training received by the HGV driver.

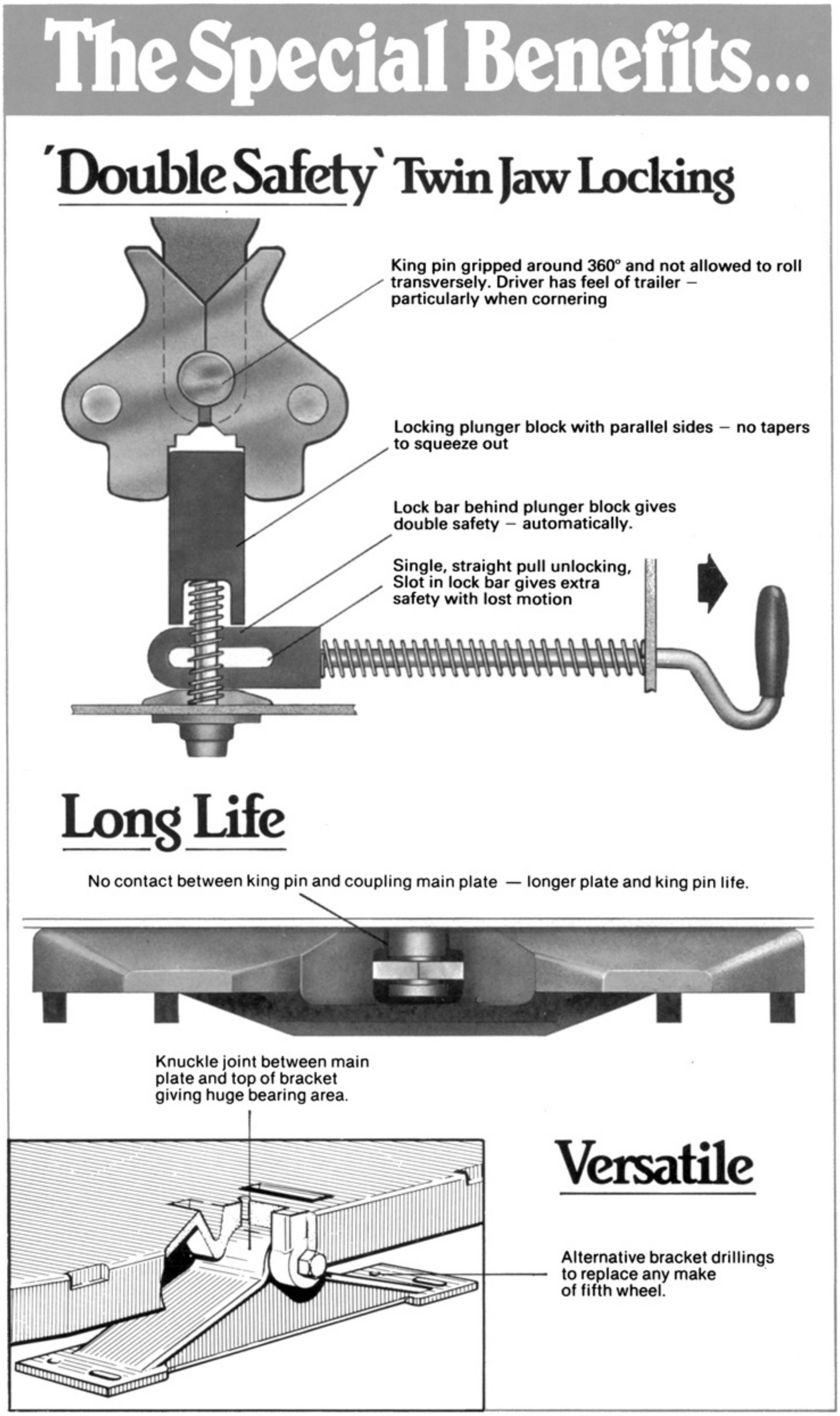


Figure 30.26 Safety features of a modern fifth-wheel coupling (*Davies Magnet*)

Jack-knifing and swing of articulated vehicles

Early opposition to the now widespread adoption of articulated commercial vehicles was based on fears of the tractive unit and semi-trailer jack-knifing, especially when braking on slippery road surfaces. The term *jack-knifing* is used to describe a condition of instability where the semi-trailer pushes the rear of the tractive unit out of line, thereby creating a toggling effect that results in the tractive unit partially rotating and closing on the trailer. Research engineers have long ago established that the most sudden form of jack-knifing occurs with premature locking of the rear brakes on the tractive unit, but that it may similarly occur if the brakes lock prematurely on the trailer and cause it to swing over (Figure 30.27). Jack-knifing can also be provoked by a sudden swerving manoeuvre, even without the brakes being applied.

Apart from ensuring at the design stage that the distribution of braking effort between the axles of articulated vehicles is such as to minimize jack-knifing, a further approach is the fitting of an anti-jack-knife device. A widely accepted type is that manufactured by the British firm of Hope Technical Developments Ltd., which comprises a sealed multidisc clutch unit that is trailer mounted and air pressure operated (Figure 30.28). Its moveable set of discs is splined to a rotatable king-pin, which is also provided with a torque reaction arm that engages with the throat of the fifth-wheel coupling plate on the tractive unit. Therefore when the vehicles brakes are operated, the clutch discs are squeezed together and damp out any dangerous or uncontrolled rotation between the tractive unit and semi-trailer. An important feature of its operation is that it provides torsional damping, related to the brake application, at the kingpin slightly before braking takes place through the road wheels, so that any would-be tendency to swing is damped out from the start. The early application of this relatively small controlling force does not impair the manoeuvrability of the tractive unit and semi-trailer, so the need to exert a more violent control at later stages is not necessary.

More recently the principles of vehicle dynamics control (Section 29.7) have been applied to tractor and trailer combinations. Such a system is intended to oppose any deviation from correct tracking of the steered vehicle. In simple terms

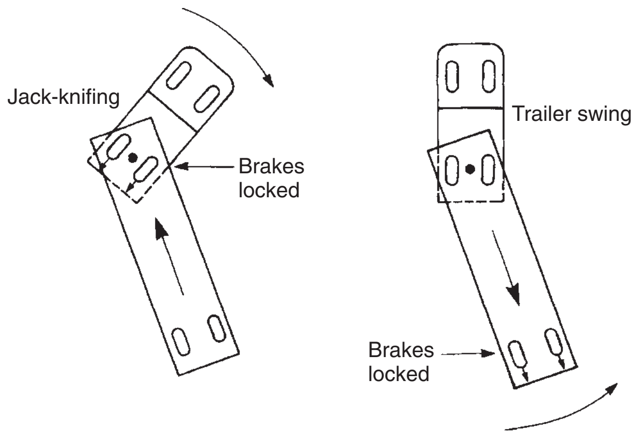


Figure 30.27 Jack-knifing and trailer swing of articulated vehicles

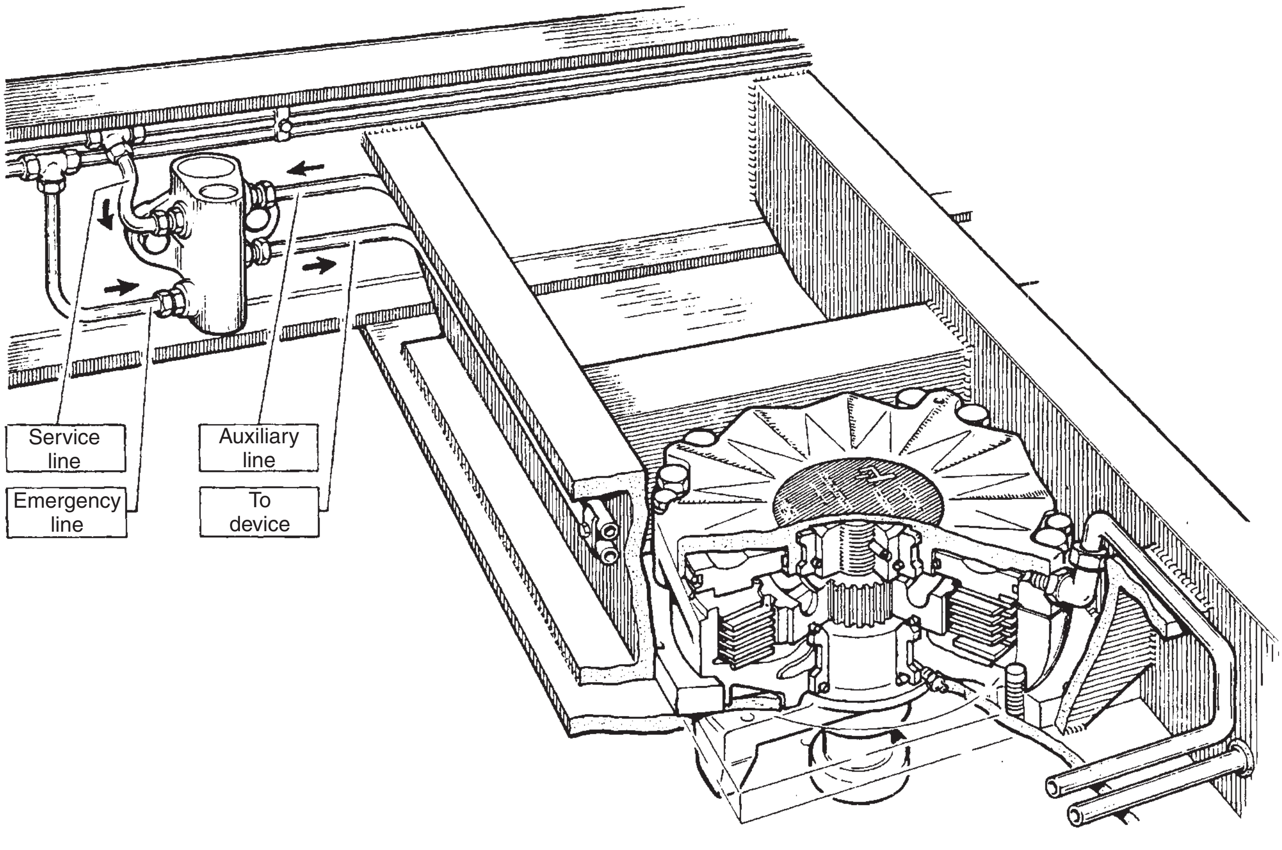
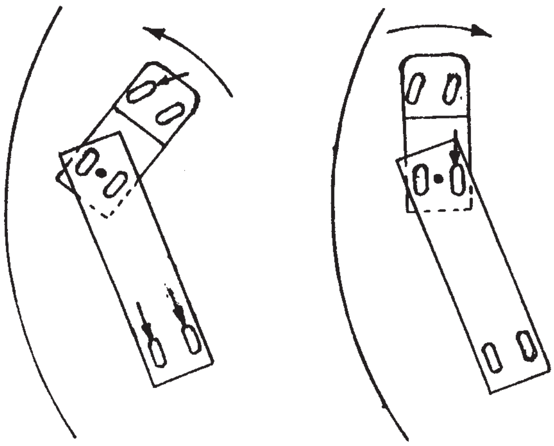


Figure 30.28 Anti-jack-knife device with pneumatic control valve (*Hope Technical Developments*)



(a) (b)

Figure 30.29 Correcting oversteer and understeer of articulated vehicles by electronically-controlled selective wheel braking: (a) oversteer – nearside front tractor brake and trailer brakes applied (b) understeer – offside rear tractor brake applied only

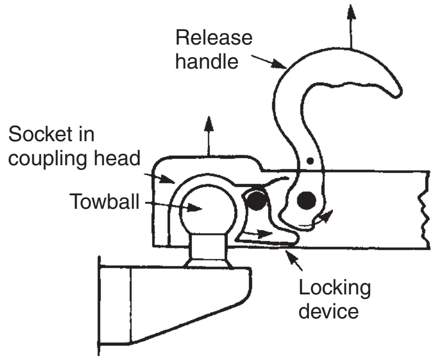


Figure 30.30 Caravan ball-and-socket coupling

if the tractor begins either to oversteer that could lead to a jack-knifing situation, or understeer and run too wide, then the wheel brakes are selectively applied by the electronic control system to correct the deviation. The overall effect is to produce an opposing yawing (or turning) action on the tractor (Figure 30.29) and within reasonable limits of speed restore vehicle stability. This type of system does not interfere with normal braking control.

Caravan couplings

A ball-and-socket type of coupling with a standard towball diameter of 50 mm (2 in) is used for caravan towing by a passenger car. The towball is mounted from a towing bracket that should be specifically designed for the purpose and is typically of T-form, so as to spread the load as widely as possible under the structure of the car. To complete the coupling a socket is formed in what is termed the coupling head, which is usually slidably mounted on the drawbar of the caravan because it also forms part of an overrun braking system. The socket itself comprises an upper cup portion that rests upon the top half of the towball and a lower spring-loaded locking device that engages the underside of the towball (Figure 30.30). A release handle must be lifted before the coupling can be disconnected. Coupling or uncoupling of the caravan is, of course, performed by respectively lowering or raising the coupling head over the towball, care being taken to apply the caravan handbrake before uncoupling on a slope. Maintenance of the coupling head is generally concerned with keeping the inside of the socket housing and its locking mechanism well greased, and applying a grease-gun to nipples that supply the bearings of the slidable drawshaft, typically twice a year.

Various types of stabilizer, employing either friction or hydraulic damping, may also be connected between the car and caravan. Their purpose is to prevent extreme swaying (or snaking as it is known) of the caravan, which could lead to serious instability of both car and caravan. Snaking is usually attributed to excessive towing speed and can be provoked by cross-wind gusts, wind buffeting from overtaking vehicles, incorrect weight distribution and wrong tyre pressures.

30.6 INTRODUCTION TO VEHICLE AERODYNAMICS

General background

In Section 1.1 it was mentioned that one of the factors opposing the motion of a vehicle was air drag. This assumes increasing importance the faster a vehicle is travelling, because the force opposing motion due to air resistance varies as the square of the speed. Therefore as speeds began to increase, the designers of early cars soon became aware of this energy-consuming factor and gradually tried to avoid sudden changes in body contour, so that the air displaced during the forward movement of the car was disturbed as little as possible. In other words, they began to adopt less box-shaped and more curved body forms that possessed a certain degree of ‘streamlining’, the purpose of which is to promote a steady airflow with no additional air crossing its path.

The term ‘aerodynamic’ may be simply defined as the science of air in motion, and as might be expected it first received serious study in connection with early aeroplanes and airships. A notable pioneer in this field was an Austrian engineer, Paul Jaray, who while engaged on airship design in the Germany Zeppelin works during World War I had scientifically developed streamlined forms as a result of both theoretical studies and practical wind tunnel tests on scale models. It was during these investigations that Jaray also conceived the idea of applying the same aerodynamic principles to the design of motor car bodies, whilst recognizing the important difference that a vehicle always moves close to a stationary road surface, which is a complicating factor. Beginning in the early 1920s Jaray designed several fully streamlined car bodies, which sought to deflect the air-flow mainly upwards, as well as rearwards, over the top of the body and then down to the rear end with the minimum of disturbance. Some ten years later a Swiss engineer, Wunibald Kamm, who was researching motor vehicle aerodynamics in Germany found that an extended tail for a streamlined body did not, as previously imagined, maintain a smooth airflow over the entire length of the body, but unhelpfully increased frictional drag of the air because of the larger surface area

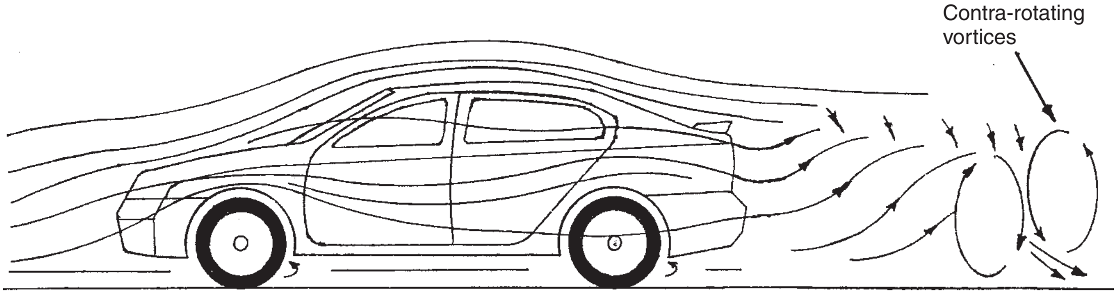


Figure 30.31 Schematic airflow around a passenger car

presented by the tail. His requirement was for a streamlined body that tapered moderately towards a relatively blunt end, which coincided with a region where the hitherto smooth airflow would in any event have separated from the body.

However, despite the rigorous application of aerodynamic principles to the motor car body by Jaray, Kamm and other research workers of that era, the seriously streamlined body found only limited acceptance by the motoring public. It did, of course, have its practical disadvantages, which could include restricted interior space, impaired rearward visibility, difficult parking if the rear overhang was increased, reduced accessibility for servicing, and from a purely styling point of view it was rarely pleasing to the eye. Although more restrained forms of body streamlining continued to influence car styling in general, it was not until the energy crisis of the early 1970s that there was a revival of interest in reducing aerodynamic drag to improve fuel consumption rather than car performance. Since then research into vehicle aerodynamics has continued unabated and now constitutes a highly specialized branch of automotive engineering, which has to acknowledge the sometimes conflicting requirements of attractive styling, acceptable handling characteristics in cross-winds, comfortable accommodation and passive safety for the occupants, and economic production.

30.7 BASIC CONSIDERATIONS OF VEHICLE AERODYNAMICS

Reasons for air drag

These are generally considered under the following headings:

Form drag It will be evident that form drag or profile drag as it is sometimes known, which derives from the basic shape of the vehicle body, represents the main source of aerodynamic drag. A visual impression of the airflow over the streamlined body of a passenger car adapted to modern styling is shown schematically in Figure 30.31. The popularly quoted 'drag coefficient' or 'Cd' value (also known as 'Cx' and 'Cw' values in France and Germany respectively), is used as a measure of how successful a manufacturer has been in producing a low drag body shape. For example, the advanced V6-engined Audi A8 model previously referred to in Section 30.2 is quoted by the manufacturer as having the commendably low Cd of 0.28, which compares to a more usual Cd in excess of 0.30.

The Cd value is non-dimensional and may be expressed as follows:

$$\mathrm{Cd} = \frac{\mathrm{D}}{0.5\,\mathrm{pAV}^2}$$

where D is the aerodynamic drag force (N) measured in a wind tunnel, p is the density of air (kg/m^3), A is the frontal area in terms of the greatest cross-sectional area of the body (m^2) and V is the vehicle speed (km/h).

Hence, it follows that given two cars with the same frontal area and running under identical conditions, the one with the lowest Cd value will generate the least air drag and therefore possesses the aerodynamically more efficient or 'slippery' body shape. This typically displays such features as a gently sloping front end with a pronounced sloping for the rear end in the case of fastback or hatchback bodies, generally well-rounded contours at the junctions between the windscreen, roof panel and side-windows, and a gradual narrowing of the body towards its front and rear ends.

Surface drag Between the surface of the moving body and the main airflow is a thin boundary layer of air, wherein the air molecules closest to the body surface tend to adhere to it, whilst those closest to the main airflow increase rapidly in speed to catch up with it. The effect is analogous to a pack of playing cards being pushed endwise as described for the lubrication process in Section 4.2. This sets up a shearing action in the air near to the body surface, which creates what is known as 'skin friction'. As a result energy is absorbed from the motive power of the car and dissipated in the form of heat. It will be evident that this type of aerodynamic drag force is chiefly dependent upon the surface area of the body.

Interference drag Any feature or fitting that interrupts the basic shape of the body, naturally has the effect of disturbing the smooth flow of air over the car. This interference to the airflow arises from superimposed eddy currents and turbulence, the latter being a more irregular swirling of the air, and again this results in energy being absorbed from the motive power of the car. There are perhaps a surprising number of sources that impose interference drag, ranging from such obvious items as number plates, spot lamps and exterior mirrors, to the less obvious ones of door handles, windscreen wipers and window mouldings, which explains of course the modern trend towards flush-fitting of these items. Further to reduce interference drag attempts are also being made to provide a smoother underbody structure for the car,

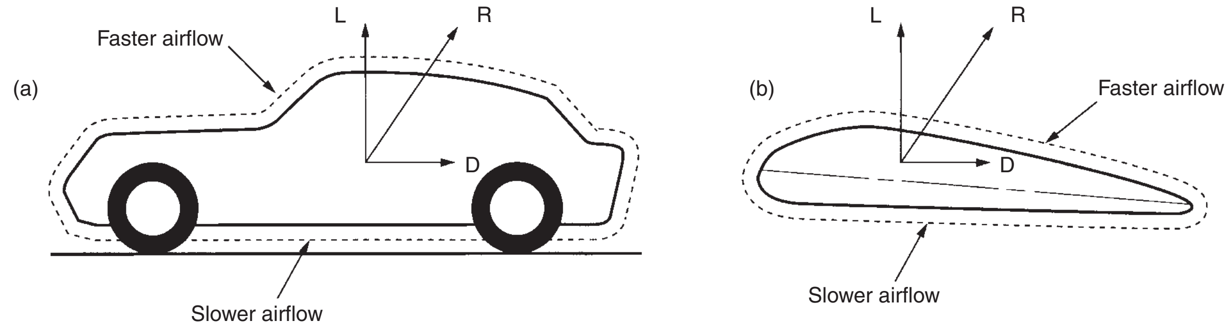


Figure 30.32 Simple comparison between the profile of a car and an aerofoil section for lift induced drag

R resultant aerodynamic force
L lift force
D drag force

together with suitable recessing for the exhaust system and other components wherever this is feasible. In recent practice, smooth plastics panels may be attached to the underside of the body and extend up to the rear suspension arrangements.

Lift induced drag In the streamlined airflow shown schematically for a modern car (Figure 30.31), it will be noticed that the disturbed flow of air downstream or in the 'wake' of the body takes the form of two trailing vortices of swirling air, in which the direction of rotation is upward from the outside of the body and downward to the centre. This pattern indicates that a pressure difference exists between the upper and lower surfaces of the car, which can impose lift induced drag forces at high speeds. It will be evident that, viewed in profile, the air flowing over the top of a car has further to travel than that flowing along its underside and must therefore be travelling faster. This will create a greater reduction in the pressure of the airflow above the car in comparison to that below it, the net result being a lifting effect on the car that resolves into a vertical lift force and a horizontal aerodynamic drag force (Figure 30.32a). In fact we can roughly liken the profile of a car to the aerofoil section of an aircraft wing (Figure 30.32b), but whereas the lifting force on the latter is needless to say very welcome, the opposite is true in the case of the motor car (let alone a Grand Prix car!), because it can cause serious instability at high speeds.

To counteract this aerodynamic lifting effect on modern high-performance cars, laterally mounted air deflectors known as 'airdams' and 'spoilers' may be deployed at the nose and rear deck respectively of the body, so as to reduce the difference in pressure of the air flowing above and below the car, but without significantly adding to the overall aerodynamic drag. The front under-bumper air dam diverts some of the air flow from the slipstream smoothly along the sides of the car, thereby increasing the speed and reducing pressure of the remaining air flow below it, which suppresses lift at the front end of the car. In contrast the rear spoiler slows down a region of fast moving air flow above the car. This not only increases its pressure to inhibit lift at the rear end of the car, but also reduces air turbulence behind it. (Wings of upside down aerofoil section are mounted on Grand Prix cars to achieve a similar but more pronounced effect). A sophisticated application of air dam and spoiler is that found in the Mitsubishi 'Active Aero System', where an automatically retractable front air dam is used in conjunction with a variable angle rear spoiler. At speeds above 50 mile/h (80 km/h) the air dam is deflected downwards and the rear spoiler is given an increased angle of attack to reduce aerodynamic lift. When the speed falls below 30 mile/h (50 km/h) the air dam is retracted and the rear spoiler returned to its original angle, the overall effect being to maintain constant the Cd value for the car.

Internal flow drag This represents a perhaps less obvious and unwanted contribution to the overall aerodynamic drag imposed on a car. It arises chiefly from the disturbance of the air directed through a stylized intake and passing through the radiator of the engine cooling system, then over the engine and numerous obstructions before escaping via the wing valances and underside of the engine compartment to join the main air flow. Carefully designed encapsulation of the engine compartment has been a recent development, which better manages the air flow emerging below the body. The internal flow drag imposed by the car interior ventilation, heating and air conditioning systems is much less significant. This is because the air passing through them is relatively slow moving, and its entry and exit points have been deliberately sited to take advantage of the air flow over the body (Section 5.6).

30.8 AEROCOUSTICS

It was earlier indicated that a high level of control over noise, vibration and harshness (NVH) has come to be expected of the modern passenger car. This also includes minimizing the effects of wind noise heard in the car interior, especially at the higher speeds of which cars are now capable. The term 'aerocoustics' (borrowed from the aerospace industry) is generally applied to the recognition and the remedies for this particular aspect of NVH. Wind or aerodynamic noise is transmitted as sound waves, which originate from pressure fluctuations in the external air flow. It therefore follows that any features of body design or equipment that interrupt the smooth flow of air over the car are also likely to generate turbulence and wind noise. Some of these sources have already been identified in connection with the previously mentioned interference drag on the body, others include the need for rigid window frames. The importance of maintaining efficient

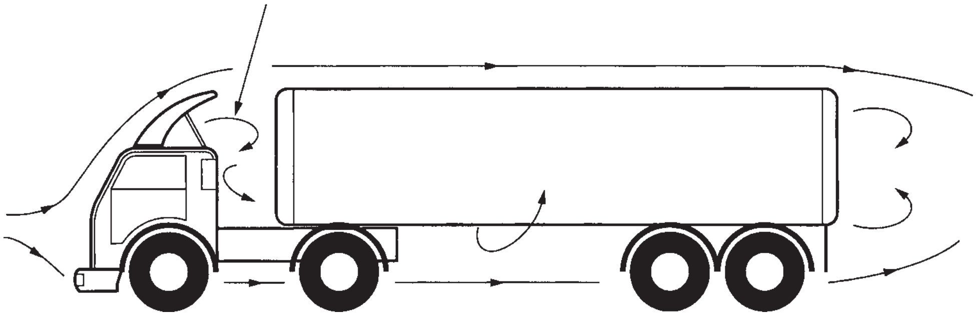


Figure 30.33 Improving the airflow over a heavy vehicle

body sealing cannot, of course, be overemphasized in minimizing the transmission of wind noise to the body interior.

30.9 HEAVY VEHICLE AERODYNAMICS

To conclude this introduction to vehicle aerodynamics mention must be made of its application to heavy vehicles, because fuel economy is of paramount importance to transport engineers. Clearly the necessarily box-like form of heavy vehicle bodies and their trailers does not readily lend itself to aerodynamic treatment, typical Cd values being in the region of 0.7 or about twice that for a passenger car, but since the late 1970s research engineers have continued to investigate means of improving matters. For example, the leading edges of the cab now tend to have a pronounced rounding, so that the airflow has less tendency to separate from the body surface and cause turbulence, which could otherwise occur with earlier slab-fronted styles of cab. The body may also have a tapered roof section at the rear.

A familiar bolt-on aid to maintaining a smooth flow of air over heavy vehicles is the 'wind deflector' mounted above the cab, which has a curvature designed to transfer with least disturbance the airflow over the vehicle body or a semi-trailer (Figure 30.33). In the latter case it helps to prevent turbulent and therefore energy sapping air currents from building up and spilling out from the gap between the tractive unit and the semi-trailer, or likewise with a drawbar trailer outfit. In any event the gap between them should be no larger than that required for free articulation. This type of cab-mounted aerodynamic device may form part of what is termed an 'air management kit', which can include an under-bumper air dam, side skirts to cover gaps and also improve stability in cross-winds, and other strategically sited deflectors according to particular vehicle requirements.

31 Alternative power sources and fuels

31.1 GENERAL BACKGROUND

A student new to the study of modern vehicle technology may be forgiven for assuming that the petrol engine, joined later by the diesel engine, has always reigned supreme during the one hundred years or so history of the motor vehicle. It may therefore be a matter of curiosity to discover that a student taking a City & Guilds of London examination in Motor Car Engineering in say 1909 was expected to answer the following questions: 'Sketch the engine of a steam car with which you are familiar and name the make'; and 'Why are electric cars not more commonly used for country work?'

The present-day, highly refined, petrol-engined passenger car would have been unimaginable to the early motorist, when the future of the petrol engine was by no means assured. It was in fact having to compete with steam and electricity as alternative sources of propulsion. The reciprocating engine for a steam car essentially comprised a boiler, in which steam was generated under pressure by the application of heat to water; together with a steam cylinder and mechanism so that the pressure exerted by the steam was controlled and modified to produce rotary motion at a drive shaft. No clutch or gearbox was required. The latter also applied to the electric car. In this case the principal parts were the batteries, electric motor and a controller.

To put matters into perspective, it is worth recalling what were considered to be the shortcomings of the petrol engine in early cars, as compared to the alternative sources of power. A reporter at the 1899 Automobile Exhibition in Paris listed these shortcomings as uncertain carburation, poor combustion, inefficient cylinder cooling, difficult starting and control of speed, and disagreeable noise and smell. In contrast a motor exhibition held in New York in the same year was entirely devoted to electric vehicles. At that time, the production of electric cars in America was only slightly less than that of steam cars and substantially more than petrol-engined cars. Nor could the performance of record breaking electric and steam cars be underestimated. Again in 1899 the Belgian, Camille Jenatzy, driving an electric car of his own design, was the first man to travel at a speed of over a mile per minute; while in 1906 the American, Fred Marriot, driving a Stanley steam car became the first man to reach a speed in excess of two miles per minute.

At this point we must pause and briefly consider why the petrol-engined car fairly soon superseded its reciprocating steam and battery-electric rivals. In favour of the steam car was its quiet operation, smooth running and useful range of power. However, early steam cars could require some twenty minutes to build up sufficient boiler pressure before the car could move off, and have to stop about every twenty miles or so to take on additional water. The battery-electric car could claim simplicity in construction, smooth running, clean operation and ease of driving. On the debit side, their operating range on fully charged batteries was originally no more than about twenty-five miles at fairly low speeds. Added to this problem was the length of time required to recharge the batteries, which typically involved an over-night charge. Although it would be only fair to add that both steam and electric powered cars were gradually improved, it was the petrol-engined car that showed the greatest potential for further development. In terms of convenience it could travel considerably longer distances on a full tank of petrol, then be quickly refilled. When Henry Ford began to mass-produce his Model T car in 1908, which brought car ownership within the financial reach of millions, the long-term future of the petrol-engined car was assured.

Although there have been many expensive attempts to revive interest in steam driven and battery-electric cars, especially during the 1973 oil crisis, these have not met with any real success. Attaining the required efficiency from a relatively small steam engine poses a considerable problem. In the case of the battery-electric car, the opinion of an early authority on motor vehicles, A. Graham Clark, is still relevant. As long ago as 1914 he wrote: 'The future of the electric car is bound up in the evolution of a light, efficient and cheap battery, and it is on this problem that the keenest intellects have been and are still engaged.' However, the role played by electricity in the more recent developments of hybrid-electric and fuel-cell vehicles will be introduced in Section 32.

31.2 MODERN REQUIREMENT FOR ALTERNATIVE FUELS

Since the oil crisis of the early 1970s research has continued into the feasibility of using alternatives to petroleum based fuels (petrol and diesel). The incentive to do this arises from two main considerations:

1 A long standing recognition that the natural fossil resources from which petroleum based fuels are derived have a finite life.
2 A more recent concern about the atmospheric build-up of carbon dioxide (CO_2) and its contribution to global warming.

The United States Department of Energy definition of an alternative fuel is, therefore, 'Alternative fuels are substantially non-petroleum and yield energy security and environmental benefits.'

31.3 CLASSIFICATION OF ALTERNATIVE FUELS

Alternative fuels for motor vehicles may be listed under the following headings:

Ethanol	Liquefied petroleum gas
Methanol	Hydrogen
Compressed natural gas	Biodiesel
Liquefied natural gas	Electricity

Where an engine has been modified to run on either a conventional or an alternative fuel, it is known as a 'bi-fuel' application. This is distinct from a 'dual-fuel' application, where an engine can run on both conventional and alternative fuels simultaneously.

Ethanol

Also referred to as ethyl alcohol or grain alcohol, ethanol is a liquid fuel derived from corn or other grain, or other agricultural products. It has long been known as an alternative fuel for motor vehicles and possesses a high octane value. It may be of interest to recall that the carburettor of the legendary Ford Model T car could be adjusted so that the engine was able to run on ethanol fuel, which was of benefit to the American farming community who could produce it from fermentation of their crops. In more recent times it has been estimated that more than four million cars run on this fuel in Brazil, where it is fermented from sugar cane. Ethanol is generally well regarded as an automotive fuel, because it promotes greater engine efficiency with lower emissions of atmospheric pollutants. A disadvantage of ethanol as compared to conventional fuels is its lower energy content by volume. In practice a larger size fuel tank is therefore required to compensate for an otherwise reduced mileage range. Ethanol fuel being used in America is a blend of 85 per cent ethanol and 15 per cent unleaded regular petrol (gasoline).

Methanol

This is another alcohol fuel and is also referred to as methyl alcohol or wood alcohol. Although methanol can be derived from a variety of feedstocks, it is now generally produced by a process using natural gas as a feedstock. Its chemical and physical characteristics are similar to those of ethanol, so as might be expected it possesses a high octane value that enhances engine performance. Methanol is also a clean burning fuel that brings benefits in reducing emissions of atmospheric pollutants. In America this property is exploited in 'fuel-flexible' vehicles, which use a blend of 85 per cent methanol and 15 per cent unleaded regular petrol. These vehicles have a sensor in the fuel line that detects the ratio of methanol to petrol in the tank. An on-board electronic control unit then instructs the engine management system to optimize the fuel injection and ignition timing accordingly. As in the case of ethanol, a larger size fuel tank is required to restore vehicle range.

Compressed natural gas

Natural gas is a mixture of hydrocarbons, the main constituent being methane. It is available either from underground or undersea gas wells, or in association with crude oil production. Natural gas will be familiar, of course, as the same fuel used for domestic cooking and heating, and there are abundant supplies in many countries. In fact it has been estimated that over one million vehicles worldwide run on natural gas with Argentina accounting for nearly half of them. The main attraction of using natural gas as a vehicle fuel is its clean burning qualities, which therefore reduce emissions of atmospheric pollutants. It also possesses a high octane value. Natural gas for vehicle applications is generally stored in a compressed state, hence the term 'Compressed Natural Gas' usually abbreviated to CNG. To increase its energy density and therefore save storage space on board vehicles, the gas is compressed to $20\,MN/m^2$ ($3000\,lbf/in^2$). For this purpose fuel storage takes the form of purpose designed pressure vessels, which are stronger than conventional fuel tanks and pressure tested to $25\,MN/m^2$ ($3750\,lbf/in^2$). In actual operation the natural gas leaving the storage pressure vessels flows, via a master shut-off valve, through a high-pressure fuel regulator. It is then injected at atmospheric pressure into a gas mixer unit, where it is mixed with air ready to enter the engine cylinders and be ignited.

Natural gas as a vehicle fuel may be stored in either a gaseous (CNG) or a liquefied (LNG) state, the basic engine operation being identical in both cases. With LNG the liquid gas is stored cyrogenically at a low temperature of $-162°C$, and requires the use of specially insulated storage vessels. This particular application is therefore better suited to commercial vehicles. An important advantage of running these vehicles on natural gas, whichever system of storage is chosen, is that their engines will be much quieter running than would be the case if they were diesels, thereby making them more compatible with overnight delivery services.

Liquid petroleum gas

Usually referred to simply as 'LPG' liquid petroleum gas mainly consists of propane, which is a product of natural gas processing and petroleum refining. In its natural state propane is a gas, but is turned into a liquid at a moderate pressure of $1120\,kN/m^2$ ($160\,lbf/in^2$). When it is drawn from its storage vessel, it changes back into a gas before being burned in the engine cylinders. Propane possesses a high octane value and is more cleanly burning than petrol, although fuel consumption and vehicle performance are generally not quite so good. LPG is used as an alternative fuel in many countries, and especially in bi-fuel conversions for passenger cars and commercial vehicles. However, it is easier and less expensive to convert a petrol-engined vehicle than it is a diesel-engined one. The conversion has to be carried out in accordance with national codes of practice to ensure safe operation. For passenger cars the LPG is stored in a robust doughnut-shaped pressure vessel that is intended to take the place of the spare wheel, which must then be secured in the boot space. In a modern AG Autogas System for petrol engine conversions, the liquid gas flows from the storage vessel and is vaporized in an underbonnet vaporizer/pressure regulator unit. This serves to stabilize the pressure of the gas before it enters a common rail, where it is fed to intake manifold gas injectors in a similar manner to that of the petrol injection system. The whole injection process is controlled by an additional engine management system.

Hydrogen

Discovered in 1766 by the scientist Henry Cavendish, who termed it ‘inflammable air’, hydrogen is the most abundant, lightest and simplest of the chemical elements. It may be produced either by electrolysis using electrical energy to split water molecules into hydrogen and oxygen, or by synthesis gas production using steam reforming of natural gas. Strictly speaking, hydrogen cannot be considered as a fuel, but rather as a transmitter of energy. For this reason it is playing an important role in the development of fuel cell powered vehicles (Section 32.7), but it can also be utilized as a gaseous fuel by injecting it into the intake manifold of conventional engines. In the latter application it has the immediate attraction that the exhaust gases are free from carbon dioxide, although small amounts of nitrogen oxides, unburned hydrocarbons and carbon monoxide will be emitted from the presence of engine lubrication. In comparison to a petrol fuelled engine, there is some reduction in power and a tendency towards backfiring into the intake manifold. The low density of hydrogen also poses a problem in its on-board storage, because for an equivalent energy content it demands considerably more space than that required for conventional fuels. To counter this problem the hydrogen may be stored either cyrogenically at the extremely low temperature of $-253°C$ when it turns into a liquid, or as a gas at the very high pressure of $30\,MN/m^2$ ($4500\,lbf/in^2$). In both cases there is a weight penalty imposed by the special on-board storage facilities required.

Biodiesel

This fuel is typically blended with standard diesel fuel. It is derived from natural sources such as vegetable oils, which are chemically reacted with an alcohol to produce compounds known as esters. Diesel fuel that results from this process is termed ‘biodiesel’. Although this fuel has similar physical properties and offers comparable performance to standard diesel fuel, it demonstrates important advantages in reducing emissions of atmospheric pollutants. These include unburned hydrocarbons, carbon monoxide and particulates. Another advantage claimed for biodiesel is that it possesses better lubricity than standard diesel fuel. A possible disadvantage of biodiesel is that when used in high-proportion blends or pure, certain types of elastomer that may be used in the vehicle fuel system can be subject to long-term degradation. However, in Britain a modest 5 per cent biodiesel and 95 per cent standard diesel fuel mix is generally favoured.

Electricity

The alternative fuels mentioned so far contain stored chemical energy, which when released through combustion in the engine cylinders provides mechanical power. In contrast the use of electricity as an alternative fuel allows a direct conversion to mechanical power when supplied to an electric motor. Electricity used to power vehicles of the battery-electric type (Section 32.2) is derived from on-board multiple batteries, which act as energy storage devices. These batteries must, of course, be initially charged and subsequently recharged from electricity produced at power stations. This electricity is then transmitted to substations, stepped down to usable lower voltages and distributed for domestic and commercial use. It is therefore from these sources that the batteries of electric vehicles may be recharged, usually overnight when demands on the electrical supply system are least and cost incentives may be offered. The most commendable advantage of operating a battery-electric vehicle is the contribution it makes to cleaner air, because it does not emit any polluting gases. Other advantages relate partly to the lower cost of using electricity as an alternative source of power, and partly to a reduction in maintenance requirements arising from simpler vehicle construction.

To conclude this introduction to vehicle alternative fuels, it must be recognized that accurate comparisons of their relative merits at point of use may not necessarily be all-inclusive. For example, hydrogen is generally regarded as an attractive non-polluting fuel, which can be stored and distributed through pipelines, but it becomes less attractive if the electricity used to produce the hydrogen is generated by burning fossil fuels. In the final analysis the viability of using an alternative fuel relies on it having widespread availability, together with safe and convenient-to-use refuelling or recharging facilities.

32 Battery-electric, hybrid and fuel-cell vehicles

32.1 GENERAL BACKGROUND

The purpose of developing battery-electric, hybrid and fuel-cell sources of power generation is to produce environmentally friendly vehicles. These modern developments are sometimes referred to as 'Clean vehicle technology' and are being vigorously pursued by motor manufacturers. Indeed, at the 10th North American International Auto Show in 1998, John Smith, Chairman of General Motors, was reported as saying 'No car company will be able to survive in the 21st century if it relies solely on internal combustion engines.' However, since clean vehicle technology is in some respects a new and still emerging branch of automotive engineering, the following is intended only as an introduction to the subject.

32.2 BATTERY-ELECTRIC VEHICLES

A battery-electric vehicle or 'BEV' may be defined as one that operates solely on the power provided by on-board rechargeable batteries, which energize an electric motor to drive the wheels. In clean vehicle technology it qualifies as a 'zero emission vehicle', usually abbreviated to 'ZEV', since it emits no pollutants into the atmosphere.

The limitations of early battery-electric passenger cars and the later attempts to revive interest in them, have previously been mentioned (Section 31.1). However, this is not to deny the successful use of battery-electric vehicles for other purposes, such as transporting goods within commercial premises, or urban door-to-door deliveries of milk and bread. In the latter case limitations in their operating range do not represent a serious disadvantage, and for this reason they are sometimes termed 'homing' vehicles.

32.3 LAYOUT OF BATTERY-ELECTRIC VEHICLES

The main units to be accommodated in battery-electric vehicles are a battery pack, electric motor and system controller. Their actual disposition on the vehicle depends on whether it is the front or the rear wheels that are being driven. The former arrangement generally applies to small cars and vans. In this case the electric motor together with a typically 10:1 ratio reduction gear forms a transaxle to drive the wheels (Figure 32.1), similar to the arrangement found with conventional power units. For specialized battery-electric vehicles, such as those used for example in industry, leisure and airports, the electric motor and reduction gear may be integrated with a live rear axle (Figure 32.2). This type of drive is used in the well-known specialist battery-electric vehicles produced by Bradshaw.

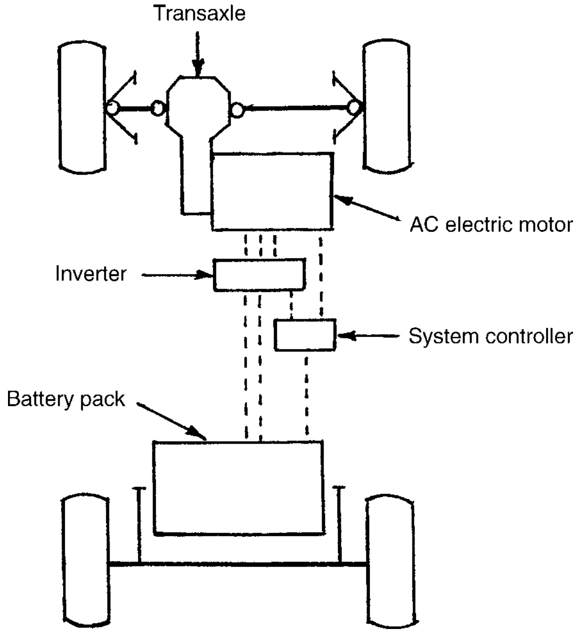


Figure 32.1 Simplified layout of battery-electric vehicle with front-wheel drive

32.4 BASIC UNITS OF BATTERY-ELECTRIC VEHICLES

Battery pack

This unit represents the on-board energy storage system for the vehicle. It comprises an interconnected group of batteries that are treated as a single unit. Since the battery pack is both heavy and bulky it is usually carried low down in the vehicle, typically on underfloor trays.

The three types of battery used in battery-electric vehicles may be categorized as lead-acid, nickel-based and lithium-based. The lead-acid battery will, of course, be most familiar as the ordinary car battery, which is used for starting, ignition, lighting and other duties. For application to the battery-electric vehicle, the lead-acid battery has to be provided with thicker plates to withstand repetitive charge and discharge cycling. These batteries are generally of the maintenance-free type for road vehicles, utilizing a gel rather than a liquid electrolyte that needs periodic topping up. Regarding their performance the lead-acid type yields the least of what is termed 'energy density', which relates battery weight or volume (and hence bulkiness) to the amount of energy stored. Nickel-based

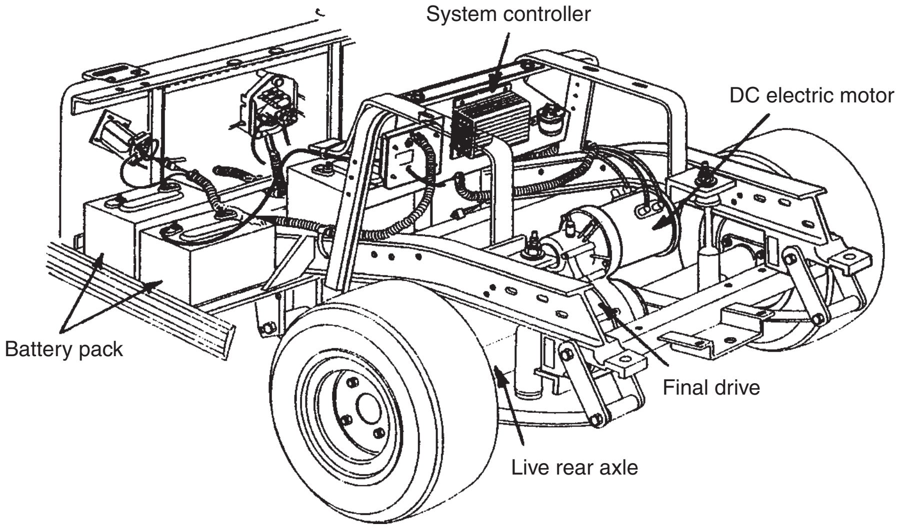


Figure 32.2 Arrangement of battery-electric specialist vehicle (*Bradshaw*)

batteries of the nickel-cadmium and nickel-metal hydride types possess in the region of 50 and 100 per cent greater energy densities respectively, and may also be recharged at a faster rate. Unlike a lead-acid battery that has lead metal and lead dioxide plates, the nickel-cadmium battery has nickel hydroxide and cadmium plates and utilizes an alkali-based electrolyte. Although similar to the nickel-cadmium battery, a special metal alloy replaces the toxic cadmium in the nickel-metal hydride battery. On the debit side, nickel-based batteries are significantly more expensive than lead-acid ones. Finally, lithium-based batteries are being developed for traction applications, following their established use in consumer electrical products such as camcorders. Their chemistry is more complex than that of the previously mentioned types of battery, and a possible disadvantage is the requirement for a battery protection system to prevent overcharging. However, a lithium-based battery can yield about 150 per cent greater energy density than a lead-acid battery and also has a potentially longer life.

Electric motor

The purpose of the electric motor is, of course, to convert electrical energy received from the battery pack into mechanical energy that drives the wheels. Since the current that flows from the battery pack is direct current (DC), it can either be utilized directly to drive a DC motor, or be converted into an alternating current (AC) by what is termed an 'inverter' to drive an AC motor.

Both types of motor, which may have various electrical configurations, can be used to drive electric vehicles. Each has its own advantages and disadvantages, but neither possesses significant advantages over the other. In purely general terms, a DC motor is typically selected for low-speed vehicles, such as specialized carriers; whereas an AC motor now tends to be favoured for on-road vehicles. Technically, a DC motor requires only a simple control system that reduces cost, but it does have a maintenance requirement in respect of its commutator and brushes, although only after extended service. An AC motor operates with continuous current reversal and therefore does not need a commutator and brushes. This leads to a simpler and lighter construction that only involves a stator and rotor, which accounts for its higher power-to-weight ratio. Although less expensive in itself and requiring virtually no maintenance, an AC motor does demand a more complex and expensive control system.

System controller

Also known as a power converter, the purpose of this unit is to control the flow of electrical energy between the battery pack and the electric motor. It therefore provides the means by which the vehicle speed can be controlled by the driver. Since a battery-electric vehicle must be bi-directional and does not have a reverse gear, the system controller must enable the electric motor to alter its direction of rotation so that the vehicle can be reversed. Another duty performed by the system controller is to provide what is termed 'regenerative braking', which can increase the operating range of the vehicle within the region of 10 to 15 per cent. In effect the electric motor is temporarily made to act as a generator, so that some of the kinetic energy of the slowing vehicle is converted into electrical energy, which can be stored by the battery pack. During this process the motor also imposes a braking effect on the vehicle, independent of its main braking system, thereby reducing brake wear.

Early system controllers for DC motors originally utilized either hand- or foot- operated contactors. In combination with a resistance grid, these selected various circuit arrangements for the battery pack and motor windings, which enabled the speed of the motor and hence that of the vehicle to be controlled in steps. In contrast, a modern system controller is an altogether more complex electronic unit. It is provided with comprehensive feedback signals relating to battery pack and motor operating conditions and can, for instance, limit motor output in the event of overheating. As regards controlling

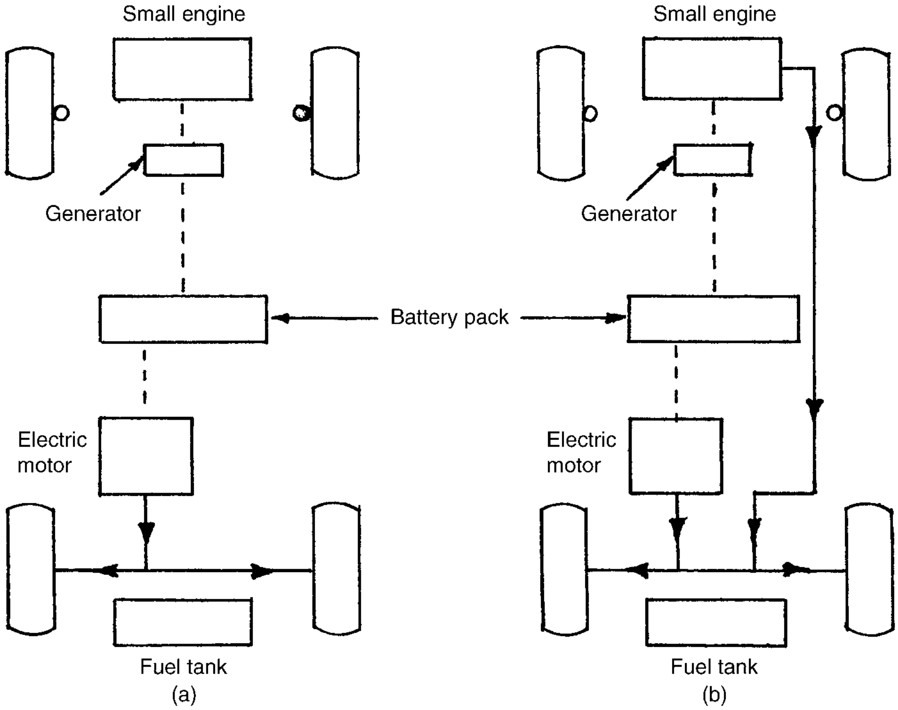


Figure 32.3 Simplified comparison between series and parallel layouts for hybrid-electric vehicles: (a) series (b) parallel

vehicle speed, when the equivalent of a conventional accelerator pedal is depressed, a microprocessor receives an increasing signal voltage, which then commands the control system to increase the available current at the motor, to raise its speed and hence that of the vehicle. Similarly, when a range selector is moved to the reverse position, the control system is commanded to alter the direction of current flow at the motor to reverse the vehicle.

Caution When servicing battery-electric vehicles it is vitally important to follow the safety precautions recommended by their manufacturers, who normally advise that only specially trained technicians should repair vehicles of this type. Truly it has been said that the battery pack of an electric vehicle should be treated with the same caution and respect as a full fuel tank of a conventionally powered vehicle.

32.5 HYBRID-ELECTRIC VEHICLES

A hybrid-electric vehicle or 'HEV' may be defined as one that combines two sources of power, one of which is electricity. In automotive practice, this means combining the engine and fuel supply of a conventional vehicle with the electric motor and battery pack of an electric vehicle, via a common drive train. The efficiency advantages claimed for hybrid-electric vehicles lie in their ability to offer a better range and performance than battery-electric vehicles, whilst using less fuel and producing fewer emissions than a conventionally powered vehicle.

Although the hybrid-electric vehicle concept received continuous development following the oil crisis of 1973, it was not until the late 1990s that the Japanese companies Toyota and Honda introduced their production Prius and Insight models. The hybrid-electric vehicle actually had much earlier origins, because before World War I the British companies Tilling-Stevens and Daimler both produced what were then known as 'petrol-electric' buses. In the case of the Daimler bus there were two separate petrol engines and electric dynamotors, each pair being coupled together and carried on either side of the bus, providing individual drives to the rear wheels. When the bus was running light the two engines drove the wheels, while the dynamotor acted as a dynamo to charge the batteries. Under heavy load running the batteries automatically supplied electrical energy to the dynamotors, so that they acted as electric motors to increase the power available at the wheels.

32.6 LAYOUT OF HYBRID-ELECTRIC VEHICLES

There are two basic configurations for hybrid-electric vehicles, these being classified as 'series' and 'parallel'.

In a series hybrid layout (Figure 32.3a) a small conventional engine is directly coupled to a generator and cuts in only when required to recharge a battery pack. This supplies energy to an electric motor that drives the wheels, so that when the engine is not running the vehicle operates in zero-emissions mode. When the battery pack state of charge falls below about three-quarters full charge, the engine cuts in so that the generator can recharge the batteries. As the engine is not directly influenced by the performance demands of the vehicle, it can operate within a narrower and more efficient range of speeds, and since it is never required to idle there is a reduction in overall emissions.

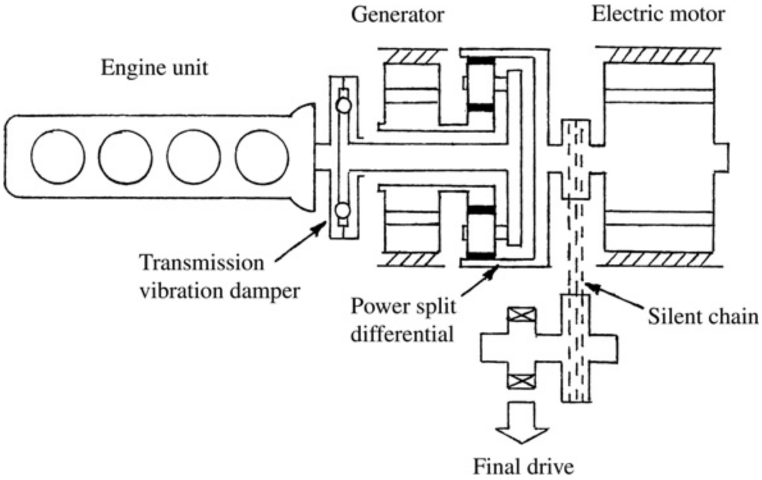


Figure 32.4 Schematic arrangement of a parallel hybrid vehicle with a power split epicyclic differential gearset

In contrast a parallel hybrid layout (Figure 32.3b) provides for either, or both, a small conventional engine and an electric motor to drive the wheels. That is, a mechanical connection is established between not only the electric motor, but also the engine, and the driven wheels. This connection may take the form of an epicyclic differential gear to apportion the power flow from the two energy sources (Figure 32.4). A battery pack is still required, of course, but may be of smaller capacity. Similarly, a medium rather than a heavy-duty electric motor can be used. With a parallel-hybrid the electric motor assists the conventional engine during start-up, acceleration for overtaking and hill climbing. Conversely, at low speed, overrunning down-hill or when standing still, the engine can be switched off altogether. This avoids inefficient operation of the engine and relatively high emissions. As in the case of series hybrid vehicles, the electric motor can also act as a generator when it is being driven by the wheels, thereby providing overrun regenerative braking.

32.7 BASIC UNITS OF HYBRID-ELECTRIC VEHICLES

Engine

The engines used in hybrid-electric vehicles are of either the petrol or the diesel types. For passenger car applications, in-line three- or four-cylinder petrol engines have so far been chosen. These are relatively smaller and lighter than found in conventional cars of equivalent size, because the engine is subject only to average rather than peak loading. The engines are designed for maximum efficiency and embody modern established technical features, such as variable valve timing and direct injection. A transverse engine mounting is adopted as for conventional powered front-wheel drive cars. For commercial vehicle application such as buses, in-line six-cylinder diesel engines are favoured for their high thermal efficiency. These engines feature modern high-pressure, direct-injection technology and are mounted transversely at the rear of the chassis. A fuel tank is, of course, required to supply the engine.

Battery pack

The battery pack, which is smaller than that required for battery-electric vehicles, serves as the energy storage device for the electric motor. High-power battery technologies are therefore being developed for hybrid-electric vehicles. The actual types of battery used are similar to those previously discussed for modern battery-electric vehicles (Section 32.4), nickel-metal hydride and lithium-based batteries currently being favoured. Unlike a battery-electric vehicle, the battery pack of a hybrid-electric vehicle does not require external recharging, because it is maintained in a constant state of charge by output from the engine and by regenerative braking.

Electric motor

The basic units of a hybrid-electric passenger car are necessarily designed to occupy little more underbonnet space than would be the case with a conventionally engined car. A compact and high-torque capacity electric motor is therefore required, which is either wholly or partly responsible for driving the wheels, according to whether a series or a parallel hybrid layout is used. A permanent magnet synchronous type of electric motor is typically employed for this purpose.

System controller

The hybrid-electric vehicle requires an altogether more complex system controller than a battery-electric vehicle. In fact it is the key to the successful operation of this type of vehicle. It is responsible for the electronic control of power flow between the mechanical and electrical elements, so that energy consumption is optimized during all aspects of vehicle operation. The system controller is therefore sometimes referred to as an Integrated Power Module (IPM). More specifically, it exercises control over the power flow into and out of the electric motor, the power output of the engine, the storage of electrical energy and that recovered from regenerative braking, and engine starting procedure. The size, weight and heat dissipation of the module must all be optimized and its reliability demonstrated in extended service.

32.8 FUEL-CELL VEHICLES

Although the purpose of this book has been to concentrate on present-day vehicle technology, it nevertheless seems appropriate at least to introduce the concept of the fuel-cell vehicle. Also known as a fuel-cell electric vehicle or 'FCEV', this type of vehicle is now attracting an increasing amount of public attention.

The basic idea of a fuel cell that directly converts the chemical energy of a fuel into electrical energy was discovered in 1838 by a British lawyer turned scientist called William Grove, who described his invention as a 'gas battery'. Other scientists continued to research the fuel cell, notably Friedrich Otwald, who in 1893 provided the first convincing theoretical explanation of its working. During World War II Thomas Bacon developed a fuel cell for use in Royal Navy submarines, which was later pursued by the Pratt & Whitney company in America to provide capsule power for the Apollo spacecraft. Since the late 1990s major motor manufacturers in America, Europe and Japan have been investigating the fuel cell as a potential power source for vehicles.

32.9 OPERATION AND TYPES OF FUEL CELL

A fuel cell is essentially another form of battery, which, so long as it receives an external supply of fuel, will produce a continuous flow of electrical energy to generate power. Therefore unlike the operation of a storage battery, a fuel cell will never run down or need recharging. In other words, a fuel cell only produces electricity while fuel is being supplied to it.

In practice a fuel cell basically comprises two electrodes with an electrolyte contained between them. One electrode is positive and the other one negative, these being termed the cathode and anode respectively as in any other electrolytic cell. The function of the electrolyte is to transport the electrically charged particles from one electrode to the other, the reactions at the electrodes being accelerated by catalysts. In simple terms, hydrogen fuel is supplied to the anode of the fuel cell and oxygen from the air enters through the cathode. At the anode the hydrogen atoms are split by the catalyst into protons and electrons, that is positive and negative charged particles, which then pursue different paths. The hydrogen protons travel through the electrolyte to the cathode. Simultaneously, the hydrogen electrons create a separate circuit to provide usable electrical energy, as they too travel to the cathode. Here the oxygen and the hydrogen protons and electrons combine on a catalyst simply to form water (Figure 32.5). To increase the output of electrical energy sufficient to drive a vehicle, fuel cells are combined into groups known as 'stacks'. In a General Motors prototype passenger car a stack of 200 fuel cells is used.

There are several types of fuel cell under development, these being classified according to the kind of electrolyte they use. This in turn determines the chemical reactions that occur in the cell, its catalyst specification, fuel required, and cell operating temperature. Each type of cell naturally has its own advantages and disadvantages, but for motor vehicle applications the 'Polymer Electrolyte Membrane' or 'Proton Exchange Membrane', usually abbreviated to PEM, type of fuel cell would seem to be regarded as the most suitable. This choice is based on its favourable power-to-weight ratio and a relatively low operating temperature of about 80°C, the latter reducing warm-up time for quicker starting. In construction the PEM fuel cell employs a solid polymer as an electrolyte, thereby avoiding the use of corrosive fluids, and its porous carbon electrodes contain a platinum catalyst. The fuel required is pure hydrogen that poses certain onboard storage and supply problems, as previously mentioned in Section 31.4. Alternatively, other fuels may be used but these require the

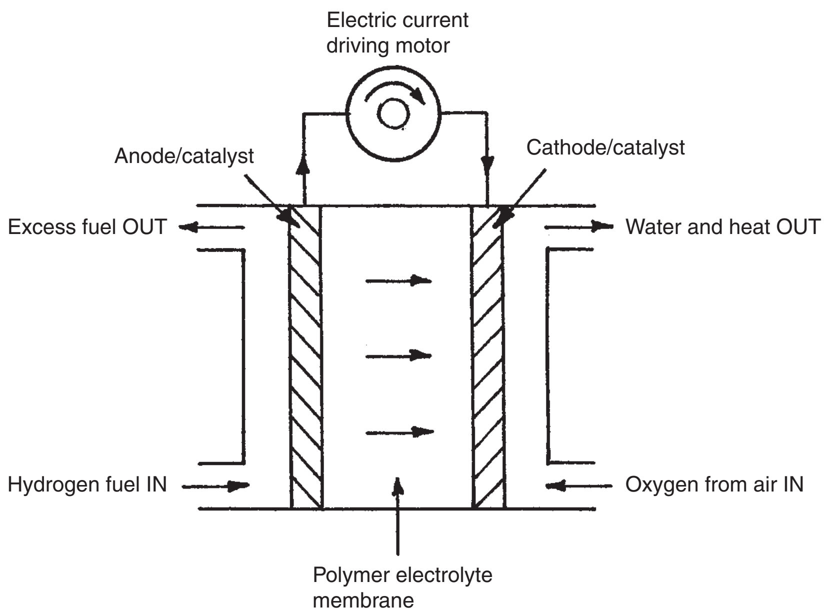


Figure 32.5 Simplified layout of a fuel-cell vehicle with front-wheel drive

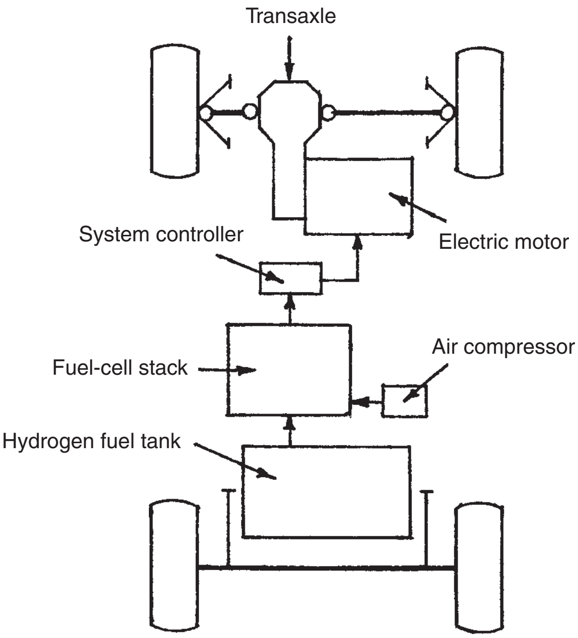


Figure 32.6 Simplified layout of a fuel-cell vehicle with front-wheel drive

addition of an onboard reformer to release the hydrogen contained in them.

32.10 LAYOUT OF FUEL-CELL VEHICLES

So far fuel-cell vehicles exist mainly in prototype forms (Figure 32.6). These are being evaluated for performance and reliability in controlled environments selected by their manufacturers, such as for airport transportation and transit bus operation. For passenger car fuel-cell application, it would seem likely that some departure from the present-day conventional layout may ultimately be required, as evidenced by the recent General Motors 'AUTOnomy' prototype vehicle. This features a 'skateboard' chassis construction, which not only allows interchangeable body styles, but also makes provision for the electric motor drive, fuel-cell stack, hydrogen fuel tanks and drive-by-wire electronic controls to be accommodated low down in the chassis framework, thereby lowering the vehicle centre of gravity for improved ride, handling and stability.

Index