

Wind Energy Handbook

Wind Energy Handbook

Second Edition

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About the Authors

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Nick Jenkins was at the University of Manchester (UMIST) from 1992 to 2008. He then moved to Cardiff University where he is now Professor of Renewable Energy. His previous career had included 14 years industrial experience, of which five years were in developing countries. He is a Fellow of the IET, IEEE and Royal Academy of Engineering and for three years was the Shimizu Visiting Professor at Stanford University.

David Sharpe has worked in the aircraft industry for the British Aircraft Corporation as a structural engineer. From 1969 to 1995 he was a Senior Lecturer in aeronautical engineering at Kingston Polytechnic and at Queen Mary College, University of London. Between 1996 and 2003 he was at Loughborough University as a Senior Research Fellow at the Centre for Renewable Energy Systems Technology. David is a member of the Royal Aeronautical Society and was a member of the British Wind Energy Association at its inception. He has been active in wind turbine aerodynamics research since 1976.

Ervin Bossanyi: After graduating in theoretical physics and completing a PhD in energy economics at Cambridge University Ervin Bossanyi has been working in wind energy since 1978. He was a research fellow at Reading University and then Rutherford Appleton Laboratory before moving into industry in 1986 where he worked on advanced control methods for the Wind Energy Group. Since 1994 he has been with international consultants Garrad Hassan where he is a principal engineer.

Preface to Second Edition

The second edition of the *Wind Energy Handbook* seeks to reflect the evolution of design rules and the principal innovations in the technology that have taken place in the ten years since the first edition was published. A major new direction in wind energy development in this period has been the expansion offshore and so the opportunity has been taken to add a new chapter on offshore wind turbines and wind farms.

The offshore chapter begins with a survey of the present state of offshore wind farm development, before consideration of resource assessment and array losses. Then wave loading on support structures is examined in depth, including a summary of the combinations of wind and wave loading specified in the load cases of the IEC standard and descriptions of applicable wave theories. Linear (Airy) wave theory and Dean stream function theory are explained, together with their translation into wave loadings by means of Morison's equation. Diffraction and breaking wave theories are also covered.

Consideration of wave loading leads to a survey of the different types of support structure deployed to date. Monopile, gravity bases, jacket structures, tripods and tripiles are described in turn. In view of their popularity, monopiles are accorded the most space and, after an outline of the key design considerations, monopile fatigue analysis in the frequency domain is explained.

Another major cost element offshore is the undersea cable system needed to transmit power to land. This subject is considered in depth in the section on the power collection and transmission cable network. Machine reliability is also of much greater importance offshore, so developments in turbine condition monitoring and other means of increasing reliability are discussed. The chapter is completed by sections covering the assessment of environmental impacts, maintenance and access, and optimum machine size.

The existing chapters in the first edition have all been revised and brought up to date, with the addition of new material in some areas. The main changes are as follows:

Chapter 1: Introduction This chapter has been brought up to date and expanded.

Chapter 2: The wind resource Descriptions of the high frequency asymptotic behaviour of turbulence spectra and the Mann turbulence model have been added.

Chapters 3 and 4: Aerodynamics of horizontal axis wind turbines The contents of Chapters 3 and 4 of the first edition have been rearranged so that the fundamentals are covered in Chapter 3 and more advanced subjects are explored in Chapter 4. Some material on field-testing and performance measurement has been omitted to make

space for a survey of wind turbine aerofoils and new sections on dynamic stall and computational fluid dynamics.

Chapter 5: Design loads for horizontal axis wind turbines The description of IEC load cases has been brought up to date and a new section on the extrapolation of extreme loads from simulations added. The size of the ‘example’ wind turbine has been doubled to 80 m, in order to be more representative of the current generation of turbines.

Chapter 6: Conceptual design of horizontal axis wind turbines The initial sections on choice of machine size, rating and number of blades have been substantially revised, making use of the NREL cost model. Variable speed operation is considered in greater depth. The section on tower stiffness has been expanded to compare tower excitation at rotational frequency and blade passing frequency.

Chapter 7: Component design New rules for designing towers against buckling are described and a section on foundation rotational stiffness has been added.

Chapter 8: The Controller Individual blade pitch control is examined in greater depth.

Chapter 9: Wind turbine installations and wind farms A survey of recent research on the impact of turbines on birds has been added.

Chapter 10: Electrical systems New sections covering (a) Grid Code requirements for the connection of large wind farms to transmission networks and (b) the impact of wind farms on generation systems have been added.

Acknowledgements for First Edition

A large number of individuals have assisted the authors in a variety of ways in the preparation of this work. In particular, however, we would like to thank David Infield for providing some of the content of Chapter 4, David Quarton for scrutinising and commenting on Chapter 5, Mark Hancock, Martin Ansell and Colin Anderson for supplying information and guidance on blade material properties reported in Chapter 7, and Ray Hicks for insights into gear design. Thanks are also due to Roger Haines and Steve Gilkes for illuminating discussions on yaw drive design and braking philosophy, respectively, and to James Shawler for assistance and discussions about Chapter 3.

We have made extensive use of ETSU and Risø publications and record our thanks to these organisations for making documents available to us free of charge and sanctioning the reproduction of some of the material therein.

While acknowledging the help we have received from the organisations and individuals referred to above, the responsibility for the work is ours alone, so corrections and/or constructive criticisms would be welcome.

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Acknowledgements for Second Edition

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In addition, several individuals took on the onerous task of scrutinising sections of the draft text. The authors are particularly grateful to Tim Camp for examining the sections on design loading, on- and offshore, Colin Morgan for providing useful comments on the sections dealing with support structures and Graeme McCann for vetting sections on the extrapolation of extreme loads from simulations and monopile fatigue analysis in the frequency domain. Nevertheless, responsibility for any errors remains with the authors. (In this connection, thanks are due to those who have pointed out errors in the first edition).

Tony Burton would also like to record his thanks to Martin Kuhn and Wim Bierbooms for providing copies of their PhD theses – entitled respectively ‘Dynamics and design optimisation of offshore wind energy conversion systems’ and ‘Constrained stochastic simulation of wind gusts for wind turbine design’ – both of which proved invaluable in the preparation of this work.

List of Symbols

Note: This list is not exhaustive, and omits many symbols that are unique to particular chapters.

a	axial flow induction factor; flange projection beyond bolt centre
a'	tangential flow induction factor
a'_t	tangential flow induction factor at the blade tip
a_0	two-dimensional lift curve slope, $(dC_l/d\alpha)$
a_1	constant defining magnitude of structural damping
A, A_D	rotor swept area
A_∞, A_w	upstream and downstream stream-tube cross-sectional areas
A_c	Charnock's constant
b	face width of gear teeth; eccentricity of bolt to tower wall in bolted flange joint
b_r	unbiased estimator of β_r
B	Number of blades
c	blade chord; Weibull scale parameter; dispersion of distribution; flat plate half width; half of cylinder immersed width
c^*	half of cylinder immersed width at time t^*
$\hat{c}$	damping coefficient per unit length
c_i	generalised damping coefficient with respect to the i th mode
C	decay constant; wave celerity, L/T ; constrained wave crest elevation
$C(v)$	Theodorsen's function, where v is the reduced frequency: $C(v) = F(v) + iG(v)$
C_d	sectional drag coefficient
C_D	drag coefficient in Morison's equation
C_{DS}	steady flow drag coefficient in Morison's equation
C_f	sectional force coefficient (i.e. C_d or C_l as appropriate)
C_l, C_L	sectional lift coefficient
C_M	inertia coefficient in Morison's equation
C_n^m	coefficient of a Kinner pressure distribution
C_p	pressure coefficient
C_P	power coefficient
C_Q	torque coefficient
C_T	thrust coefficient; total cost of wind turbine

C_{TB}	total cost of baseline wind turbine
C_x	coefficient of sectional blade element force normal to the rotor plane
C_y	coefficient of sectional blade element force parallel to the rotor plane
$C(\Delta r, n)$	coherence – i.e. normalised cross spectrum – for wind speed fluctuations at points separated by distance Δr measured in the across wind direction
$C_{jk}(n)$	coherence – i.e. normalised cross spectrum – for longitudinal wind speed fluctuations at points j and k
d	streamwise distance between vortex sheets in a wake; water depth
d_1	pitch diameter of pinion gear
d_{PL}	pitch diameter of planet gear
D	drag force; tower diameter; rotor diameter; flexural rigidity of plate; constrained wave trough elevation
E	energy capture, i.e. energy generated by turbine over defined time period; modulus of elasticity
$E\{\}$	time averaged value of expression within brackets
$E[H_s \bar{U}]$	expected value of significant wave height conditional on a hub-height mean wind speed $\bar{U}$
f	tip loss factor; Coriolis parameter; wave frequency; source intensity
$f()$	probability density function
$f_1(t)$	support structure first mode hub displacement
$f_j(t)$	blade tip displacement in j th mode
$f_{in}(t)$	blade tip displacement in i th mode at the end of the n th time step
$f_J(t)$	blade j first mode tip displacement
f_p	wave frequency corresponding to peak spectral density
$f_T(t)$	hub displacement for tower first mode
F	force; force per unit length
F_X	load in x (downwind) direction
F_Y	load in y direction
F_t	force between gear teeth at right angles to the line joining the gear centres
$F(\mu)$	function determining the radial distribution of induced velocity normal to the plane of the rotor
$F()$	cumulative probability distribution function
$F(x U_k)$	cumulative probability distribution function for variable x conditional on $U = U_k$
g	acceleration due to gravity; vortex sheet strength; peak factor, defined as the number of standard deviations of a variable to be added to the mean to obtain the extreme value in a particular exposure period, for zero-up-crossing frequency, ν
g_0	peak factor as above, but for zero upcrossing frequency n_0
G	geostrophic wind speed; shear modulus; gearbox ratio
$G(f)$	transfer function divided by dynamic magnification ratio
$G(t)$	t second gust factor
h	height of atmospheric boundary layer; duration of time step; thickness of thin-walled panel; maximum height of single gear tooth contact above critical root section
H	hub height; wave height; hub height above mean sea level

H_1	1 year extreme wave height
H_{50}	50 year extreme wave height
H_{jk}	elements of transformational matrix, $\mathbf{H}$, used in wind simulation
$H_i(n)$	complex frequency response function for the i th mode
$H(f)$	frequency dependant transfer function
H_s	significant wave height
H_{s1}	1 year extreme significant wave height based on 3 hour reference period
H_{s50}	50 year extreme significant wave height based on 3 hour reference period
H_B	breaking wave height
I	turbulence intensity; second moment of area; moment of inertia; electrical current (shown in bold when complex)
I_B	blade inertia about root
I_0	ambient turbulence intensity
I_{15}	expected value of hub height turbulence intensity at reference mean wind speed of 15 m/s
I_+	added turbulence intensity
I_{++}	added turbulence intensity above hub height
I_R	inertia of rotor about horizontal axis in its plane
I_u	longitudinal turbulence intensity
I_v	lateral turbulence intensity
I_w	vertical turbulence intensity
I_{wake}	total wake turbulence intensity
j	$\sqrt{-1}$
k	shape parameter for Weibull function; shape parameter for GEV distribution; integer; reduced frequency, $(\omega c/2W)$; wave number, $2\pi/L$; surface roughness
k_i	generalised stiffness with respect to the i th mode, defined as $m_i \omega_i^2$
k_1, k_2	marine conditions reference period conversion factors
K	constant on right hand side of Bernoulli equation
K_C	Keulegan-Carpenter number
K_P	power coefficient based on tip speed
K_{SMB}	size reduction factor accounting for the lack of correlation of wind fluctuations over structural element or elements
$K_{Sx}(n_1)$	size reduction factor accounting for the lack of correlation of wind fluctuations at resonant frequency over structural element or elements
$K_v()$	modified Bessel function of the second kind and order v
$K(\chi)$	function determining the induced velocity normal to the plane of a yawed rotor
L	length scale for turbulence (subscripts and superscripts according to context); lift force; wave length
L_u^x	integral length scale for the along wind turbulence component, u , measured in the longitudinal direction, x
m	mass per unit length, integer; depth below seabed of effective monopole fixity; inverse slope of log-log plot of S-N curve
m_i	generalised mass with respect to the i th mode
m_{T1}	generalised mass of tower, nacelle and rotor with respect to tower first mode
M	moment; integer; tower top mass
$\overline{M}$	mean bending moment

M_0	peak quasi-static mudline moment
$M_1(t)$	fluctuating cantilever root bending moment due to excitation of first mode
M_T	teeter moment
M_X	blade in-plane moment (i.e. moment causing bending in plane of rotation); tower side-to-side moment
M_Y	blade out-of-plane moment (i.e. moment causing bending out of plane of rotation); tower fore-aft moment
M_Z	blade torsional moment; tower torsional moment
M_{YS}	low-speed shaft moment about rotating axis perpendicular to axis of blade 1
M_{ZS}	low-speed shaft moment about rotating axis parallel to axis of blade 1
M_{YN}	moment exerted by low-speed shaft on nacelle about (horizontal) y-axis
M_{ZN}	moment exerted by low-speed shaft on nacelle about (vertical) z-axis
n	frequency (Hz); number of fatigue loading cycles; integer; distance measured normal to a surface
n_0	zero up-crossing frequency of quasistatic response
n_1	frequency (Hz) of 1st mode of vibration
N	number of blades; number of time steps per revolution; integer; design fatigue life in number of cycles for a given constant stress range
$N(r)$	centrifugal force
$N(S)$	number of fatigue cycles to failure at stress level S
p	static pressure
$p()$	probability density function
P	aerodynamic power; electrical real (active) power
$P_n^m()$	associated Legendre polynomial of the first kind
$q(r, t)$	fluctuating aerodynamic lift per unit length
Q	rotor torque; electrical reactive power
Q_a	aerodynamic torque
$\dot{Q}$	rate of heat flow
$\bar{Q}$	mean aerodynamic lift per unit length
Q_D	dynamic factor defined as ratio of extreme moment to gust quasistatic moment
Q_g	load torque at generator
Q_L	loss torque
$Q_n^m()$	associated Legendre polynomial of the second kind
$Q_1(t)$	generalised load, defined in relation to a cantilever blade by Equation (A5.13)
r	radius of blade element or point on blade; correlation coefficient between power and wind speed; radius of tubular tower; radius of monopile
r'	radius of point on blade
r_1, r_2	radii of points on blade or blades
R	blade tip radius; ratio of minimum to maximum stress in fatigue load cycle; electrical resistance
Re	Reynold's number
$R_u(n)$	normalised power spectral density, $n.S_u(n)/\sigma_u^2$, of longitudinal wind-speed fluctuations, u , at a fixed point
s	distance inboard from the blade tip; distance along the blade chord from the leading edge; separation between two points; Laplace operator; slip of induction machine
s_1	separation between two points measured in the along-wind direction

S	wing area; autogyro disc area; fatigue stress range; surface area
$\mathbf{S}$	electrical complex (apparent) power (bold indicates a complex quantity)
$S()$	uncertainty or error band
$S_{jk}(n)$	cross spectrum of longitudinal wind-speed fluctuations, u , at points j and k (single-sided)
$S_M(n)$	single-sided power spectrum of bending moment
$S_{Q1}(n)$	single-sided power spectrum of generalised load
$S_u(n)$	single-sided power spectrum of longitudinal wind-speed fluctuations, u , at a fixed point
$S_u^0(n)$	single-sided power spectrum of longitudinal wind-speed fluctuations, u , as seen by a point on a rotating blade (also known as rotationally sampled spectrum)
$S_u^0(r_1, r_2, n)$	cross spectrum of longitudinal wind-speed fluctuations, u , as seen by points at radii r_1 and r_2 on a rotating blade or rotor (single-sided)
$S_v(n)$	single-sided power spectrum of lateral wind speed fluctuations, v , at a fixed point
$S_w(n)$	single-sided power spectrum of vertical wind-speed fluctuations, w , at a fixed point
$S_{\eta\eta}(n)$	single-sided power spectrum of sea surface elevation
t	time; gear tooth thickness at critical root section; tower wall thickness; monopole wall thickness
T	rotor thrust; duration of discrete gust; wind-speed averaging period; wave period for regular waves
T_c	mean period between wave crests
T_p	peak wave period, $1/f_p$
T_z	mean zero crossing wave period
u	fluctuating component of wind speed in the x -direction; induced velocity in x -direction; in-plane plate deflection in x -direction; gear ratio; water particle velocity in x -direction
u^*	friction velocity in boundary layer
U_∞	free stream velocity
$U, U(t)$	instantaneous wind speed in the along-wind direction
$\overline{U}$	mean component of wind speed in the along-wind direction – typically taken over a period of 10 min or 1 h
U_{ave}	annual average wind speed at hub height
U_d	streamwise velocity at the rotor disc
U_w	streamwise velocity in the far wake
U_{e1}	extreme 3 s gust wind speed with 1 year return period
U_{e50}	extreme 3 s gust wind speed with 50 year return period
U_0	turbine upper cut-out speed
U_r	turbine rated wind speed, defined as the wind speed at which the turbine's rated power is reached
U_{ref}	reference wind speed defined as 10 min mean wind speed at hub height with 50 year return period
U_1	strain energy of plate flexure
U_2	in-plane strain energy

v	fluctuating component of wind speed in the y -direction; induced velocity in y -direction; in-plane plate deflection in y -direction
V	airspeed of an autogyro; longitudinal air velocity at rotor disc, $U_\infty(1 - a)$ (Section 7.1.9); voltage (shown in bold when complex)
$V(t)$	instantaneous lateral wind speed
VA	electrical volt-amperes
V_f	fibre volume fraction in composite material
V_t	blade tip speed
w	fluctuating component of wind speed in the z -direction; induced velocity in z -direction; out-of-plane plate deflection; weighting factor; water particle velocity in z -direction
$w(r)$	blade shell skin thickness (Section 6.4.2)
W	wind velocity relative to a point on rotating blade; electrical power loss
x	downwind co-ordinate – fixed and rotating axis systems; downwind displacement
$x(t)$	stochastic component of a variable
x_n	length of near wake region
x_0	mode of distribution
$\bar{x}_1$	first-mode component of steady tip displacement
X	electrical inductive reactance
X_n	coefficient of n th term in Dean's stream function
y	lateral co-ordinate with respect to vertical axis (starboard positive) – fixed-axis system
y	lateral co-ordinate with respect to blade axis – rotating-axis system
y	lateral displacement; reduced variate of distribution; height above seabed
z	vertical co-ordinate (upwards positive) – fixed-axis system
z	radial co-ordinate along blade axis – rotating-axis system
Z	section modulus; externally applied load on flanged joint
Z	electrical impedance (bold indicates a complex quantity)
z_0	ground roughness length
z_1	number of teeth on pinion gear
$z(t)$	periodic component of a variable
Greek	
α	angle of attack – i.e. angle between air flow incident on the blade and the blade chord line; wind-shear power law exponent; exponent of reduced variate in 3 parameter Weibull distribution; exponent of Jonswap spectrum peak shape parameter
α_x	Meridional elastic imperfection reduction factor
β	inclination of local blade chord to rotor plane (i.e. blade twist plus pitch angle, if any); radius of environmental contour
β_r	probability weighted moment to power r
χ	wake skew angle: angle between the axis of the wake of a yawed rotor and the axis of rotation of rotor; buckling strength reduction factor
χ_{M1}	weighted mass ratio defined in Section 5.8.6
δ_a	logarithmic decrement of aerodynamic damping
δ_s	logarithmic decrement of structural damping

δ	logarithmic decrement of combined aerodynamic and structural damping; width of tower shadow deficit region; depth of surface irregularity
δ_3	angle between axis of teeter hinge and the line perpendicular to both the rotor axis and the low-speed shaft axis
ε	proportion of axial stress to total stress
$\varepsilon_1, \varepsilon_2, \varepsilon_3$	proportion of time in which a variable takes the maximum, mean or minimum values in a three-level square wave
ϕ	flow angle of resultant velocity W to rotor plane; velocity potential
$\Phi()$	standard normal distribution function
$\Phi(x, y, z, t)$	velocity potential due to unit source
γ	yaw angle; Euler's constant ($= 0.5772$); Jonswap spectrum peak shape parameter
γ_L	load factor
γ_{mf}	partial safety factor for material fatigue strength
γ_{mu}	partial safety factor for material ultimate strength
Γ	blade circulation; vortex strength
$\Gamma()$	gamma function
η	ellipsoidal parameter; shaft tilt; one eighth of Lock number (defined in Section 5.8.8); skewness parameter; water surface elevation
η_b	crest elevation above still water level for a breaking wave
κ	von Karman's constant
$\kappa(t - t_0)$	auto correlation function
$\kappa_L(s)$	cross-correlation function between velocity components at points in space a distance s apart, in the direction parallel to the line joining them
$\kappa_T(s)$	cross-correlation function between velocity components at points in space a distance s apart, in the direction perpendicular to the line joining them
$\kappa_u(r, \tau)$	auto-correlation function for along-wind velocity component at radius r on stationary rotor
$\kappa_u^0(r, \tau)$	auto-correlation function for along-wind velocity component as seen by a point at radius r on a rotating rotor
$\kappa_u(r_1, r_2, \tau)$	cross-correlation function between along-wind velocity components at radii r_1 and r_2 (not necessarily on same blade), for stationary rotor
$\kappa_u^0(r_1, r_2, \tau)$	cross-correlation function between along-wind velocity components as seen by points (not necessarily on same blade) at radii r_1 and r_2 on a rotating rotor
λ	tip speed ratio; latitude; ratio of longitudinal to transverse buckle half wave- lengths; relative shell slenderness; curling factor of breaking wave
λ_r	tangential speed of blade element at radius r divided by wind speed: local speed ratio
$\lambda(d)$	ratio measuring influence of loading near cantilever root on first mode resonance
$\lambda^*(d)$	approximate value of $\lambda(d)$
Λ	yaw rate
μ	non-dimensional radial position, r/R ; viscosity; coefficient of friction
$\mu_i(r)$	mode shape of i th blade mode
$\mu_1(y)$	mode shape of first mode of offshore support structure
$\mu_i(z)$	mode shape of i th tower mode
$\mu_T(z)$	tower first mode shape

$\mu_{Tj}(r)$	normalized rigid body deflection of blade j resulting from excitation of tower first mode
μ_z	mean value of variable z
ν	ellipsoidal co-ordinate; mean zero up-crossing frequency; rank in series of data points; kinematic viscosity
θ	wind-speed direction change; random phase angle; cylindrical panel co-ordinate; brake disc temperature
ρ	air density; water density
$\rho_u^0(r_1, r_2, \tau)$	normalized cross-correlation function between along-wind velocity components as seen by points (not necessarily on same blade) at radii r_1 and r_2 on a rotating rotor (<i>i.e.</i> $\kappa_u^0(r_1, r_2, \tau)/\sigma_u^2$)
σ	blade solidity; standard deviation; stress
$\bar{\sigma}$	mean stress
σ_{cr}	elastic critical buckling stress
σ_M	standard deviation of bending moment
σ_{M1}	standard deviation of first-mode resonant bending moment, at blade root for blade resonance, and at tower base for tower resonance
σ_{MB}	standard deviation of quasistatic bending moment (or bending moment back-ground response)
σ_{Mh}	standard deviation of hub dishing moment
σ_{MT}	standard deviation of teeter moment for rigidly mounted, two-bladed rotor
$\sigma_{\bar{M}}$	standard deviation of mean of blade root bending moments for two-bladed rotor
σ_{Q1}	standard deviation of generalized load with respect to first mode
σ_r	rotor solidity
σ_u	standard deviation of fluctuating component of wind in along-wind direction
σ_v	standard deviation of wind speed in across-wind direction
σ_w	standard deviation of wind speed in vertical direction
σ_{x1}	standard deviation of first-mode resonant displacement, referred to blade tip for blade resonance, and to nacelle for tower resonance
τ	time interval; non-dimensional time; shear stress
ν	Poisson's ratio
ω	angular frequency (rad/s)
ω_d	demand generator rotational speed
ω_i	natural frequency of i th mode (rad/s)
ω_g	generator rotational speed
ω_r	induction machine rotor rotational speed
ω_s	induction machine stator field rotational speed
Ω	rotational speed of rotor; earth's rotational speed
ξ	damping ratio
ψ	blade azimuth
ψ	angle subtended by cylindrical plate panel; stream function parameter with respect to fixed reference frame; wake amplification factor
$\bar{\psi}$	stream function parameter with respect to frame of reference moving at same speed as wave crests and troughs
$\psi_{uu}(r, r', n)$	real part of normalised cross spectrum (see Appendix 1, section A1.4)
ζ	teeter angle

Subscript

<i>a</i>	aerodynamic
<i>B</i>	baseline
<i>c</i>	compressive
<i>d</i>	disc; drag; design
<i>e1</i>	extreme value with return period of 1 year
<i>e50</i>	extreme value with return period of 50 years
<i>ext</i>	extreme
<i>f</i>	fibre
<i>i</i>	mode <i>i</i>
<i>j</i>	mode <i>j</i>
<i>J</i>	blade <i>J</i>
<i>k</i>	characteristic
<i>l</i>	lift
<i>m</i>	matrix
<i>M</i>	moment
<i>max</i>	maximum value of variable
<i>min</i>	minimum value of variable
<i>n</i>	value at end of <i>n</i> th time step
<i>Q</i>	generalised load
<i>R</i>	value at tip radius, <i>R</i>
<i>s</i>	structural
<i>t</i>	tensile
<i>T</i>	thrust
<i>u</i>	downwind; ultimate
<i>v</i>	lateral
<i>w</i>	vertical
<i>w</i>	wake
<i>x</i>	deflection in along-wind direction

Superscripts

0	rotationally sampled (applied to wind-speed spectra)

Figures C1 and C2 – Co-ordinate Systems

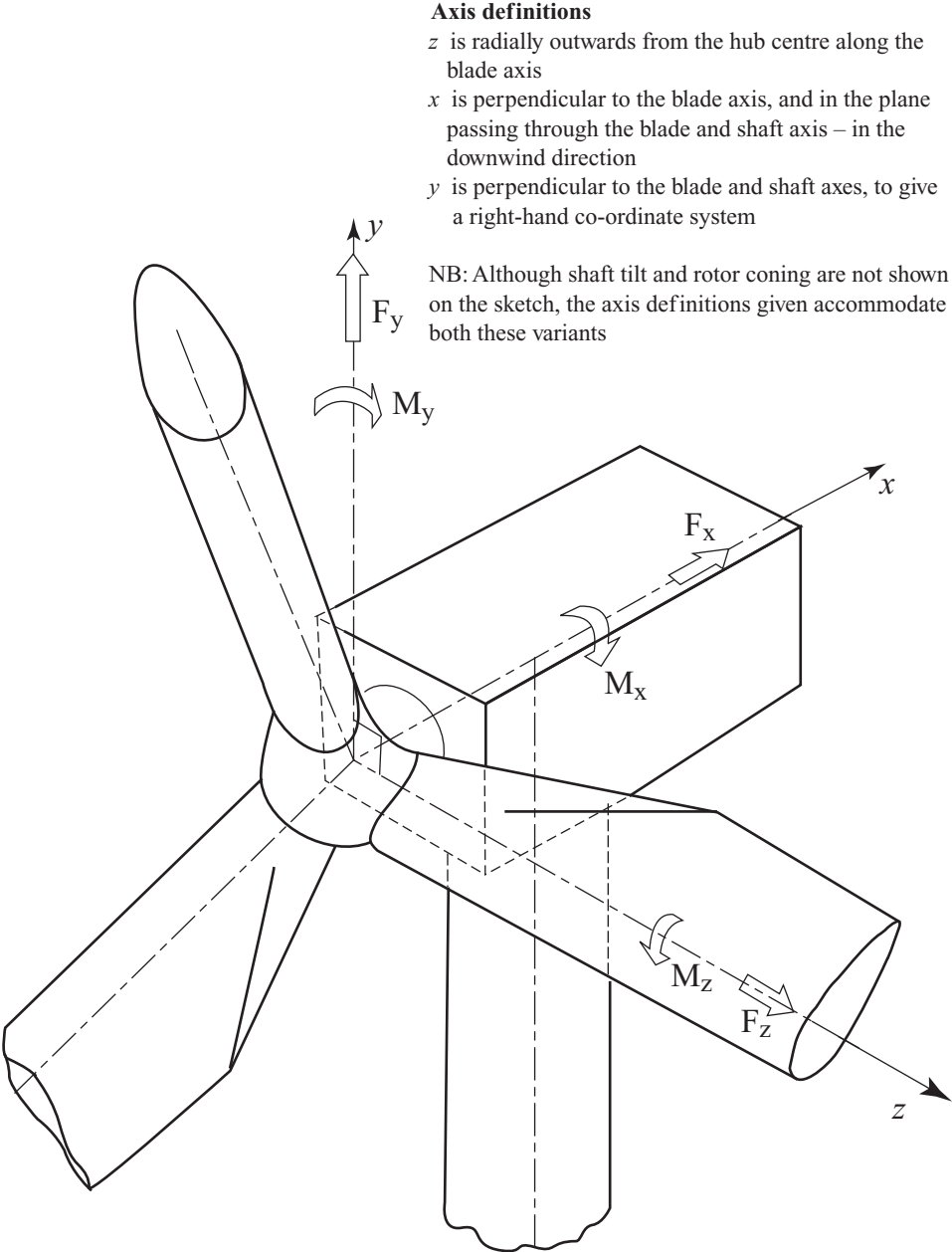


Figure C1 Co-ordinate System for Blade Loads, Positions and Deflections (rotates with blade)

Axis Definitions

x is along the shaft axis, in the downwind direction

z is perpendicular to the shaft axis and in the vertical plane passing through the shaft axis

y is perpendicular to the shaft axis and horizontal to starboard, giving a right-hand co-ordinate system

NB: Although shaft tilt and rotor coning are not shown on the sketch, the axis definitions given accommodate both these variants

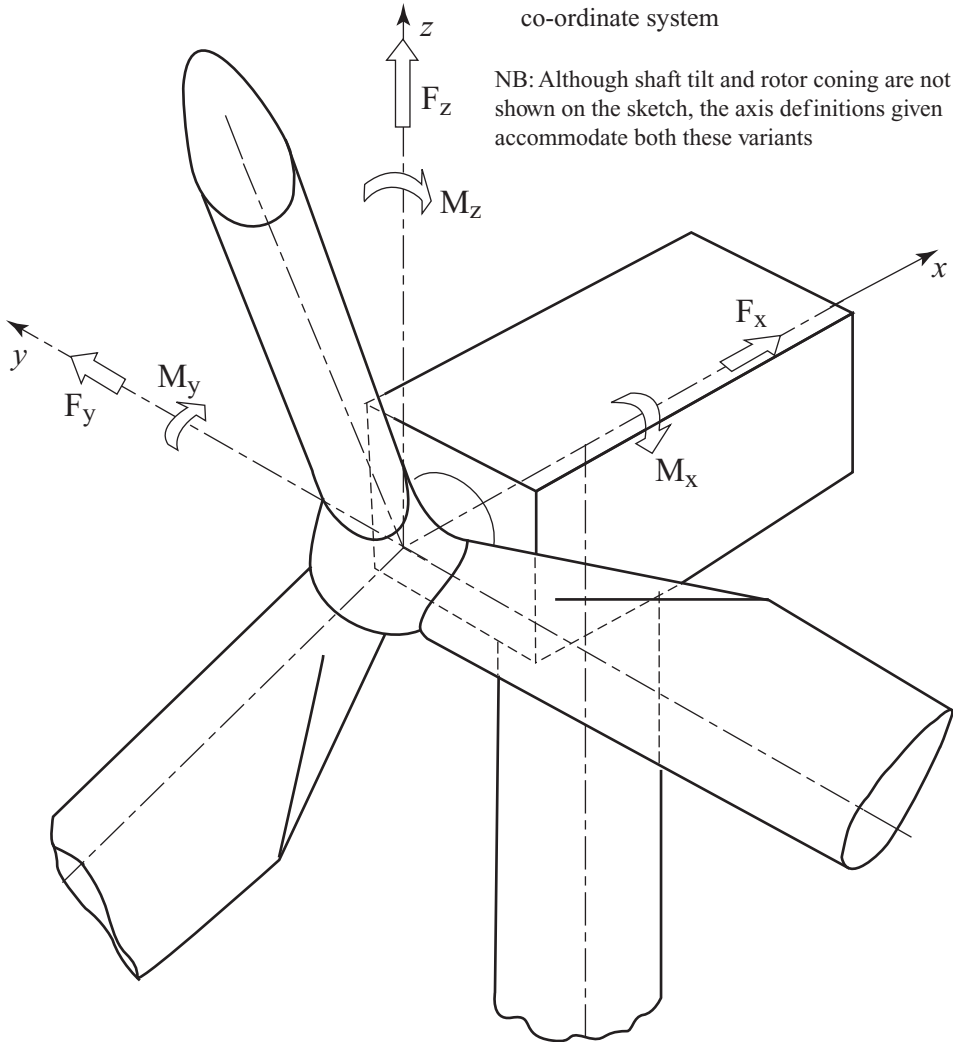


Figure C2 Fixed Co-ordinate System for Hub Loads and Deflections, and Positions with Respect to Hub



Plate 1 Figure 1.2(a) Rhyll Flats Offshore Wind Farm: 25×3.6 MW Siemens wind turbines. Rhyll Flats was built and is operated by RWE npower renewables. Photographer: Guy Woodland. Photos reproduced courtesy of RWE npower renewables.



Plate 2 Figure 1.2(c) Assembly of Siemens 3.6 MW wind turbine. Photographer: Guy Woodland. Photos reproduced courtesy of RWE npower renewables.



Plate 3 Figure 7.8 Blade Production. View of veneer lay-up in mould to make one blade skin. The blade is completed by glueing face and camber skins together. Reproduced by permission of NEG-Micon



Lillgrund 2008/01/01
Windturbin foundation Swinoujskie Piering

Plate 4 Figure 11.48 Gravity bases for Lillgrund under construction on barge at quayside at Swinoujskie. (Lillgrund Pilot Project, 2008). Reproduced by permission of Vattenfall



Lillgrund Wind Power Plant 2008-09-21
The foundations will be lifted up from the barge at Lillgrund

Plate 5 Figure 11.49 Lowering of gravity base by floating crane during installation at Lillgrund windfarm. (Lillgrund Pilot Project, 2008). Reproduced by permission of Vattenfall

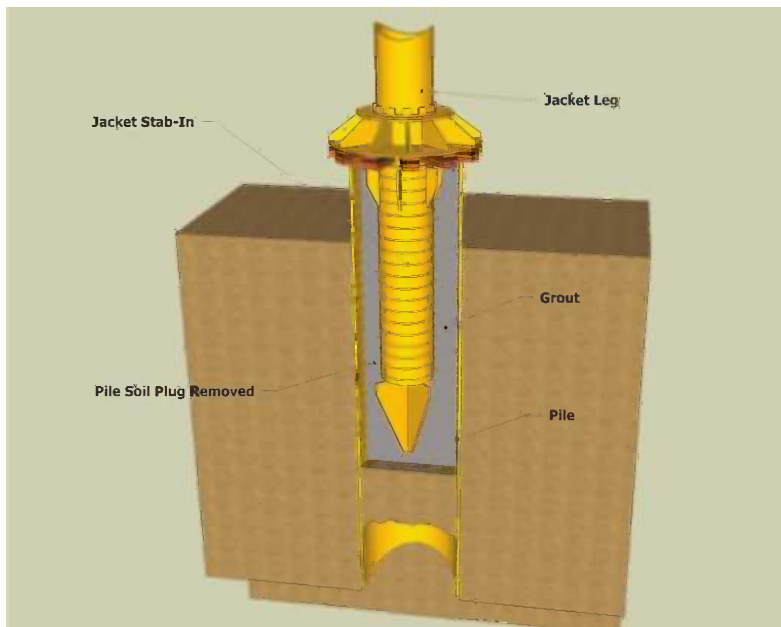


Plate 6 Figure 11.52 Anchorage of jacket leg to pile using concentric jacket stab-in and grouted joint—Ormonde windfarm. Reproduced by permission of Offshore Design Engineering Ltd (ode)



Plate 7 Figure 11.55 Tripile structure after installation. The yellow pile tops, which sport circumferential stripe markings, extend up to platform level with the cylinders above forming part of the three-arm transition structure. Reproduced by permission of BARD Group/Scheer



Plate 8 Figure 11.59 Offshore sub-station at Lillgrund. Lillgrund Pilot Project, 2008. Reproduced by permission of Vattenfall

Introduction

1.1 Historical development

Windmills have been used for at least 3000 years, mainly for grinding grain or pumping water; while in sailing ships the wind has been an essential source of power for even longer. From medieval times, horizontal axis windmills were an integral part of the rural economy and only fell into disuse with the advent of cheap fossil-fuelled stationary engines and then the spread of rural electrification (Musgrove, 2010). The use of windmills (or wind turbines) to generate electricity can be traced back to the late nineteenth century with the 12 kW direct current windmill generator constructed by Charles Brush in the USA and the research undertaken by Poul la Cour in Denmark. However, for much of the twentieth century there was little interest in using wind energy for electricity generation, other than for battery charging for remote dwellings; and these low power systems were quickly removed once access to the electricity grid became available. One notable development was the 1250 kW Smith-Putnam wind turbine constructed in the USA in 1941. This remarkable machine had a steel rotor 53 m in diameter, full span pitch control and flapping blades to reduce loads. Although a blade spar failed catastrophically in 1945, it remained the largest wind turbine constructed for some 40 years (Putnam, 1948).

Golding (1955) and Shepherd and Divone in Spera (1994) provide a fascinating history of early wind turbine development. They record the 100 kW 30 m diameter Balaclava wind turbine in the then USSR in 1931 and the Andrea Enfield 100 kW 24 m diameter pneumatic design constructed in the UK in the early 1950s. In this turbine, hollow blades, open at the tip, were used to draw air up through the tower where another turbine drove the generator. In Denmark the 200 kW 24 m diameter Gedser machine was built in 1956, while Electricite de France tested a 1.1 MW 35 m diameter turbine in 1963. In Germany, Professor Ulrich Hutter constructed a number of innovative, lightweight turbines in the 1950s and 1960s. In spite of these technical advances and the enthusiasm of Golding at the Electrical Research Association in the UK, among others, there was little sustained interest in wind generation until the price of oil rose dramatically in 1973.

The sudden increase in the price of oil stimulated a number of substantial, government funded programmes of research, development and demonstration. In the USA this led to the construction of a series of prototype turbines starting with the 38 m diameter 100 kW Mod-0 in 1975 and culminating in the 97.5 m diameter 2.5 MW Mod-5B in 1987. Similar programmes were pursued in the UK, Germany and Sweden. There was considerable uncertainty as to which architecture might prove most cost-effective and several innovative concepts were investigated at full scale. In Canada, a 4 MW vertical axis Darrieus wind turbine was constructed and this concept was also investigated in the 34 m diameter Sandia Vertical Axis test facility in the USA. In the UK, an alternative, vertical axis design using straight blades to give an 'H' type rotor was proposed by Dr Peter Musgrove and a 500 kW prototype constructed (Musgrove, 2010). In 1981 an innovative horizontal axis 3 MW wind turbine was built and tested in the USA. This used hydraulic transmission and, as an alternative to a yaw drive, the entire structure was orientated into the wind. The best choice for the number of blades remained unclear for some while and large horizontal axis turbines were constructed with one, two or three blades.

Much important scientific and engineering information was gained from these government funded research programmes and the prototypes generally worked as designed. However, the problems of operating very large wind turbines, unmanned and in difficult wind climates, were often underestimated and the reliability of the prototypes was not good. At the same time as the multi-megawatt prototypes were being constructed private companies, often with considerable state support, were manufacturing much smaller, often simpler, turbines for commercial sale. In particular the financial support mechanisms in California in the mid-1980s resulted in installation of a very large number of quite small (<100 kW) wind turbines. A number of these designs also suffered from various problems but, being smaller, they were generally easier to repair and modify. The Danish wind turbine concept emerged of a three-bladed, upwind stall regulated rotor and a fixed-speed, induction generator drive train. This deceptively simple architecture proved to be remarkably successful and was implemented on turbines as large as 60 m in diameter and at ratings of up to 1.5 MW. However, at large rotor diameters and generator ratings, the architecture ceases to be effective as aerodynamic stall is increasingly difficult to predict, an induction generator no longer easily provides enough damping and torsional compliance in the drive train and the requirements of the electrical Transmission System Operators for connection to the network, the Grid Codes become difficult to meet. Hence, as the size of commercially available turbines approached or exceeded that of the large prototypes of the 1980s the concepts investigated then of variable speed operation, full-span control of the blade pitch and advanced materials were used increasingly by designers.

In 1991 the first offshore wind farm was constructed at Vindeby consisting of eleven, 450 kW wind turbines located up to 3 km offshore. Throughout the 1990s small numbers of offshore wind turbines were placed close to shore, while in 2002 the Horns Rev, 160 MW wind farm, some 20 km off the western coast of Denmark, was constructed. This was the first project to use an offshore substation that increased the power collection voltage of 30 kV to 150 kV for transmission to shore. At the time of writing (2010) a 500 MW wind offshore wind farm (Greater Gabbard) is under construction off the coast of England with 1000 MW projects under development. The wind turbines that have been installed in these offshore wind farms have been marinised versions of 3-bladed, upwind terrestrial designs. However, the possibility of higher blade tip-speeds, because of more relaxed noise constraints and a reduced emphasis on visual appearance in sites far from land, are leading to an interest in the development of very large, lower solidity rotors with two or even one blade.

The stimulus for the development of wind energy in 1973 was the increase in the price of oil and concern over limited fossil fuel resources. From around 1990, the main driver for use of wind turbines to generate electrical power was the very low CO₂ emissions (over the entire life cycle of manufacture, installation, operation and de-commissioning) and the potential of wind energy to help mitigate climate change. Then from around 2006 the very high oil price and concerns over security of energy supplies led to a further increase of interest in wind energy and a succession of policy measures were put in place in many countries to encourage its use. In 2007 the European Union declared a policy that 20% of all energy should be from renewable sources by 2020. Because of the difficulty of using renewable energy for transport and heat, this implies that in some countries 30–40% of electrical energy should come from renewables, with wind energy likely to play a major part. Energy policy continues to develop rapidly with many countries adopting ambitions to reduce greenhouse gas emissions of up to 80% by 2050 in order to mitigate climate change.

The development of wind energy in some countries has been more rapid than in others and this difference cannot be explained simply by differences in the wind speeds. Important factors include the financial support mechanisms for wind generated electricity, access to the electrical network, the process by which the local authorities give permission for the construction of wind farms and the perception of the general population, particularly with respect to visual impact. The development of offshore sites, although at considerably increased cost, is in response to these concerns over the environmental impact of wind farms.

As a relatively new generation technology, wind energy requires financial support to encourage its development and stimulate investment from private companies. Such support is provided in many countries and recognises the contribution wind generation makes to climate change mitigation and security of national energy supplies. There is presently an active debate as to the best mechanism of providing such support so that it stimulates the development of wind energy at minimum cost and without distorting the electricity market.

Feed-in-Tariffs are offered in a number of countries, most notably Germany and Spain. A fixed price is paid for each kWh generated from renewable sources with different rates for wind energy, photovoltaic solar energy and other renewable energy technologies. This support mechanism has the benefit of giving certainty of the revenue stream from a successful project and is credited by its supporters for the very rapid development of wind energy, and other renewables, in these countries. An alternative approach is a quota or Renewable Portfolio Standard system where a government places an obligation on electricity suppliers to source a certain fraction of the energy they supply from renewable sources. An example is the UK Renewable Obligation Certificate system where renewable energy generators are awarded green certificates for energy generated from renewable sources. These green certificates are traded independently from the electrical energy and electricity suppliers who fail to acquire their quota pay a buy-out price. With this support mechanism risk is transferred to the project developer and only the most commercially attractive renewable technologies are developed. Historically, Capacity Auctions have also been used, such as the earlier UK NFFO mechanism and similar examples in Ireland and France. A national government determines the volume of wind energy required and conducts an auction for capacity on price. Capacity Auctions suffered from some wind farm developers bidding low to secure agreements and then not constructing projects.

Although the form of these support mechanisms, and particularly their stability, is important, it may be argued that other factors including access to the electricity grid, speed of the planning/permitting system and public acceptability play a critical role in determining the rate

of deployment of wind energy. It is also likely that more general support measures for low carbon electricity generation, for example the European Union Emissions Trading Scheme or wider carbon taxes, will provide significant support for the development of wind energy in the future.

1.2 Modern wind turbines

The power output from a wind turbine is given by the well-known expression:

$$P = \frac{1}{2} C_p \rho A U^3 \quad (1.1)$$

where

ρ is the density of air (1.25 kg/m³)

C_p is the power coefficient

A is the rotor swept area

U is the wind speed

The density of air is rather low, 800 times less than water which powers hydro plant, and this leads directly to the large size of a wind turbine. Depending on the design wind speed chosen, a 3 MW wind turbine may have a rotor that is more than 90 m in diameter. The power coefficient describes that fraction of the power in the wind that may be converted by the turbine into mechanical work. It has a theoretical maximum value of 0.593 (the Betz limit) and rather lower peak values are achieved in practice (see Chapter 3). The power coefficient of a rotor varies with the tip speed ratio, (the ratio of rotor tip speed to free wind speed) and is only a maximum for a unique tip speed ratio. Incremental improvements in the power coefficient are continually being sought by detailed design changes of the rotor and by operating at variable speed it is possible to maintain the maximum power coefficient over a range of wind speeds. However, these measures will give only a modest increase in the power output. Major increases in the output power can only be achieved by increasing the swept area of the rotor or by locating the wind turbines on sites with higher wind speeds.

Hence, over the last 40 years there has been a continuous increase in the rotor diameter of commercially available wind turbines from less than 30 m to more than 100 m. A tripling of the rotor diameter leads to a nine times increase in power output. The influence of the wind speed is, of course, more pronounced with a doubling of wind speed leading to an eight fold increase in power. Thus, there have been considerable efforts to ensure that wind farms are developed in areas of the highest wind speeds and the turbines optimally located within wind farms. In certain countries very high towers are being used (more than 100 m high) to take advantage of the increase of wind speed with height.

In the past a number of studies were undertaken to determine the ‘optimum’ size of a wind turbine by balancing the complete costs of manufacture, installation and operation of various sizes of wind turbines against the revenue generated (Molly *et al.*, 1993). The results indicated a minimum cost of energy would be obtained with wind turbine diameters in the range of 35–60 m, depending on the assumptions made. However, these estimates would now appear to be too low and there is no obvious point at which rotor diameters and, hence, output

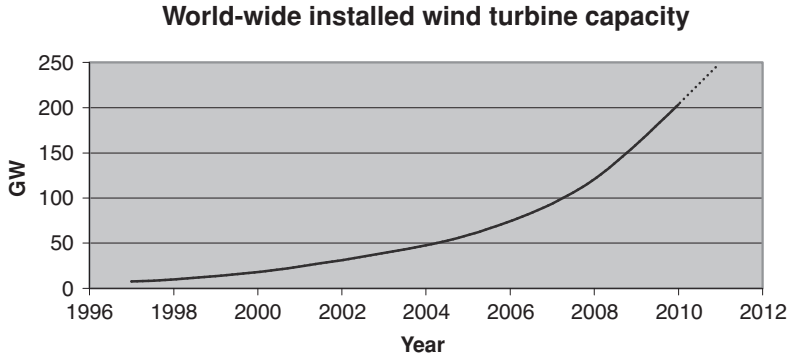


Figure 1.1 Wind power capacity world-wide (World Wind Energy Association, 2009)

power will be limited particularly for offshore wind turbines where the very large components can be transported by ship directly from the factory to site.

All modern electricity generating wind turbines use the lift force derived from the blades to drive the rotor. A high rotational speed of the rotor is desirable in order to reduce the gearbox ratio required and this leads to low solidity rotors (the ratio of blade area/rotor swept area). The low solidity rotor acts as an effective energy concentrator and as a result the energy generated over a wind turbine's life is much less than that used for its manufacture and installation. An energy balance analysis of a 3 MW wind turbine showed that the expected average time to generate a similar quantity of energy to that used for its manufacture, operation, transport, dismantling and disposal was 6–7 months (European Wind Energy Association (EWEA), 2009). A similar time was calculated for installation both onshore and offshore.

Until around 2000, the installed wind turbine generating capacity was so low that its output was viewed by electricity Transmission System Operators simply as negative load that supplied energy but played no part in supporting the operation of the power system and maintaining its stability. Since then, with the very much increased capacity of wind generation (Figure 1.1), turbines are required to contribute to the operation of the power system. The requirements for their performance are defined through the Grid Codes, issued by the Transmission System Operators. Compliance is mandatory before connection to the network is allowed. The Grid Codes specify operational requirements so that in addition to contributing energy, or real power,¹ the wind turbines provide ancillary services particularly for voltage and frequency control. At present the Grid Codes differ in detail from country to country but generally specify that wind turbines must remain stable and connected to the network in case of electrical faults on the network, define their performance in terms of reactive power for voltage control and their ability to vary real power for frequency support. As wind turbines become an ever increasing fraction of electricity generating capacity, it is likely that further requirements will be placed on them to replicate the ancillary services previously provided by conventional synchronous generators. Compliance with the Grid Code requirements is difficult to achieve

¹In a large interconnected electric power transmission system, real power controls system frequency while reactive power determines the voltages of the network.

with simple fixed speed induction generators using the Danish concept and these regulations are a major driver for the use of variable speed generators.

1.3 Scope of the book

The use of wind energy to generate electricity is now well accepted with a large industry manufacturing and installing tens of GWs of new capacity each year. Although there are exciting new developments, particularly in very large wind turbines, and many challenges remain, there is a considerable body of established knowledge concerning the science and technology of wind turbines. This book records some of this knowledge and presents it in a form suitable for use by students (at final year undergraduate or post-graduate level) and by those involved in the design, manufacture or operation of wind turbines. The overwhelming majority of wind turbines presently in use are horizontal axis connected to a large electricity network. These turbines are the subject of this book.

Chapter 2 discusses the wind resource. Particular reference is made to wind turbulence due to its importance in wind turbine design. Chapter 3 sets out the basis of the aerodynamics of horizontal axis wind turbines, while Chapter 4 discusses aspects of their performance. Any wind turbine design starts with establishing the design loads and these are discussed in Chapter 5. Chapter 6 sets out the various design options for horizontal axis wind turbines with approaches to the design of some of the important components examined in Chapter 7. The functions of the wind turbine controller and some of the possible analysis techniques described are discussed in Chapter 8. In Chapter 9 wind farms and the development of wind energy projects are reviewed with particular emphasis on environmental impact. Chapter 10 considers how wind turbines interact with the electrical power system while Chapter 11 deals with the important topic of offshore wind energy.

The book attempts to record well-established knowledge that is relevant to wind turbines which are currently commercially significant. Thus, it does not discuss a number of interesting research topics or areas where wind turbine technology is still evolving. Although they were investigated in considerable detail in the 1980s, large vertical axis wind turbines have not proved to be commercially competitive and are not currently manufactured in significant numbers. Hence, the particular issues of vertical axis turbines are not dealt with in this text.

There are presently some 2 billion people in the world without access to mains electricity and, in conjunction with other generators (e.g. diesel engines), wind turbines may in the future be an effective means of providing some of them with power. However, autonomous power systems are extremely difficult to design and operate reliably, particularly in remote areas of the world and with limited budgets. A small autonomous AC power system has all the technical challenges of a large national electricity system but, due to the low inertia of the generators, requires a very fast, sophisticated control system to maintain stable operation. Over the last 30 years there have been a number of attempts to operate autonomous wind-diesel systems on islands or for other remote communities throughout the world, but with only limited success. This class of installation has its own particular problems and again, given the very limited size of the market at present, this specialist area is not dealt with in this book.

Installations of offshore wind turbines are now commencing in significant numbers (Figure 1.2). The first offshore wind farms were installed in rather shallow waters and used marinised terrestrial designs. A number of larger offshore wind farms used offshore substations to increase the transmission voltage, while prototype floating wind turbines for deeper waters



(a)



(b)



(c)

Figure 1.2 (a) Rhyl Flats Offshore Wind Farm. See Plate 1 for the colour figure. (b) 3.6 MW nacelle prior to lifting. (c) Assembly of 3.6 MW wind turbine. Rhyl Flats Offshore Wind Farm consists of 25×3.6 MW Siemens wind turbines. Hub height – 80 m above mean sea level (MSL). Height to blade tip – 134 m above MSL. Rhyl Flats Offshore Wind Farm was built and is operated by RWE npower renewables. Photographer: Guy Woodland. Photos reproduced courtesy of RWE npower renewables. See Plate 2 for the colour figure

have been deployed. Very large wind farms with multi-megawatt turbines many kilometres offshore are now being planned and will be constructed over the coming years.

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The wind resource

2.1 The nature of the wind

The energy available in the wind varies as the cube of the wind speed, so an understanding of the characteristics of the wind resource is critical to all aspects of wind energy exploitation, from the identification of suitable sites and predictions of the economic viability of wind farm projects through to the design of wind turbines themselves, and understanding their effect on electricity distribution networks and consumers.

From the point of view of wind energy, the most striking characteristic of the wind resource is its variability. The wind is highly variable, both geographically and temporally. Furthermore this variability persists over a very wide range of scales, both in space and time. The importance of this is amplified by the cubic relationship to available energy.

On a large scale, spatial variability describes the fact that there are many different climatic regions in the world, some much windier than others. These regions are largely dictated by the latitude, which affects the amount of insolation. Within any one climatic region, there is a great deal of variation on a smaller scale, largely dictated by physical geography – the proportion of land and sea, the size of land masses and the presence of mountains or plains, for example. The type of vegetation may also have a significant influence through its effects on the absorption or reflection of solar radiation, affecting surface temperatures, and on humidity.

More locally, the topography has a major effect on the wind climate. More wind is experienced on the tops of hills and mountains than in the lee of high ground or in sheltered valleys, for instance. More locally still, wind velocities are significantly reduced by obstacles such as trees or buildings.

At a given location, temporal variability on a large scale means that the amount of wind may vary from one year to the next, with even longer scale variations on a scale of decades or more. These long-term variations are not well understood, and may make it difficult to make accurate predictions of the economic viability of particular wind farm projects, for instance.

On timescales shorter than a year, seasonal variations are much more predictable, although there are large variations on shorter timescales still, which although reasonably well

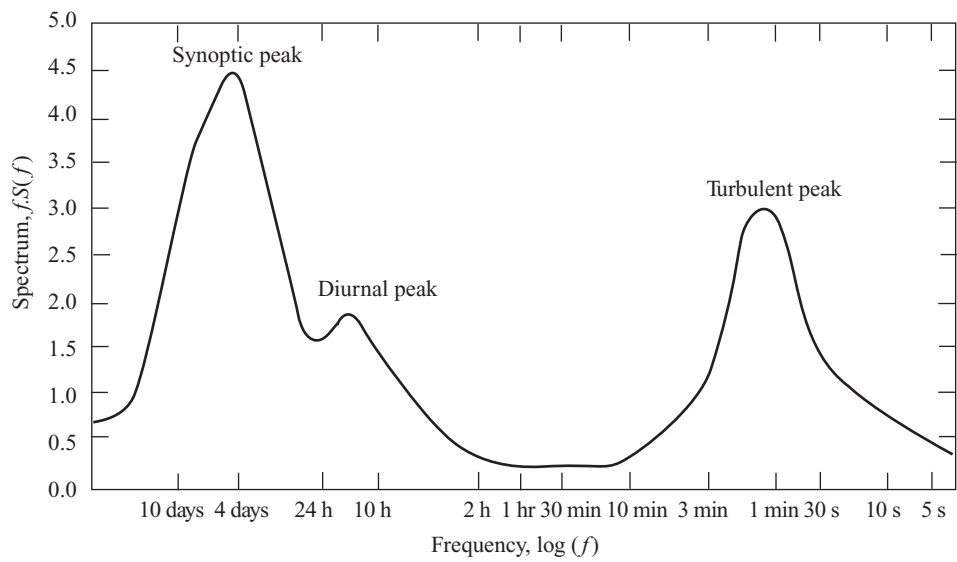


Figure 2.1 Wind spectrum from Brookhaven based on work by van der Hoven (1957)

understood are often not very predictable more than a few days ahead. Depending on location, there may also be considerable variations with the time of day (diurnal variations) which again are usually fairly predictable. On these timescales, the predictability of the wind is important for integrating large amounts of wind power into the electricity network, to allow the other generating plant supplying the network to be organised appropriately.

On still shorter timescales of minutes down to seconds or less, wind speed variations known as turbulence can have a very significant impact on the design and performance of the individual wind turbines, as well as on the quality of power delivered to the network and its effect on consumers.

Van der Hoven (1957) constructed a wind speed spectrum from long and short-term records at Brookhaven, New York, showing clear peaks corresponding to the synoptic, diurnal and turbulent effects referred to above (Figure 2.1). Of particular interest is so-called ‘spectral gap’ occurring between the diurnal and turbulent peaks, showing that the synoptic and diurnal variations can be treated as quite distinct from the higher frequency fluctuations of turbulence. There is very little energy in the spectrum in the region between two hours and ten minutes. As indicated in the next section, the nature of the wind regime in different geographical locations can vary widely, so the Van der Hoven spectrum cannot be assumed to be universally applicable. Nevertheless the concept of a spectral gap at frequencies below ten minutes is very often used, often implicitly, when making assumptions about the wind regime.

2.2 Geographical variation in the wind resource

Ultimately the winds are driven almost entirely by the sun’s energy, causing differential surface heating. The heating is most intense on land masses closer to the equator, and obviously the greatest heating occurs in the daytime, which means that the region of greatest heating

moves around the earth's surface as it spins on its axis. Warm air rises and circulates in the atmosphere to sink back to the surface in cooler areas. The resulting large scale motion of the air is strongly influenced by Coriolis forces due to the earth's rotation. The result is a large-scale global circulation pattern. Certain identifiable features of this are well known, such as the trade winds and the 'roaring forties'.

The non-uniformity of the earth's surface, with its pattern of land masses and oceans, ensures that this global circulation pattern is disturbed by smaller-scale variations on continental scales. These variations interact in a highly complex and non-linear fashion to produce a somewhat chaotic result, which is at the root of the day-to-day unpredictability of the weather in particular locations. Clearly though, underlying tendencies remain which lead to clear climatic differences between regions. These differences are tempered by more local topographical and thermal effects.

Hills and mountains result in local regions of increased wind speed. This is partly a result of altitude – the boundary layer flow over the earth's surface means that wind speed generally increases with height above the ground, and hill tops and mountain peaks may 'project' into the higher wind speed layers. It is also partly a result of the acceleration of the wind flow over and around hills and mountains, and funnelling through passes or along valleys aligned with the flow. Equally, topography may produce areas of reduced wind speed, such as sheltered valleys, areas in the lee of a mountain ridge or where the flow patterns result in stagnation points.

Thermal effects may also result in considerable local variations. Coastal regions are often windy because of differential heating between land and sea. While the sea is warmer than the land, a local circulation develops in which surface air flows from the land to the sea, with warm air rising over the sea and cool air sinking over the land. When the land is warmer the pattern reverses. The land will heat up and cool down more rapidly than the sea surface, and so this pattern of land and sea breezes tends to reverse over a 24-hour cycle. These effects were important in the early development of wind power in California, where an ocean current brings cold water to the coast, not far from desert areas which heat up strongly by day. An intervening mountain range funnels the resulting air flow through its passes, generating locally very strong and reliable winds (which are well correlated with peaks in the local electricity demand caused by air conditioning loads).

Thermal effects may also be caused by differences in altitude. Thus, cold air from high mountains can sink down to the plains below, causing quite strong and highly stratified 'downslope' winds.

The brief general descriptions of wind speed variations in Sections 2.1–2.5 are illustrative, and more detailed information can be found in standard meteorological texts. Section 9.1.3, in Chapter 9, describes how the wind regimes at candidate sites can be assessed, while wind forecasting is covered in Section 2.9.

Section 2.6 presents a more detailed description of the high frequency wind fluctuations known as turbulence, which are crucial to the design and operation of wind turbines and have a major influence on wind turbine loads. Extreme winds are also important for the survival of wind turbines, and these are described in Section 2.8.

2.3 Long-term wind speed variations

There is evidence that the wind speed at any particular location may be subject to very slow long-term variations. Although the availability of accurate historical records is a limitation,

careful analysis by, for example, Palutikoff *et al.* (1991) has demonstrated clear trends. Clearly these may be linked to long-term temperature variations for which there is ample historical evidence. There is also much debate at present about the likely effects of global warming, caused by human activity, on climate, and this will undoubtedly affect wind climates in the coming decades.

Apart from these long-term trends there may be considerable changes in windiness at a given location from one year to the next. These changes have many causes. They may be coupled to global climate phenomena such as *el niño*, changes in atmospheric particulates resulting from volcanic eruptions, and sunspot activity, to name a few.

These changes add significantly to the uncertainty in predicting the energy output of a wind farm at a particular location during its projected lifetime.

2.4 Annual and seasonal variations

While year-to-year variation in annual mean wind speeds remains hard to predict, wind speed variations during the year can be well characterised in terms of a probability distribution. The Weibull distribution has been found to give a good representation of the variation in hourly mean wind speed over a year at many typical sites. This distribution takes the form

$$F(U) = \exp \left(- \left(\frac{U}{c} \right)^k \right) \quad (2.1)$$

where $F(U)$ is the fraction of time for which the hourly mean wind speed exceeds U . It is characterised by two parameters: a ‘scale parameter’ c and a ‘shape parameter’ k which describes the variability about the mean. c is related to the annual mean wind speed $\bar{U}$ by the relationship

$$\bar{U} = c\Gamma(1 + 1/k) \quad (2.2)$$

where Γ is the complete gamma function. This can be derived by consideration of the probability density function

$$f(U) = -\frac{dF(U)}{dU} = k \frac{U^{k-1}}{c^k} \exp \left(- \left(\frac{U}{c} \right)^k \right) \quad (2.3)$$

since the mean wind speed is given by

$$\bar{U} = \int_0^\infty U f(U) dU \quad (2.4)$$

A special case of the Weibull distribution is the Rayleigh distribution, with $k = 2$, which is actually a fairly typical value for many locations. In this case, the factor $\Gamma(1 + 1/k)$ has the value $\sqrt{\pi}/2 = 0.8862$. A higher value of k , such as 2.5 or 3, indicates a site where the variation of hourly mean wind speed about the annual mean is small, as is sometimes the case

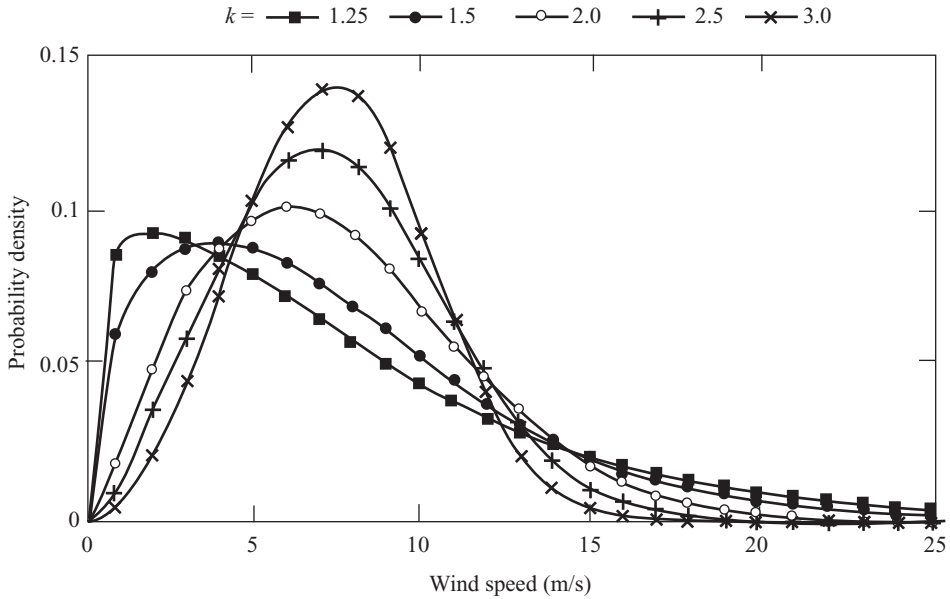


Figure 2.2 Example Weibull distributions

in the trade wind belts, for instance. A lower value of k , such as 1.5 or 1.2, indicates greater variability about the mean. A few examples are shown in Figure 2.2. The value of $\Gamma(1 + 1/k)$ varies little, between about 1.0 and 0.885: see Figure 2.3.

The Weibull distribution of hourly mean wind speeds over the year is clearly the result of a considerable degree of random variation. However, there may also be a strong underlying

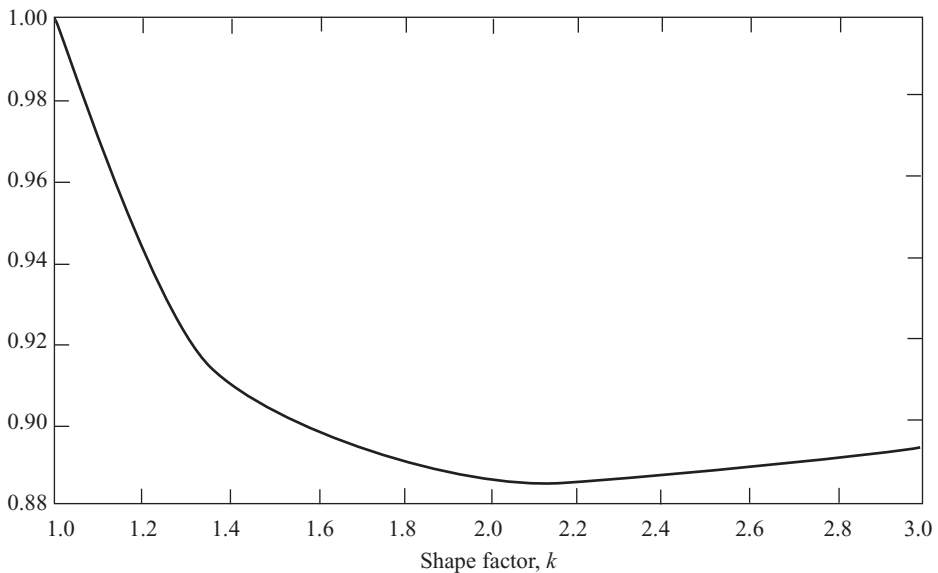


Figure 2.3 The factor $\Gamma(1 + 1/k)$

seasonal component to these variations, driven by the changes in insolation during the year as a result of the tilt of the earth's axis of rotation. Thus, in temperate latitudes the winter months tend to be significantly windier than the summer months. There may also be a tendency for strong winds or gales to develop around the time of the spring and autumn equinoxes. Tropical regions also experience seasonal phenomena, such as monsoons and tropical storms, which affect the wind climate. Indeed the extreme winds associated with tropical storms may significantly influence the design of wind turbines intended to survive in these locations.

Although a Weibull distribution gives a good representation of the wind regime at many sites, this is not always the case. For example, some sites showing distinctly different wind climates in summer and winter can be represented quite well by a double-peaked 'bi-Weibull' distribution, with different scale factors and shape factors in the two seasons, that is,

$$F(U) = F_1 \exp\left(-\left(\frac{U}{c_1}\right)^{k_1}\right) + (1 - F_1) \exp\left(-\left(\frac{U}{c_2}\right)^{k_2}\right) \quad (2.5)$$

Certain parts of California are good examples of this.

2.5 Synoptic and diurnal variations

On shorter timescales than the seasonal changes described in Section 2.4, wind speed variations are somewhat more random and less predictable. Nevertheless, these variations contain definite patterns. The frequency content of these variations typically peaks at around four days or so. These are the 'synoptic' variations which are associated with large-scale weather patterns such as areas of high and low pressure and associated weather fronts as they move across the earth's surface. Coriolis forces induce a circular motion of the air as it tries to move from high to low pressure regions. These coherent large-scale atmospheric circulation patterns may typically take a few days to pass over a given point, although they may occasionally 'stick' in one place for longer before finally moving on or dissipating.

Following the frequency spectrum to still higher frequencies, many locations will show a distinct diurnal peak, at a frequency of 24 hours. This is usually driven by local thermal effects. Intense heating in the daytime may cause large convection cells in the atmosphere, which die down at night. This process is described in more detail in Section 2.6 as it also contributes significantly to turbulence, on timescales representative of the size of the convection cells. Land and sea breezes, caused by differential heating and cooling between land and sea, also contribute significantly to the diurnal peak. The daily direction reversal of these winds would be seen as a 12-hour peak in the spectrum of wind speed magnitude.

2.6 Turbulence

2.6.1 The nature of turbulence

Turbulence refers to fluctuations in wind speed on a relatively fast timescale, typically less than about ten minutes. In other words it corresponds to the highest frequency spectral peak in Figure 2.1. It is useful to think of the wind as consisting of a mean wind speed determined by the seasonal, synoptic and diurnal effects described above, which varies on a timescale of

one to several hours, with turbulent fluctuations superimposed. These turbulent fluctuations then have a zero mean when averaged over about ten minutes. This description is a useful one as long as the ‘spectral gap’ as illustrated in Figure 2.1 is reasonably distinct.

Turbulence is generated mainly from two causes: ‘friction’ with the earth’s surface, which can be thought of as extending as far as flow disturbances caused by topographical features such as hills and mountains; and thermal effects which can cause air masses to move vertically as a result of variations in temperature and, hence, in the density of the air. Often these two effects are interconnected, such as when a mass of air flows over a mountain range and is forced up into cooler regions where it is no longer in thermal equilibrium with its surroundings.

Turbulence is clearly a complex process, and one which cannot be represented simply in terms of deterministic equations. Clearly it does obey certain physical laws, such as those describing the conservation of mass, momentum and energy. However, in order to describe turbulence using these laws it is necessary to take account of temperature, pressure, density and humidity as well as the motion of the air itself in three dimensions. It is then possible to formulate a set of differential equations describing the process, and in principle the progress of the turbulence can be predicted by integrating these equations forward in time starting from certain initial conditions and subject to certain boundary conditions. In practice of course, the process can be described as ‘chaotic’ in that small differences in initial conditions or boundary conditions may result in large differences in the predictions after a relatively short time. For this reason it is generally more useful to develop descriptions of turbulence in terms of its statistical properties.

There are many statistical descriptors of turbulence which may be useful, depending on the application. These range from simple turbulence intensities and gust factors to detailed descriptions of the way in which the three components of turbulence vary in space and time as a function of frequency.

The turbulence intensity is a measure of the overall level of turbulence. It is defined as

$$I = \frac{\sigma}{\bar{U}} \quad (2.6)$$

where σ is the standard deviation of wind speed variations about the mean wind speed $\bar{U}$, usually defined over ten minutes or an hour. Turbulent wind speed variations can be considered to be roughly Gaussian, meaning that the speed variations are normally distributed, with standard deviation σ , about the mean wind speed $\bar{U}$. However, the tails of the distribution may be significantly non-Gaussian, so this approximation is not reliable for estimating, say, the probability of a large gust within a certain period.

The turbulence intensity clearly depends on the roughness of the ground surface and the height above the surface. However, it also depends on topographical features such as hills or mountains, especially when they lie upwind, as well as more local features such as trees or buildings. It also depends on the thermal behaviour of the atmosphere: for example, if the air near to the ground warms up on a sunny day it may become buoyant enough to rise up through the atmosphere, causing a pattern of convection cells which are experienced as large-scale turbulent eddies.

Clearly as the height above ground increases, the effects of all these processes which are driven by interactions at the earth’s surface become weaker. Above a certain height, the air flow can be considered largely free of surface influences. Here it can be considered to be driven by large scale synoptic pressure differences and the rotation of the earth. This air flow

is known as the geostrophic wind. At lower altitudes, the effect of the earth's surface can be felt. This part of the atmosphere is known as the boundary layer. The properties of the boundary layer are important in understanding the turbulence experienced by wind turbines.

2.6.2 The boundary layer

The principal effects governing the properties of the boundary layer are the strength of the geostrophic wind, the surface roughness, Coriolis effects due to the earth's rotation and thermal effects.

The influence of thermal effects can be classified into three categories: stable, unstable and neutral stratification. Unstable stratification occurs when there is a lot of surface heating, causing warm air near the surface to rise. As it rises, it expands due to reduced pressure and, therefore, cools adiabatically. If the cooling is not sufficient to bring the air into thermal equilibrium with the surrounding air then it will continue to rise, giving rise to large convection cells. The result is a thick boundary layer with large-scale turbulent eddies. There is a lot of vertical mixing and transfer of momentum, resulting in a relatively small change of mean wind speed with height.

If the adiabatic cooling effect causes the rising air to become colder than its surroundings, its vertical motion will be suppressed. This is known as stable stratification. It often occurs on cold nights when the ground surface is cold. In this situation, turbulence is dominated by friction with the ground and wind shear (the increase of mean wind speed with height) can be large.

In the neutral atmosphere, adiabatic cooling of the air as it rises is such that it remains in thermal equilibrium with its surroundings. This is often the case in strong winds, when turbulence caused by ground roughness causes sufficient mixing of the boundary layer. For wind energy applications, neutral stability is usually the most important situation to consider, particularly when considering the turbulent wind loads on a turbine, since these are largest in strong winds. Nevertheless, unstable conditions can be important as they can result in sudden gusts from a low level, and stable conditions can give rise to significant asymmetric loadings due to high wind shear. There can also be rapid changes in wind direction with height in this situation.

In the following sections, a series of relationships are presented which describe the properties of the atmospheric boundary layer, such as turbulence intensities, spectra, length scales and coherence functions. These relationships are partly based on theoretical considerations and partly on empirical fits to a wide range of observations from many researchers taken in various conditions and in various locations.

In the neutral atmosphere, the boundary layer properties depend mainly on the surface roughness and the Coriolis effect. The surface roughness is characterised by the roughness length z_o . Typical values of z_o are shown in Table 2.1.

The Coriolis parameter f is defined as

$$f = 2\Omega \sin(|\lambda|) \quad (2.7)$$

where Ω is the angular velocity of the earth's rotation, and λ is the latitude. In temperate latitudes, the height of the boundary layer is given by

$$h = u^*/(6f) \quad (2.8)$$

Table 2.1 Typical surface roughness lengths

Type of Terrain	Roughness length z_0 (m)
Cities, forests	0.7
Suburbs, wooded countryside	0.3
Villages, countryside with trees and hedges	0.1
Open farmland, few trees and buildings	0.03
Flat grassy plains	0.01
Flat desert, rough sea	0.001

but it is clear from the division that this and the subsequent derivations cannot be valid at the equator, where $h = 0$, so a pragmatic recommendation is to use a latitude of 22.5° for all tropical regions. Here u^* is known as the friction velocity, given by

$$u^*/\overline{U}(z) = \kappa/[\ln(z/z_0) + \Psi] \quad (2.9)$$

where κ is the von Karman constant (approximately 0.4), z is the height above ground and z_0 is the surface roughness length. Ψ is a function which depends on stability: it is negative for unstable conditions, giving rise to low wind shear; and positive for stable conditions, giving high wind shear. For neutral conditions, ESDU (1985) gives $\Psi = 34.5 fz/u^*$, which is small compared to $\ln(z/z_0)$ for situations of interest here. If Ψ is ignored, the wind shear is then given by a logarithmic wind profile:

$$\overline{U}(z) \propto \ln(z/z_0) \quad (2.10)$$

A power law approximation,

$$\overline{U}(z) \propto z^\alpha \quad (2.11)$$

is often used, where the exponent α is typically about 0.14, but varies with the type of terrain. However, the value of α should also depend on the height interval over which the expression is applied, making this approximation less useful than the logarithmic profile.

However, the wind turbine design standards typically specify that a given exponent should be used; the IEC and GL standards, for example, specify an exponent of $\alpha = 0.20$ for normal wind conditions onshore and $\alpha = 0.14$ for normal wind conditions offshore. Both standards specify an exponent of $\alpha = 0.11$ for extreme wind conditions (onshore and offshore).

If there is a change in the surface roughness, the wind shear profile changes gradually downwind of the transition point, from the original to the new profile. Essentially, a new boundary layer starts, and the height of the boundary between the new and old boundary layers increases from zero at the transition point until the new boundary layer is fully established. The calculation of wind shear in the transition zone is covered by, for example, Cook (1985).

By combining Equations 2.8 and 2.9, we obtain the wind speed at the top of the boundary layer as

$$\bar{U}(h) = \frac{u^*}{\kappa} \left(\ln \left(\frac{u^*}{f z_0} \right) - \ln 6 + 5.75 \right) \quad (2.12)$$

This is similar to the so-called geostrophic wind speed, G , which is the notional wind speed driving the boundary layer as calculated from the pressure field. The geostrophic wind speed is given by

$$G = \frac{u^*}{\kappa} \sqrt{\left[\ln \left(\frac{u^*}{f z_0} \right) - A \right]^2 + B^2} \quad (2.13)$$

where, for neutral conditions, $A = \ln 6$ and $B = 4.5$. This relationship is often referred to as the geostrophic drag law.

The effect of surface roughness is not only to cause the wind speed to decrease closer to the ground. There is also a change in direction between the ‘free’ pressure-driven geostrophic wind and the wind close to the ground. Although the geostrophic wind is driven by the pressure gradients in the atmosphere, Coriolis forces act to force the wind to flow at right angles to the pressure gradient, causing a characteristic circulating pattern. Thus, in the northern hemisphere, wind flowing from high pressure in the south to low pressure in the north will be forced eastwards by Coriolis effects, in effect to conserve angular momentum on the rotating earth. The result is that the wind circulates anti-clockwise around low-pressure areas and clockwise around high-pressure areas, or the other way round in the southern hemisphere. Close to the ground, these flow directions are modified due to the effect of surface friction. The total direction change, α , from the geostrophic to the surface wind is given by

$$\sin \alpha = \frac{-B}{\sqrt{\left[\ln \left(\frac{u^*}{f z_0} \right) \right]^2 + B^2}} \quad (2.14)$$

2.6.3 Turbulence intensity

The turbulence intensity in the neutral atmosphere clearly depends on the surface roughness. For the longitudinal component, the standard deviation σ_u is approximately constant with height, so the turbulence intensity decreases with height. More precisely, the relationship $\sigma_u \approx 2.5u^*$ may be used to calculate the standard deviation, with the friction velocity u^* calculated as in the previous section. More recent work (ESDU, 1985) suggests a variation given by

$$\sigma_u = \frac{75\eta(0.538 + 0.09 \ln(z/z_0))^p u^*}{1 + 0.156 \ln(u^*/f \cdot z_0)} \quad (2.15)$$

where

$$\eta = 1 - 6fz/u^* \quad (2.16)$$

$$p = \eta^{16} \quad (2.17)$$

This approximates to $\sigma_u = 2.5u^*$ close to the ground, but gives larger values at greater heights. The longitudinal turbulence intensity is then

$$I_u = \sigma_u / \overline{U} \quad (2.18)$$

The lateral (v) and vertical (w) turbulence intensities are given (ESDU, 1985) by

$$I_v = \frac{\sigma_v}{\overline{U}} = I_u \left(1 - 0.22 \cos^4 \left(\frac{\pi z}{2h} \right) \right) \quad (2.19)$$

$$I_w = \frac{\sigma_w}{\overline{U}} = I_u \left(1 - 0.45 \cos^4 \left(\frac{\pi z}{2h} \right) \right) \quad (2.20)$$

Note that specific values of turbulence intensity for use in design calculations are prescribed in some of the standards used for wind turbine design calculations, and these may not always correspond with the above expressions. For example, the now superseded Danish standard (DS472, 1992) specified

$$I_u = 1.0 / \ln(z/z_0) \quad (2.21)$$

with $I_v = 0.8 I_u$ and $I_w = 0.5 I_u$.

The IEC edition 2 standard (IEC, 1999) gives

$$I_u = I_{15}(a + 15/\overline{U})/(a + 1) \quad (2.22)$$

where $I_{15} = 0.18$ for ‘higher turbulence sites’ and 0.16 for ‘lower turbulence sites’, with corresponding values of a of 2 and 3 respectively. For the lateral and vertical components, a choice is allowed: either $I_v = 0.8 I_u$ and $I_w = 0.5 I_u$, or an isotropic model with $I_u = I_v = I_w$.

The IEC edition 3 standard (IEC, 2005) specifies

$$I_u = I_{ref}(0.75 + 5.6/\overline{U}) \quad (2.23)$$

where $I_{ref} = 0.16, 0.14$ or 0.12 depending on the wind class. For lateral and vertical components, I_v must be at least $0.7 I_u$, and I_w at least $0.5 I_u$. Standard deviations are assumed constant with height, so the turbulence intensity will change with height as the mean wind speed changes due to wind shear.

The earlier Germanischer Lloyd rules (GL, 1993) simply specified 20% turbulence intensity, but the later edition (GL, 2003) follows IEC edition 2.

Figure 2.4 shows example longitudinal turbulence intensities for the GL, IEC and Danish standards. The low value for the Danish standard is for 90 m height with roughness length 0.01 m; the high value is for 30 m height with roughness length 0.3 m. The high values for IEC editions 2 and 3 are the same, but the low value is significantly lower in edition 3.

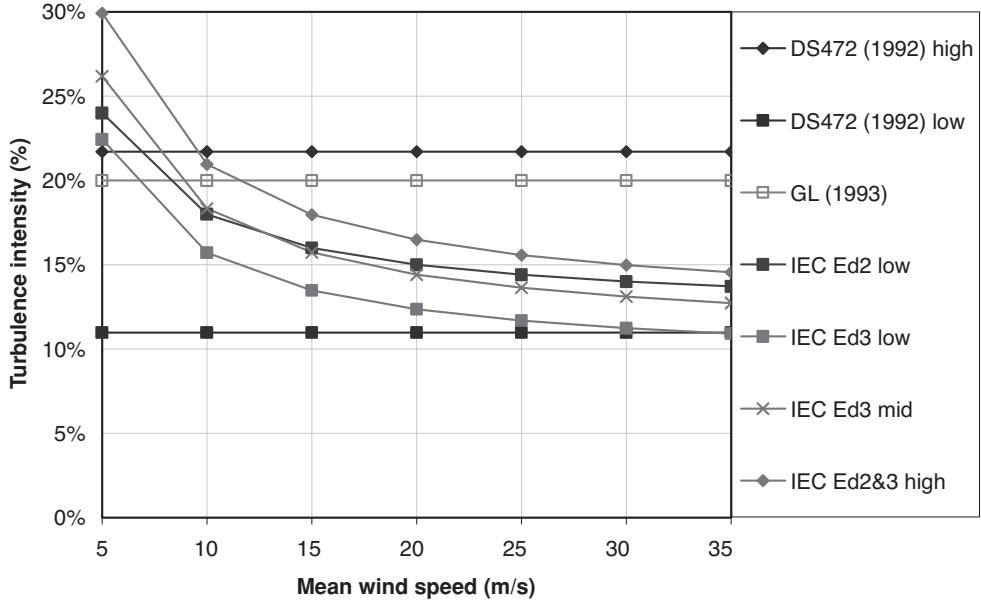


Figure 2.4 Turbulence intensities according to various standards

2.6.4 Turbulence spectra

The spectrum of turbulence describes the frequency content of wind speed variations. According to the Kolmogorov law, the spectrum must approach an asymptotic limit proportional to $n^{-5/3}$ at high frequency (here n denotes the frequency, in Hz). This relationship is based on the decay of turbulent eddies to higher and higher frequencies as turbulent energy is dissipated as heat.

Two alternative expressions for the spectrum of the longitudinal component of turbulence are commonly used, both tending to this asymptotic limit. These are the Kaimal and the von Karman spectra, which take the following forms:

$$\text{Kaimal: } \frac{nS_u(n)}{\sigma_u^2} = \frac{4nL_{1u}/\bar{U}}{(1 + 6nL_{1u}/\bar{U})^{5/3}} \quad (2.24)$$

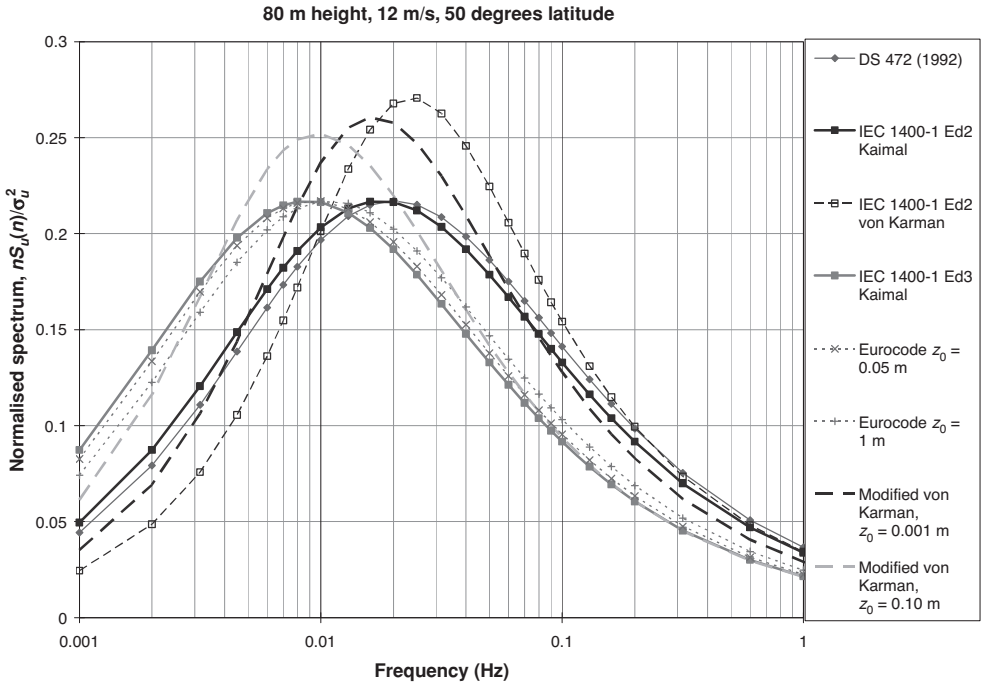
$$\text{von Karman: } \frac{nS_u(n)}{\sigma_u^2} = \frac{4nL_{2u}/\bar{U}}{(1 + 70.8(nL_{2u}/\bar{U})^2)^{5/6}} \quad (2.25)$$

where $S_u(n)$ is the autospectral density function for the longitudinal component and L_{1u} and L_{2u} are length scales. In order for these two forms to have the same high frequency asymptotic limit, these length scales must be related by the ratio $(36/70.8)^{-5/4}$, that is, $L_{1u} = 2.329 L_{2u}$. The appropriate length scales to use are discussed in the next section.

According to Petersen *et al.* (1998), the von Karman spectrum gives a good description for turbulence in wind tunnels, although the Kaimal spectrum may give a better fit to empirical observations of atmospheric turbulence. Nevertheless the von Karman spectrum is often used for consistency with analytical expressions for the correlations. The length scale L_{2u} is identified as the integral length scale of the longitudinal component in the longitudinal direction, denoted ^xL_u and defined as $\int_0^\infty \kappa(r_x) dr_x$ where $\kappa(r_x)$ is the cross-correlation function between the turbulence component u at two points separated longitudinally by a distance r_x and measured simultaneously (similar definitions apply to the integral length scales of the longitudinal component of turbulence in the lateral and vertical directions, ^yL_u and ^zL_u , which are used in the definitions of cross spectra below, and also to the integral length scales of the lateral and vertical components in the three directions). It is important to recognise that the power spectra and accompanying length scales are theoretical constructs, and attempts to fit them to real atmospheric data result in semi-empirical models in which the length scales may not be fully consistent with theory.

The Kaimal spectrum has a lower, broader peak than the von Karman spectrum: see Figures 2.5 and 2.6. More recent work suggests that the von Karman spectrum gives a good representation of atmospheric turbulence above about 150 m, but has some deficiencies at lower altitudes. Several modifications have been suggested (Harris, 1990) and a modified von Karman spectrum of the following form is recommended (ESDU, 1985):

$$\frac{nS_u(n)}{\sigma_u^2} = \beta_1 \frac{2.987nL_{3u}/\bar{U}}{(1 + (2\pi nL_{3u}/\bar{U})^2)^{5/6}} + \beta_2 \frac{1.294nL_{3u}/\bar{U}}{(1 + (\pi nL_{3u}/\bar{U})^2)^{5/6}} F_1 \quad (2.26)$$



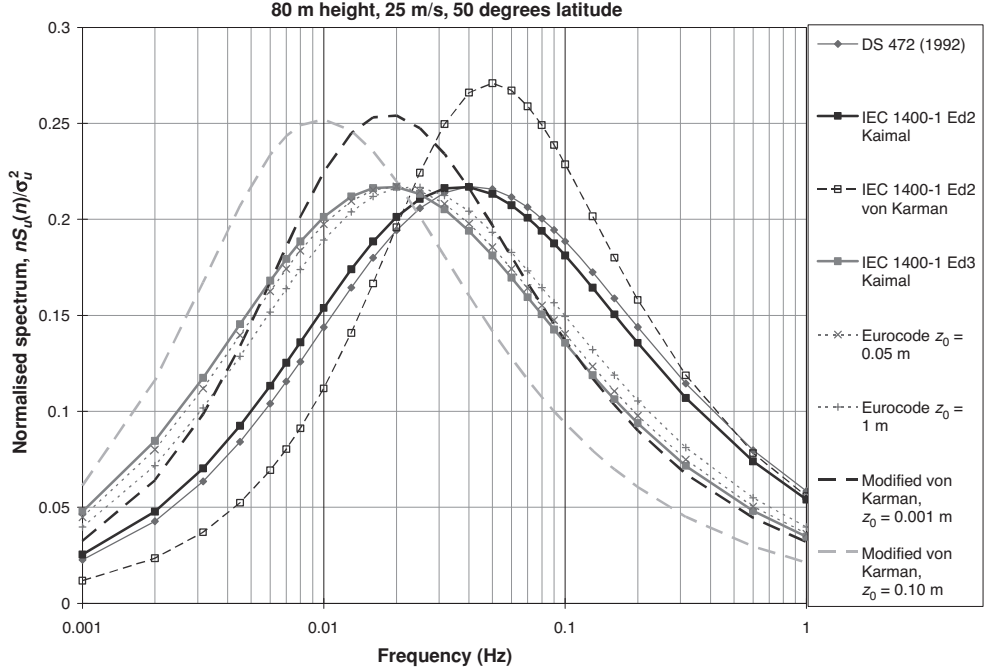


Figure 2.6 Comparison of spectra at 25 m/s

All three of these spectra have corresponding expressions for the lateral and vertical components of turbulence. The Kaimal spectra have the same form as for the longitudinal component but with different length scales, L_{1v} and L_{1w} respectively. The von Karman spectrum for the i component ($i = v$ or w) is

$$\frac{nS_i(n)}{\sigma_i^2} = \frac{4(nL_{2i}/\bar{U})(1 + 755.2(nL_{2i}/\bar{U})^2)}{(1 + 283.2(nL_{2i}/\bar{U})^2)^{11/6}} \quad (2.27)$$

where $L_{2v} = {}^xL_v$ and $L_{2w} = {}^xL_w$. For the modified von Karman spectrum of equation 2.26 it is

$$\frac{nS_i(n)}{\sigma_i^2} = \beta_1 \frac{2.987(nL_{3i}/\bar{U})(1 + \frac{8}{3}(4\pi nL_{3i}/\bar{U})^2)}{(1 + (4\pi nL_{3i}/\bar{U})^2)^{11/6}} + \beta_2 \frac{1.294nL_{3i}/\bar{U}}{(1 + (2\pi nL_{3i}/\bar{U})^2)^{5/6}} F_{2i} \quad (2.28)$$

2.6.5 Length scales and other parameters

To use the spectra defined above, it is necessary to define the appropriate length scales. Additional parameters β_1 , β_2 , F_1 , and F_2 are also required for the modified von Karman model.

The length scales are dependent on the surface roughness z_0 , as well as on the height above ground (z): proximity to the ground constrains the size of turbulent eddies and, thus, reduces the length scales. If there are many small obstacles on the ground of typical height z' , the height above ground should be corrected for the effect of these by assuming that the effective ground

surface is at a height $z' = 2.5z_0$ (ESDU, 1975). Far enough above the ground (i.e. for z greater than some height z_i) the turbulence is no longer constrained by the proximity of the surface and becomes isotropic. According to ESDU (1975), $z_i = 1000z_0^{0.18}$ and above this height ${}^xL_u = 280$ m, and ${}^yL_u = {}^zL_u = {}^xL_v = {}^zL_v = 140$ m. Even for very small roughness lengths z_0 , the isotropic region is well above the height of a wind turbine and the following corrections for $z < z_i$ should be applied:

$$\begin{aligned} {}^xL_u &= 280(z/z_i)^{0.35} \\ {}^yL_u &= 140(z/z_i)^{0.38} \\ {}^zL_u &= 140(z/z_i)^{0.45} \\ {}^xL_v &= 140(z/z_i)^{0.48} \\ {}^zL_v &= 140(z/z_i)^{0.55} \end{aligned} \quad (2.29)$$

together with ${}^xL_w = {}^yL_w = 0.35z$ (for $z < 400$ m). Expressions for ${}^yL_v$ and ${}^zL_w$ are not given. The length scales ${}^xL_u$, ${}^xL_v$ and ${}^xL_w$ can be used directly in the von Karman spectra. For the Kaimal spectra we already have $L_{1u} = 2.329 {}^xL_u$, and to achieve the same high frequency asymptotes for the other components we also have $L_{1v} = 3.2054 {}^xL_v$, $L_{1w} = 3.2054 {}^xL_w$.

Later work based on measurements for a greater range of heights (Harris, 1990; ESDU, 1985) takes into account an increase in length scales with the thickness of the boundary layer, h , which also implies a variation of length scales with mean wind speed. This yields more complicated expressions for the nine length scales in terms of z/h , σ_u/u^* and the Richardson number $u^*/(fz_0)$.

Note that some of the standards used for wind turbine loading calculations prescribe that certain turbulence spectra and/or length scales are to be used. These are often simplified compared to the expressions given above. Thus, the Danish standard (DS 472, 1992) specifies a Kaimal spectrum with

$$\begin{aligned} L_{1u} &= 150 \text{ m, or } 5z \text{ for } z < 30 \text{ m} \\ L_{1v} &= 0.3 L_{1u} \\ L_{1w} &= 0.1 L_{1u} \end{aligned} \quad (2.30)$$

while the IEC edition 2 standard (IEC, 1999) gives a choice between a Kaimal model with

$$\begin{aligned} A_1 &= 21 \text{ m, or } 0.7z \text{ for } z < 30 \text{ m} \\ L_{1u} &= 8.1 A_1 = 170.1 \text{ m, or } 5.67z \text{ for } z < 30 \text{ m} \\ L_{1v} &= 2.7 A_1 = 0.3333 L_{1u} \\ L_{1w} &= 0.66 A_1 = 0.08148 L_{1u} \end{aligned} \quad (2.31)$$

and an isotropic von Karman model with

$$\begin{aligned} {}^xL_u &= 73.5 \text{ m, or } 2.45z \text{ for } z < 30 \text{ m} \\ {}^xL_v &= {}^xL_w = 0.5 {}^xL_u \end{aligned} \quad (2.32)$$

The IEC edition 3 standard (IEC, 2005) gives a choice between a slightly different Kaimal model and the Mann model. The Kaimal model has the same form (equation 2.24) but with

$$\begin{aligned} \Lambda_1 &= 42 \text{ m, or } 0.7z \text{ for } z < 60 \text{ m} \\ L_{1u} &= 8.1 \Lambda_1 = 340.2 \text{ m, or } 5.67z \text{ for } z < 60 \text{ m} \\ L_{1v} &= 2.7 \Lambda_1 = 0.3333 L_{1u} \\ L_{1w} &= 0.66 \Lambda_1 = 0.08148 L_{1u} \end{aligned} \quad (2.33)$$

The Mann model has a rather different form and is described below in Section 2.6.8.

The Eurocode (2005) standard for wind loading specifies a longitudinal spectrum of Kaimal form with $L_{1u} = 1.7 L_i$, where

$$L_i = 300(z/200)^\alpha \quad (2.34)$$

for $z < 200 \text{ m}$, with $\alpha = 0.67 + 0.05 \ln(z_0)$. This standard is used for buildings, but not usually for wind turbines.

With so many variables, it is difficult to present a concise comparison of the different spectra, so a few examples are presented in the figures below. These are plots of the normalised longitudinal spectrum $n S_u(n)/\sigma_u^2$ against frequency, which means that the area under the curve is representative of the fraction of total variance in any given frequency range. A typical hub height of 80 m has been used, with 50° latitude assumed for the modified von Karman model.

Figure 2.5 shows spectra for a typical rated wind speed of 12 m/s. The IEC edition 2 Kaimal spectrum is clearly very similar to DS 472, while the IEC edition 3 spectrum has clearly moved to lower frequencies, being now more consistent with Eurocode (in fact identical for 80 m height and $z_0 = 0.01 \text{ m}$). Note the characteristic difference between the Kaimal and von Karman spectra, the latter being rather more sharply peaked. The modified von Karman spectrum is intermediate in shape; with a very small roughness length the peak is at a similar frequency to the IEC edition 2 spectra, but with higher roughness length it comes closer to edition 3.

Figure 2.6 shows a similar figure for a typical cut-out wind speed of 25 m/s. All peaks have moved to higher frequency as expected, but the modified von Karman now matches IEC edition 3 better with the very small roughness length.

2.6.6 Asymptotic limits

Other spectra may also be used, but to comply with the IEC standard the high frequency asymptotic behaviour must tend to the following relationship:

$$\frac{S_u(n)}{\sigma_u^2} \xrightarrow{n \rightarrow \infty} .005 \left(\frac{\Lambda_1}{U} \right)^{-2/3} n^{-5/3} \quad (2.35)$$

with Λ_1 defined as above: it is a function only of height above ground, but differs between edition 2 and edition 3 for heights above 30 m. Expressing this as

$$\frac{S_u(n)}{\sigma_u^2} \xrightarrow{n \rightarrow \infty} A \bar{U}^{-2/3} n^{-5/3} \quad (2.36)$$

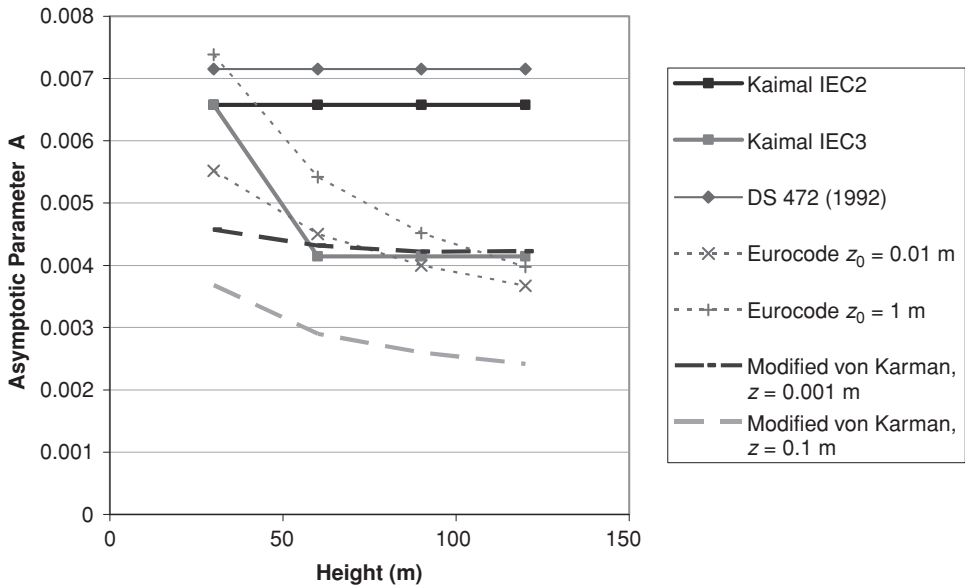


Figure 2.7 Some asymptotic limits

the asymptotic parameter A can then be compared for different spectra, as in Figure 2.7. This shows how the DNV asymptote is similar to IEC edition 2 (both Kaimal and von Karman spectra in that standard have the same asymptote), but the asymptote becomes much lower for IEC edition 3 above 30 m height, and is now more comparable with Eurocode. The ESDU modified von Karman spectrum is more difficult to characterise because the asymptote now varies also with wind speed, surface roughness and geographical latitude. Figure 2.7 shows the results for 20 m/s wind speed and 50° latitude for two different roughness lengths. The asymptote can be made to match the IEC edition 3 specification, but only by choosing a very small roughness length – even smaller to match edition 2. However, if such a low roughness length is selected, the turbulence intensity implied by the ESDU model will be much smaller than that required by the standards. It is common practice when using the ESDU model to adjust the surface roughness at each wind speed until the turbulence intensity matches the standard, although this clearly makes little physical sense. Obviously it is not possible to adjust the roughness to match both the required asymptote and the required turbulence intensity at the same time. Compared to the physical model, therefore, the standards are probably conservative, which is surely to be expected. It can also be argued that the physical model is valid for flat terrain, while many wind farms are built in complex terrain where the turbulence intensities will indeed be higher, and the length scales shorter.

Note also that the IEC edition 3 standard further specifies that $S_v(n) = S_w(n) = \frac{4}{3} S_u(n)$ in the high frequency limit.

2.6.7 Cross-spectra and coherence functions

The turbulence spectra presented in the preceding sections describe the temporal variation of each component of turbulence at any given point. However, as the wind turbine blade sweeps out its trajectory, the wind speed variations it experiences are not well represented by these

single-point spectra. The spatial variation of turbulence in the lateral and vertical directions is clearly important, since this spatial variation is ‘sampled’ by the moving blade and, thus, contributes to the temporal variations experienced by it.

In order to model these effects, the spectral description of turbulence must be extended to include information about the cross-correlations between turbulent fluctuations at points separated laterally and vertically. Clearly these correlations decrease as the distance separating two points increases. The correlations are also smaller for high frequency than for low frequency variations. They can, therefore, be described by ‘coherence’ functions, which describe the correlation as a function of frequency and separation. The coherence $C(\Delta r, n)$ is defined by

$$C(\Delta r, n) = \frac{|S_{12}(n)|}{\sqrt{S_{11}(n)S_{22}(n)}} \quad (2.37)$$

where n is frequency, $S_{12}(n)$ is the cross-spectrum of variations at the two points separated by Δr , and $S_{11}(n)$ and $S_{22}(n)$ are the spectra of variations at each of the points (usually these can be taken as equal).

Starting from von Karman spectral equations, and assuming Taylor’s frozen turbulence hypothesis, an analytical expression for the coherence of wind speed fluctuations can be derived. Accordingly for the longitudinal component at points separated by a distance Δr perpendicular to the wind direction, the coherence $C_u(\Delta r, n)$ is:

$$C_u(\Delta r, n) = 0.994 \left(A_{5/6}(\eta_u) - \frac{1}{2} \eta_u^{5/3} A_{1/6}(\eta_u) \right) \quad (2.38)$$

Here $A_j(x) = x^j K_j(x)$ where K is a fractional order modified Bessel function, and

$$\eta_u = \Delta r \sqrt{\left(\frac{0.747}{L_u} \right)^2 + \left(c \frac{2\pi n}{U} \right)^2} \quad (2.39)$$

with $c = 1$. L_u is a local length scale which can be defined as:

$$L_u(\Delta r, n) = 2f_u(n) \sqrt{\frac{({}^y L_u \Delta y)^2 + ({}^z L_u \Delta z)^2}{\Delta y^2 + \Delta z^2}} \quad (2.40)$$

where Δy and Δz are the lateral and vertical components of the separation Δr , and ${}^y L_u$ and ${}^z L_u$ are the lateral and vertical length scales for the longitudinal component of turbulence. Normally $f_u(n) = 1$, but ESDU (1975) suggests a modification at low frequencies where the wind becomes more anisotropic, with $f_u(n) = \text{MIN}(1.0, 0.04n^{-2/3})$.

The IEC (1999) edition 2 standard allows only an isotropic turbulence model to be used if the von Karman spectrum is used, in which ${}^x L_u = 2{}^y L_u = 2{}^z L_u$, and then $L_u = {}^x L_u$, and $f_u(n) = 1$.

The modified von Karman model described in Equation 2.26 also uses $f_u(n) = 1$, but the factor c in Equation 2.39 is modified instead (ESDU, 1985).

For the lateral and vertical components, the corresponding equations are as follows. The analytical derivation for the coherence, based as before on the von Karman spectrum and Taylor's hypothesis, is

$$C_i(\Delta r, n) = \frac{0.597}{2.869\gamma_i^2 - 1} [4.781\gamma_i^2 A_{5/6}(\eta_i) - A_{11/6}(\eta_i)] \quad (2.41)$$

for $i = u$ or v , where η_i is calculated as in Equation 2.39 but with L_u replaced by L_v or L_w respectively, and with $c = 1$. Also

$$\gamma_i = \frac{\eta_i L_i(\Delta r, n)}{\Delta r} \quad (2.42)$$

and L_v and L_w are given by expressions analogous to equation 2.40.

The expressions for spatial coherence in Equations 2.38 and 2.41 above are derived theoretically from the von Karman spectrum, although there are empirical factors in some of the expressions for length scales, for example. If a Kaimal rather than a von Karman spectrum is used as the starting point, there are no such relatively straightforward analytical expressions for the coherence functions. In this case a simpler, and purely empirical, exponential model of coherence is often used. The IEC (1999) edition 2 standard, for example, gives the following expression for the coherence of the longitudinal component of turbulence:

$$C_u(\Delta r, n) = \exp \left(-H \Delta r \sqrt{\left(\frac{0.12}{L_c} \right)^2 + \left(\frac{n}{U} \right)^2} \right) \quad (2.43)$$

where $H = 8.8$ and $L_c = L_u$. This can also be written

$$C_u(\Delta r, n) = \exp(-1.4\eta_u) \quad (2.44)$$

with η_u as in Equation 2.39.

The standard also states that this may also be used with the von Karman model, as an approximation to Equation 2.38. The standard does not specify the coherence of the other two components to be used in conjunction with the Kaimal model, so the following expression is often used:

$$C_v(\Delta r, n) = C_w(\Delta r, n) = \exp \left(-H \Delta r \frac{n}{U} \right) \quad (2.45)$$

In the IEC (2004) edition 3 standard a slightly modified form is specified, in which $H = 12$ and $L_c = L_{1u}$.

The three turbulence components are usually assumed to be independent of one another. This is a reasonable assumption, although it ignores the effect of Reynolds stresses which result in a small correlation between the longitudinal and vertical components near to the ground, an effect which is captured by the Mann model described in Section 2.6.8.

Clearly there are significant discrepancies between the various recommended spectra and coherence functions. Also these wind models are applicable to flat sites, and there is only limited understanding of the way in which turbulence characteristics change over hills and in complex terrain. Given the important effect of turbulence characteristics on wind turbine loading and performance, this is clearly an area in which there is scope for further research.

2.6.8 The Mann model of turbulence

Alongside the Kaimal model, edition 3 of the IEC standard (IEC, 2005) gives the option to use a rather different form of turbulence model developed by Mann (1994, 1998). The other models described above make use of a one-dimensional fast Fourier transform (FFT) to generate time histories from spectra, applied to each turbulence component independently. In contrast, the Mann model is based on a three-dimensional spectrum tensor representation of the turbulence, and one three-dimensional FFT is then used to generate all three components of turbulence simultaneously. The three-dimensional spectrum tensor is derived from rapid distortion of isotropic turbulence by a uniform mean vertical velocity shear, and the method, therefore, does not treat the three turbulence components as independent, but embodies in particular the correlation between the longitudinal and vertical components resulting from Reynolds stresses. The spectral density for any three-dimensional wavenumber vector is derived, and all three components of turbulence are then generated simultaneously by summing a set of such wavenumber vectors, each with the appropriate amplitude and a random phase.

This is in many ways a rather elegant approach, but in practice there are some computational limitations which can make it difficult to use. The summation requires a three-dimensional FFT to achieve reasonable computation time. The number of points in the longitudinal, lateral and vertical directions must be a power of two for efficient FFT computation. In the longitudinal direction, the number of points is determined by the length of time history required and the maximum frequency of interest, and is, therefore, typically at least 1024. The maximum wavelength used is the length of the turbulence history to be generated (i.e. the mean wind speed multiplied by duration of the required time series) and the minimum wavelength is twice the longitudinal spacing of points (which is the mean wind speed divided by the maximum frequency of interest). In the lateral and vertical directions, a much smaller number of points must be used, perhaps as low as 32, depending on available computer memory. The maximum wavelength must be significantly greater than the rotor diameter, since the solution is spatially periodic, with period equal to the maximum wavelength in each direction. The number of FFT points then determines the minimum wavelength in these directions. With a realistic number of points, the resulting turbulence spectra are deficient at the high frequency end (Veldkamp, 2006). Mann (1998) suggests that this may be realistic, because it represents averaging of the turbulence over finite volumes of space which is appropriate for practical engineering applications. However, a practical simulation tool will perform all necessary spatial averaging in any case, and so the high frequency variations are really lost. Mann (1998) does suggest a remedy for this, but in practice it is extremely intensive computationally.

2.7 Gust wind speeds

It is often useful to know the maximum gust speed which can be expected to occur in any given time interval. This is usually represented by a gust factor G , which is the ratio of the gust wind speed to the hourly mean wind speed. G is obviously a function of the turbulence intensity, and it also clearly depends on the duration of the gust – thus, the gust factor for a one-second gust will be larger than for a three-second gust, since every three-second gust has within it a higher one-second gust.

While it is possible to derive expressions for gust factors starting from the turbulence spectrum (Greenway, 1979; ESDU, 1983), an empirical expression due to Weirringa (1973)

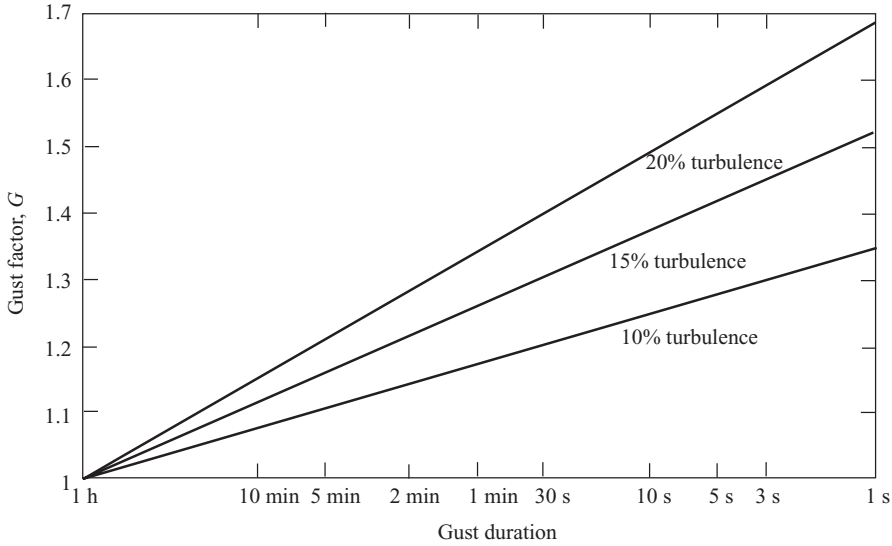


Figure 2.8 Gust factors calculated from equation 2.46

is often used as it is much simpler, and agrees well with theoretical results. Accordingly the t -second gust factor is given by

$$G(t) = 1 + 0.42I_u \ln \frac{3600}{t} \quad (2.46)$$

where I_u is the longitudinal turbulence intensity. Figure 2.8 shows the gust factors for several different turbulence intensities and gust durations calculated according to this expression.

2.8 Extreme wind speeds

In addition to the foregoing descriptions of the average statistical properties of the wind, it is clearly of interest to be able to estimate the long-term extreme wind speeds which might occur at a particular site.

A probability distribution of hourly mean wind speeds such as the Weibull distribution will yield estimates of the probability of exceedance of any particular level of hourly mean wind speed. However, when used to estimate the probability of extreme winds, an accurate knowledge of the high wind speed tail of the distribution is required, and this will not be very reliable since almost all of the data which was used to fit the parameters of the distribution will have been recorded at lower wind speeds. Extrapolating the distribution to higher wind speeds cannot be relied upon to give an accurate result.

Fisher and Tippett (1928) and Gumbel (1958) have developed a theory of extreme values which is useful in this context. If a measured variable (such as hourly mean wind speed $\bar{U}$) conforms to a particular cumulative probability distribution $F(\bar{U})$, that is, $F(\bar{U}) \rightarrow 1$ as $\bar{U}$ increases, then the peak values of hourly mean wind speed in a given period (a year, for example) will have a cumulative probability distribution of F^N , where N is the number of

independent peaks in the period. In the UK, for example, according to Cook (1982) there are about 100 independent wind speed peaks per year, corresponding to the passage of individual weather systems. Thus, if $F(\bar{U}) = 1 - \exp(-(\bar{U}/c)^k)$ as for a Weibull distribution, the wind speed peaks in one year will have a cumulative probability distribution given approximately by $[1 - \exp(-(\bar{U}/c)^k)]^{100}$. However, as indicated above, this is unlikely to give accurate estimates for extreme hourly means, since the high-wind tail of the distribution cannot be considered to be reliably known. However, Fisher and Tippett (1928) demonstrated that for any cumulative probability distribution function which converges towards unity at least exponentially (as is usually the case for wind speed distributions, including the Weibull distribution), the cumulative probability distribution function for extreme values $\hat{U}$ will always tend towards an asymptotic limit

$$F(\hat{U}) = \exp(-\exp(-a(\hat{U} - U')))$$

as the observation period increases. U' is the most likely extreme value, or the *mode* of the distribution, while $1/a$ represents the width or spread of the distribution and is termed the *dispersion*.

This makes it possible to estimate the distribution of extreme values based on a fairly limited set of measured peak values, for example a set of measurements of the highest hourly mean wind speeds $\hat{U}$ recorded during each of N storms. The N measured extremes are ranked in ascending order, and an estimate of the cumulative probability distribution function is obtained as

$$\tilde{F}(\hat{U}) \cong \frac{m(\hat{U})}{N + 1}$$

where $m(\hat{U})$ is the rank, or position in the sequence (starting with the lowest), of the observation $\hat{U}$. Then a plot of $-\ln(-\ln(\tilde{F}(\hat{U})))$ against $\hat{U}$ is used to estimate the mode U' and dispersion $1/a$ by fitting a straight line to the data points. This is the method due to Gumbel. Lieblein (1974) has developed a numerical technique which gives a less biased estimate of U' and $1/a$ than a simple least squares fit to a Gumbel plot.

Having made an estimate of the cumulative probability distribution of extremes $F(\hat{U})$, the M -year extreme hourly mean wind speed can be estimated as the value of $\hat{U}$ corresponding to the probability of exceedance $F = 1 - 1/M$.

According to Cook (1985), a better estimate of the probability of extreme winds is obtained by fitting a Gumbel distribution to extreme values of wind speed squared. This is because the cumulative probability distribution function of wind speed squared is closer to exponential than the distribution of wind speed itself, and it converges much more rapidly to the Gumbel distribution. Therefore, by using this method to predict extreme values of wind speed squared, more reliable estimates can be obtained from a given number of observations.

2.8.1 Extreme winds in standards

The design of wind turbines must allow them to withstand extremes of wind speed, as well as respond well to the more 'typical' conditions described above. Therefore, the various standards also specify the extremes of wind speed which must be designed for. This includes extreme mean wind speeds as well as various types of severe gust.

Extreme conditions may be experienced with the machine operating, parked or idling with or without various types of fault or grid loss, or during a particular operation such as a shut-down event. The extreme wind conditions may be characterised by a ‘return time’: for example, a 50-year gust is one which is so severe that it can be expected to occur on average only once every 50 years. It would be reasonable to expect a turbine to survive such a gust, provided there was no fault on the turbine.

It is always possible that the turbine happens to be shut down on account of a fault when a gust occurs. If the fault impairs the turbine’s ability to cope with a gust, for example if the yaw system has failed and the turbine is parked at the wrong angle to the wind, then the turbine may have to withstand even greater loads. However, the probability of the most extreme gusts occurring at the same time as a turbine fault is very small, and so it is usual to specify that a turbine with a fault need only be designed to withstand, for example, the annual extreme gust and not the 50-year extreme gust.

For this to be valid, it is important that the faults in question are not correlated with extreme wind conditions. Grid loss is not considered to be a fault with the turbine, and is actually quite likely to be correlated with extreme wind conditions.

Clearly the extreme wind speeds and gusts (both in terms of magnitude and shape) may be quite site-specific. They may differ considerably between flat coastal sites and rugged hill tops, for example.

The IEC standard, for example, specifies a ‘reference wind speed’ V_{ref} which is five times the annual mean wind speed. The 50-year extreme wind speed is then given by 1.4 times V_{ref} at hub height, and varying with height using a power law exponent of 0.11. The annual extreme wind speed is taken as 75% of the 50-year value in the 1999 edition 2 standard, or 80% in the 2004 edition 3 standard.

The IEC edition 2 standard goes on to define a number of transient events which the turbine must be designed to withstand. These include:

- Extreme operating gust (EOG): a decrease in speed, followed by a steep rise, a steep drop and a rise back to the original value. The gust amplitude and duration vary with the return period.
- Extreme direction change (EDC): this is a sustained change in wind direction, following a cosine-shaped curve. The amplitude and duration of the change once again depend on the return period.
- Extreme coherent gust (ECG): this is a sustained change in wind speed, again following a cosine-shaped curve with the amplitude and duration depending on the return period.
- Extreme coherent gust with direction change (ECD): simultaneous speed and direction transients similar to EDC and ECG.
- Extreme wind shear (EWS): a transient variation in the horizontal and vertical wind gradient across the rotor. The gradient first increases and then falls back to the initial level, following a cosine-shaped curve.

These transient events are deterministic gusts intended to represent the extreme turbulent variations which would be expected to occur at the specified return period. They are not intended to occur *in addition* to the normal turbulence described previously. Such deterministic coherent gusts, however, have little basis in terms of actual measured or theoretical wind characteristics. Therefore, in edition 3 of the standard, some of the extreme loads are estimated

instead by carrying out a large number of simulations and applying statistical extrapolation methods to the peak loads from each simulation. Appropriate probability distributions are fitted to the simulated peaks, and the 50-year extreme load is estimated from the tail of the distribution – see Section 5.14.

2.9 Wind speed prediction and forecasting

Because of the variable nature of the wind resource, the ability to forecast wind speed some time ahead is often valuable. Such forecasts fall broadly into two categories: predicting short-term turbulent variations over a time scale of seconds to minutes ahead, which may be useful for assisting with the operational control of wind turbines or wind farms, and longer-term forecasts over periods of a few hours or days, which may be useful for planning the deployment of other power stations on the network.

Short-term forecasts necessarily rely on statistical techniques for extrapolating the recent past, whereas the longer-term forecasts can make use of meteorological methods. A combination of meteorological and statistical forecasts can give very useful predictions of wind farm power output.

2.9.1 Statistical methods

The simplest statistical prediction is known as a ‘persistence’ forecast: the prediction is set equal to the last available measurement. In other words the last measured value is assumed to persist into the future without any change:

$$\hat{y}_k = y_{k-1}$$

where y_{k-1} is the measured value at step $k-1$ and $\hat{y}_k$ is the prediction for the next step.

A more sophisticated prediction might be some linear combination of the last n measured values, that is,

$$\hat{y}_k = \sum_{i=1}^n a_i y_{k-i}$$

This is known as an n^{th} order autoregressive model, or $\text{AR}(n)$. We can now define the prediction error at step k by

$$e_k = \hat{y}_k - y_k$$

and then use the recent prediction errors to improve the prediction:

$$\hat{y}_k = \sum_{i=1}^n a_i y_{k-i} + \sum_{j=1}^m b_j e_{k-j}$$

This is known as an n^{th} order autoregressive, m^{th} order moving average model, or $\text{ARMA}(n, m)$. This can be further extended to an ARMAX model, where the X stands for an ‘exogenous’ variable: another measured variable which is included in the prediction because it influences y .

The model parameters a_i , b_j can be estimated in various ways. A useful technique is the method of recursive least squares, or RLS (Ljung & Söderström, 1983). Estimates of the model parameters are updated on each timestep in such a way as to minimise the expected value of the sum of squares of the prediction errors. By including a so-called ‘forgetting factor’, the influence of older observations can be progressively reduced, leading to an adaptive estimation of the parameters, which will gradually change to accommodate variations in the statistical properties of the variable y .

Bossanyi (1985) investigated the use of ARMA models for wind speed predictions from a few seconds to a few minutes ahead, obtaining reductions in rms prediction errors of up to 20% when compared to a persistence forecast. The best results were obtained when predicting ten minutes ahead from one-minute data.

Kariniotakis *et al.* (1997) compare ARMA methods against a selection of more recent techniques such as neural network, fuzzy logic and wavelet-based methods. The fuzzy logic method is tentatively selected as giving the best predictions over periods of ten minutes to two hours, with improvements of 10–18% compared to persistence.

Nielsen and Madsen (1999) use an ARX model with recursive least squares to predict wind farm power output based on previous values of power output, and measured wind speed as an exogenous variable, supplemented by a function describing the diurnal variations of wind speed and by meteorological forecasts of wind speed and direction. Predictions up to 48 hours ahead are considered, and the inclusion of meteorological forecasts is shown to improve the predictions significantly, especially for the longer period forecasts.

2.9.2 Meteorological methods

As indicated in the previous section, much better predictions can be made by using meteorological forecasts than by using purely statistical methods, when predictions over timescales of a few hours or days are considered. Very sophisticated meteorological forecasts are available from highly detailed simulation models of the atmosphere, fed by many recorded observations of pressure, temperature, wind speed, etc. over wide areas of land and sea.

Landberg (1997, 1999) describes the use of such models to predict wind farm output, by extrapolating the large-scale wind predictions produced by these models down to the specific wind farm site. The geostrophic drag law and the logarithmic wind shear profile (Section 2.6.2) are used to extrapolate the wind forecasts down to ground level. Modifications to the flow resulting from the topography, the physical geography and surface roughness conditions in the area surrounding the wind farm are then modelled by the WA³P programme (Mortensen *et al.*, 1993). A turbine wake interaction model PARK (Sanderhoff, 1993) then takes account of wind direction in relation to the actual turbine positions to calculate wake losses, and finally a statistical model (as described in the previous section) combines the meteorological forecasts with recent measurements on the wind farm to give predictions of the energy output which are good enough to be useful in planning the deployment of other power stations on the network.

2.10 Turbulence in wakes and wind farms

As it extracts energy from the wind, a turbine leaves behind it a wake characterised by reduced wind speeds and increased levels of turbulence. Another turbine operating in this wake, or deep inside a wind farm where the effects of a number of wakes may be felt simultaneously,

will therefore produce less energy and suffer greater structural loading than a turbine operating in the free stream.

Immediately behind a turbine, its wake can crudely be considered as a region of reduced wind speed slightly larger in diameter than the turbine itself (Vermeulen, 1980). The reduction in velocity is directly related to the thrust coefficient of the turbine, since this determines the momentum extracted from the flow. As this reduced wind speed region convects downstream, the wind speed gradient between the wake and the free flow outside the wake results in additional shear-generated turbulence, which assists the transfer of momentum into the wake from the surrounding flow. Thus, the wake and the surrounding flow start to mix, and the region of mixing spreads inwards to the centre of the wake, as well as outwards to make the width of the wake increase. In this way, the velocity deficit in the wake is eroded and the wake becomes broader but shallower until the flow has fully recovered far downstream. The rate at which this occurs is dependent on the ambient turbulence level.

In addition to this shear-generated turbulence, the turbine itself generates additional turbulence directly, as a result of the tip vortices shed by the blades and the general disturbance to the flow caused by the blades, nacelle and tower. This ‘mechanical’ component of turbulence is of relatively high frequency and decays relatively quickly. Bossanyi (1983) develops a theoretical model which describes how this additional turbulence might decay: large eddies give rise to smaller ones, and so the turbulent energy moves to higher and higher frequencies until it is eventually dissipated as heat. The model predicts a faster rate of decay in low winds and in high ambient turbulence intensities.

Beyond the near wake region, once the shear-generated turbulence has reached the centre of the wake and started to erode the centreline velocity deficit, the mean velocity variation in the wake can be described well by an axisymmetric profile with a Gaussian cross-section. The development of the wake profile downstream of this point can be reasonably well predicted, for example using an eddy viscosity model (Ainslie, 1988). This is partly theoretical, derived from the Navier-Stokes equation of fluid flow, but with some empirical terms.

Theoretical models capable of predicting turbulence levels in the wake are less well developed. Quarton and Ainslie (1989) examined a number of different sets of wake turbulence measurements, both in wind tunnels using small wind turbine models or gauze simulators and behind full-size turbines in the free stream. An empirical formula for added turbulence I_+ at a downstream distance x from the turbine was found to give a good fit to the various measurements:

$$I_+ = 4.8C_T^{0.7}I_0^{0.68}(x/x_n)^{-0.57}$$

where C_T is the turbine thrust coefficient, I_0 the ambient turbulence intensity and x_n the length of the near wake region. Here I_0 and I_+ are expressed as percentages. On the basis of further work, an improved expression was subsequently proposed by Hassan (1992):

$$I_+ = 5.7C_T^{0.7}I_0^{0.68}(x/x_n)^{-0.96}$$

The added turbulence is defined as the square root of the additional wind speed variance normalised by the mean wind speed, that is,

$$I_+ = \sqrt{I_{\text{wake}}^2 - I_0^2}$$

where I_{wake} is the total wake turbulence intensity, at any given downstream distance.

The length of the near wake region, x_n , is calculated according to Vermeulen (1980) in terms of the rotor radius R and the thrust coefficient C_T as

$$x_n = \frac{nr_0}{\left(\frac{dr}{dx}\right)}$$

where

$$\begin{aligned} r_0 &= R\sqrt{\frac{m+1}{2}} \\ m &= \frac{1}{\sqrt{1-C_T}} \\ n &= \frac{\sqrt{0.214 + 0.144m} (1 - \sqrt{0.134 + 0.124m})}{(1 - \sqrt{0.214 + 0.144m})\sqrt{0.134 + 0.124m}} \end{aligned}$$

and dr/dx is the wake growth rate:

$$\begin{aligned} \frac{dr}{dx} &= \sqrt{\left(\frac{dr}{dx}\right)_\alpha^2 + \left(\frac{dr}{dx}\right)_m^2 + \left(\frac{dr}{dx}\right)_\lambda^2} \\ \left(\frac{dr}{dx}\right)_\alpha &= 2.5I_0 + 0.005 \end{aligned}$$

is the growth rate contribution due to ambient turbulence,

$$\left(\frac{dr}{dx}\right)_m = \frac{(1-m)\sqrt{1.49+m}}{(1+m)9.76}$$

is the contribution due to shear-generated turbulence, and

$$\left(\frac{dr}{dx}\right)_\lambda = 0.012B\lambda$$

is the contribution due to mechanical turbulence, where B is the number of blades and λ is the tip speed ratio.

Deep inside a wind farm, the reduction in wind speed and the increase in turbulence intensity are the result of the superposition of wakes from many upwind turbines. Frandsen and Thøgersen (1999) and Frandsen (2003) propose a model based on the geostrophic drag law which takes into account the additional ‘surface roughness’ caused by the turbines themselves. This leads to a formula for added turbulence above hub height:

$$I_{++} = \frac{0.36}{1 + 0.2\sqrt{s_1 s}/C_T}$$

where s_1 and s are the inter-turbine spacings, normalised by rotor diameter, within a row and between rows. Since this does not apply below hub height, the average added turbulence intensity I_+ is then calculated as

$$I_+ = \frac{1}{2} \left(I_0 + \sqrt{I_0^2 + I_{++}^2} \right)$$

This approach has been included in the IEC (2005) edition 3 standard as an informative Annex. However, this approach has not been thoroughly validated against measurements and no consensus has yet emerged on a sufficiently well validated formula for turbulence intensity within a wind farm for use in wind turbine design calculations.

2.11 Turbulence in complex terrain

Predicting the turbulence intensity and spectrum at a given point within an area of complex terrain is not straightforward. Hilly terrain upwind of the site in question will lead to generally higher turbulence levels, and some authors have suggested that this can be calculated from a ‘regional roughness length’ which takes the topography into account as well as the surface roughness (Tieleman, 1992). On the other hand, distortion of the flow by the local terrain may reduce the turbulence intensity. At heights above ground which are of importance for wind turbines, rapid distortion theory applies, which means that the variance of the turbulent fluctuations will not change much as the flow passes over terrain features such as hills. Therefore, if there is acceleration of the flow as it passes over a hill, the turbulence intensity will decrease, and the length scale will increase, resulting in a shift of the turbulence spectrum towards lower frequencies, without any change of shape (Schlez, 2000). This effect is, therefore, easily estimated once a model such as WAS^P has been used to calculate the speed-up factor at a particular point. However, the effect is also accompanied by a shift of turbulent energy from the longitudinal to the lateral and vertical components of turbulence, causing the turbulence on hill tops to be more isotropic (Petersen *et al.*, 1998).

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Aerodynamics of horizontal axis wind turbines*

3.1 Introduction

A wind turbine is a device for extracting kinetic energy from the wind. By removing some of its kinetic energy the wind must slow down but only that mass of air which passes through the rotor disc is affected. Assuming that the affected mass of air remains separate from the air which does not pass through the rotor disc and does not slow down, a boundary surface can be drawn containing the affected air mass and this boundary can be extended upstream as well as downstream forming a long stream-tube of circular cross-section. No air flows across the boundary and so the mass flow rate of the air flowing along the stream-tube will be the same for all stream-wise positions along the stream-tube. Because the air within the stream-tube slows down, but does not become compressed, the cross-sectional area of the stream-tube must expand to accommodate the slower moving air (see Figure 3.1).

Although kinetic energy is extracted from the airflow, a sudden step change in velocity is neither possible nor desirable because of the enormous accelerations and forces this would

*To study the aerodynamics of wind turbines some knowledge of fluid dynamics in general is necessary and, in particular, aircraft aerodynamics. Excellent text books on aerodynamics are readily available, a bibliography is given, as Further Reading, at the end of this chapter, and any abbreviated account of the subject that could have been included in these pages would not have done it justice; recourse to text books would have been necessary anyway. Some direction on which aerodynamics topics are necessary for the study of wind turbines would, however, be useful to the reader and a brief introduction is given in Appendix 3A. For Sections 3.2 and 3.3 a knowledge of Bernoulli's theorem for steady, incompressible flow is required together with the concept of continuity. For Section 3.4, which may be omitted at first reading, an understanding of vortices is desirable and the flow field induced by vortices. The Biot-Savart law, which will be familiar to those with knowledge of electric and magnetic fields, is used to determine velocities induced by vortices. The Kutta-Joukowski theorem for determining the force on a bound vortex should also be studied. For Sections 3.5, 3.6 and 3.7 a knowledge of the lift and drag of aerofoils is essential, including stalled flow.

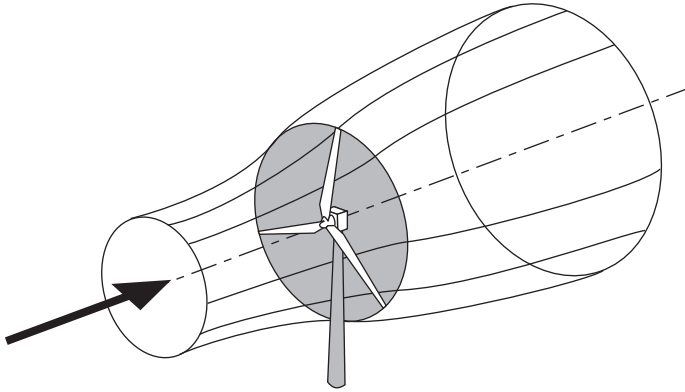


Figure 3.1 The energy extracting stream-tube of a wind turbine

require. Pressure energy *can* be extracted in a step-like manner, however, and all wind turbines, whatever their design, operate in this way.

The presence of the turbine causes the approaching air, upstream, gradually to slow down such that when the air arrives at the rotor disc its velocity is already lower than the free-stream wind speed. The stream-tube expands as a result of the slowing down and, because no work has yet been done on, or by, the air, its static pressure rises to absorb the decrease in kinetic energy.

As the air passes through the rotor disc, by design, there is a drop in static pressure such that, on leaving, the air is below the atmospheric pressure level. The air then proceeds downstream with reduced speed and static pressure: this region of the flow is called the 'wake'. Eventually, far downstream, the static pressure in the wake must return to the atmospheric level for equilibrium to be achieved. The rise in static pressure is at the expense of the kinetic energy and so causes an additional slowing down of the wind. Thus, between the far upstream and far wake conditions, no change in static pressure exists but there is a reduction in kinetic energy.

3.2 The actuator disc concept

The mechanism described above accounts for the extraction of kinetic energy but in no way explains what happens to that energy: it may well be put to useful work but some may be spilled back into the wind as turbulence and eventually be dissipated as heat.

Nevertheless, we can begin an analysis of the aerodynamic behaviour of wind turbines without any specific turbine design just by considering the energy extraction process. The general device that carries out this task is called an actuator disc (see Figure 3.2).

Upstream of the disc the stream-tube has a cross-sectional area smaller than that of the disc and an area larger than the disc downstream. The expansion of the stream-tube is because the mass flow rate must be the same everywhere. The mass of air which passes through a given cross-section of the stream-tube in a unit length of time is ρAU , where ρ is the air

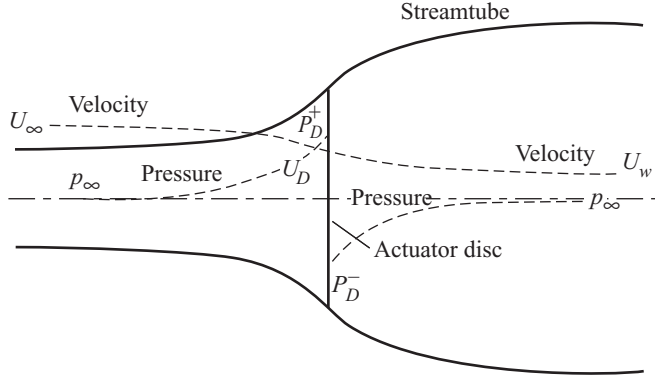


Figure 3.2 An energy extracting actuator disc and stream-tube

density, A is the cross-sectional area and U is the flow velocity. The mass flow rate must be the same everywhere along the stream-tube and so

$$\rho A_{\infty} U_{\infty} = \rho A_D U_D = \rho A_W U_W \quad (3.1)$$

The symbol ∞ refers to conditions far upstream, D refers to conditions at the disc and W refers to conditions in the far wake.

It is usual to consider that the actuator disc induces a velocity variation which must be superimposed on the free stream velocity. The stream-wise component of this induced flow at the disc is given by $-aU_{\infty}$, where a is called the axial flow induction factor, or the inflow factor. At the disc, therefore, the net stream-wise velocity is

$$U_D = U_{\infty}(1 - a) \quad (3.2)$$

3.2.1 Simple momentum theory

The air that passes through the disc undergoes an overall change in velocity, $U_{\infty} - U_W$ and a rate of change of momentum equal to the overall change of velocity times the mass flow rate:

$$\text{Rate of change of momentum} = (U_{\infty} - U_W)\rho A_D U_D \quad (3.3)$$

The force causing this change of momentum comes entirely from the pressure difference across the actuator disc because the stream-tube is otherwise completely surrounded by air at atmospheric pressure, which gives zero net force.

Therefore,

$$(p_D^+ - p_D^-)A_D = (U_{\infty} - U_W)\rho A_D U_{\infty}(1 - a) \quad (3.4)$$

To obtain the pressure difference ($p_D^+ - p_D^-$) Bernoulli's equation is applied separately to the upstream and downstream sections of the stream-tube: separate equations are necessary because the total energy is different upstream and downstream. Bernoulli's equation states

that, under steady conditions, the total energy in the flow, comprising kinetic energy, static pressure energy and gravitational potential energy, remains constant provided no work is done on or by the fluid. Thus, for a unit volume of air,

$$\frac{1}{2}\rho U^2 + p + \rho gh = \text{const} \quad (3.5)$$

Upstream, therefore, we have

$$\frac{1}{2}\rho_{\infty}U_{\infty}^2 + p_{\infty} + \rho_{\infty}gh_{\infty} = \frac{1}{2}\rho_D U_D^2 + p_D^+ + \rho_D gh_D \quad (3.6)$$

Assuming the flow to be incompressible ($\rho_{\infty} = \rho_D$) and horizontal ($h_{\infty} = h_D$) then,

$$\frac{1}{2}\rho U_{\infty}^2 + p_{\infty} = \frac{1}{2}\rho U_D^2 + p_D^+ \quad (3.6a)$$

Similarly, downstream,

$$\frac{1}{2}\rho U_W^2 + p_{\infty} = \frac{1}{2}\rho U_D^2 + p_D^- \quad (3.6b)$$

Subtracting these equations we obtain

$$(p_D^+ - p_D^-) = \frac{1}{2}\rho(U_{\infty}^2 - U_W^2) \quad (3.6c)$$

Equation 3.4 then gives

$$\frac{1}{2}\rho(U_{\infty}^2 - U_W^2)A_D = (U_{\infty} - U_W)\rho A_D U_{\infty}(1 - a) \quad (3.7)$$

and so,

$$U_W = (1 - 2a)U_{\infty} \quad (3.8)$$

That is, half the axial speed loss in the stream-tube takes place upstream of the actuator disc and half downstream.

3.2.2 Power coefficient

The force on the air becomes, from Equation 3.4

$$T = (p_D^+ - p_D^-)A_D = 2\rho A_D U_{\infty}^2 a(1 - a) \quad (3.9)$$

As this force is concentrated at the actuator disc the rate of work done by the force is TU_D and hence the power extraction from the air is given by

$$\text{Power} = TU_D = 2\rho A_D U_{\infty}^3 a(1 - a)^2 \quad (3.10)$$

A *power coefficient* is then defined as

$$C_P = \frac{\text{Power}}{\frac{1}{2}\rho U_\infty^3 A_D} \quad (3.11)$$

where the denominator represents the power available in the air, in the absence of the actuator disc.

Therefore,

$$C_P = 4a(1 - a)^2 \quad (3.12)$$

3.2.3 The Lanchester-Betz limit

The maximum value of C_P occurs when

$$\frac{dC_P}{da} = 4(1 - a)(1 - 3a) = 0$$

that gives a value of $a = \frac{1}{3}$

Hence,

$$C_{P_{\max}} = \frac{16}{27} = 0.593 \quad (3.13)$$

The maximum achievable value of the power coefficient is known as the Lanchester-Betz limit after Frederic Lanchester (1915), a British aeronautical pioneer, and Albert Betz (1919) the German aerodynamicist. To date, no wind turbine has been designed that is capable of exceeding the Lanchester-Betz limit. The limit is caused not by any deficiency in design because, as yet in our discussion, we have no design. However, because the stream-tube has to expand *upstream* of the actuator disc the cross-section of the tube where the air is at the full, free-stream velocity is smaller than the area of the disc.

C_P could, perhaps, more fairly be defined as

$$C_P = \frac{\text{Power} \cdot \text{extracted}}{\text{Power} \cdot \text{available}} = \frac{\text{Power} \cdot \text{extracted}}{\frac{16}{27} \left(\frac{1}{2} \rho U_\infty^3 A_D \right)} \quad (3.14)$$

but this not the accepted definition of C_P .

3.2.4 The thrust coefficient

The force on the actuator disc caused by the pressure drop, given by Equation 3.9, can also be non-dimensionalised to give a *coefficient of thrust* C_T

$$C_T = \frac{\text{Thrust}}{\frac{1}{2}\rho U_\infty^2 A_D} \quad (3.15)$$

$$C_T = 4a(1 - a) \quad (3.16)$$

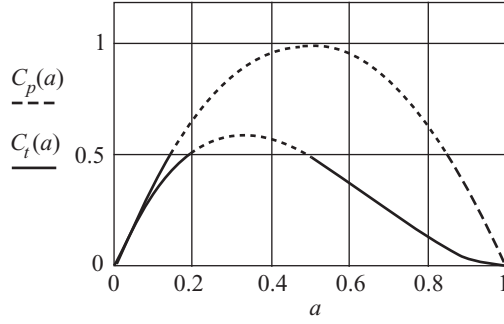


Figure 3.3 Variation of C_P and C_T with axial induction factor a

A problem arises for values of $a \geq \frac{1}{2}$ because the wake velocity, given by $(1 - 2a)U_\infty$, becomes zero, or even negative: in these conditions the momentum theory, as described, no longer applies and an empirical modification has to be made (Section 3.6).

The variation of power coefficient and thrust coefficient with a is shown in Figure 3.3.

3.3 Rotor disc theory

The manner in which the extracted energy is converted into usable energy depends upon the particular turbine design. Most wind energy converters employ a rotor with a number of blades rotating with an angular velocity Ω about an axis normal to the rotor plane and parallel to the wind direction. The blades sweep out a disc and, by virtue of their aerodynamic design, develop a pressure difference across the disc, which, as discussed in the previous section, is responsible for the loss of axial momentum in the wake. Associated with the loss of axial momentum is a loss of energy that can be collected by, say, an electrical generator attached to the rotor shaft. As well as a thrust, the rotor experiences a torque in the direction of rotation that will oppose the torque that the generator exerts. The work done by the aerodynamic torque on the generator is converted into electrical energy. The required aerodynamic design of the rotor blades to provide a torque as well as a thrust is discussed in Section 3.5.

3.3.1 Wake rotation

The exertion of a torque on the rotor disc by the air passing through it requires an equal and opposite torque to be imposed upon the air. The consequence of the reaction torque is to cause the air to rotate in a direction opposite to that of the rotor; the air gains angular momentum and so in the wake of the rotor disc the air particles have a velocity component in a direction which is tangential to the rotation as well as an axial component (see Figure 3.4).

The acquisition of the tangential component of velocity by the air means an increase in its kinetic energy that is compensated for by a fall in the static pressure of the air in the wake in addition to that which is described in the previous section.

The flow entering the actuator disc has no rotational motion at all. The flow exiting the disc does have rotation and that rotation remains constant as the fluid progresses down the wake. The transfer of rotational motion to the air takes place entirely across the thickness of

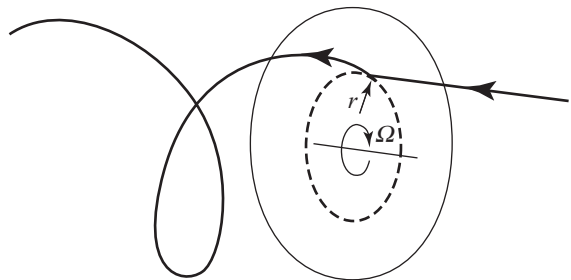


Figure 3.4 The trajectory of an air particle passing through the rotor disc

the disc (see Figure 3.5). The change in tangential velocity is expressed in terms of a tangential flow induction factor a' . Upstream of the disc the tangential velocity is zero. Immediately downstream of the disc the tangential velocity is $2r\Omega a'$. At the middle of the disc thickness, a radial distance r from the axis of rotation, the induced tangential velocity is $r\Omega a'$. Because it is produced in reaction to the torque the tangential velocity is opposed to the motion of the rotor.

An abrupt acquisition of tangential velocity cannot occur in practice and must be gradual. Figure 3.5 shows the flow accelerating in the tangential direction as it is ‘squeezed’ between the blades: the separation of the blades has been reduced for effect, but it is the increasing

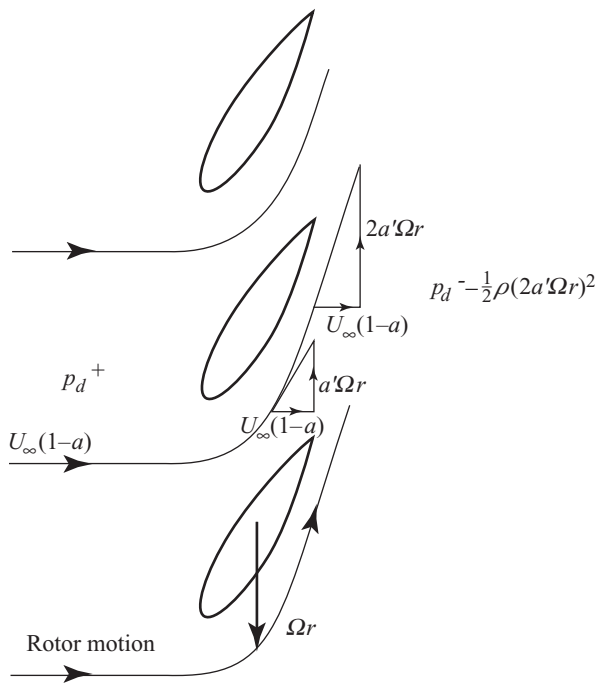


Figure 3.5 Tangential velocity grows across the disc thickness

solid blockage that the blades present to the flow close to the blade roots that causes high values of tangential velocity.

3.3.2 Angular momentum theory

The tangential velocity will not be the same for all radial positions and it may well also be that the axial induced velocity is not the same. To allow for variation of both induced velocity components consider only an annular ring of the rotor disc which is of radius r and of radial width δr .

The increment of rotor torque acting on the annular ring will be responsible for imparting the tangential velocity component to the air, whereas the axial force acting on the ring will be responsible for the reduction in axial velocity. The whole disc comprises a multiplicity of annular rings and each ring is assumed to act independently in imparting momentum only to the air that actually passes through the ring.

The torque on the ring will be equal to the rate of change of angular momentum of the air passing through the ring.

Thus, torque = rate of change of angular momentum = mass flow rate $\times$ change of tangential velocity $\times$ radius

$$\delta Q = \rho \delta A_D U_\infty (1 - a) 2\Omega a' r^2 \quad (3.17)$$

where δA_D is taken as being the area of an annular ring.

The driving torque on the rotor shaft is also δQ and so the increment of rotor shaft power output is

$$\delta P = \delta Q \Omega$$

The total power extracted from the wind by slowing it down is, therefore, determined by the rate of change of axial momentum given by Equation 3.10 in Section 3.2.2

$$\delta P = 2\rho \delta A_D U_\infty^3 (1 - a)^2$$

Hence,

$$2\rho \delta A_D U_\infty^3 a(1 - a)^2 = \rho \delta A_D U_\infty (1 - a) 2\Omega^2 a' r^2$$

and

$$U_\infty^2 a(1 - a) = \Omega^2 r^2 a'$$

r is the tangential velocity of the spinning annular ring and so $\lambda_r = r\Omega/U_\infty$ is called the *local speed ratio*. At the edge of the disc $r = R$ and $\lambda = R\Omega/U_\infty$ is known as the *tip speed ratio*.

Thus,

$$a(1 - a) = \lambda_r^2 a' \quad (3.18)$$

The area of the ring is $\delta A_D = 2\pi r \delta r$, therefore, the incremental shaft power is, from Equation 3.17,

$$\delta P = \delta Q \Omega = \left(\frac{1}{2} \rho U_\infty^3 2\pi r \delta r \right) 4a'(1-a)\lambda_r^2$$

The term in brackets represents the power flux through the annulus, the term outside the brackets, therefore, is the efficiency of the blade element in capturing the power.

Blade element efficiency

$$\eta_r = 4a'(1-a)\lambda_r^2 \quad (3.19)$$

In terms of power coefficient

$$\begin{aligned} \frac{dC_P}{dr} &= \frac{4\pi\rho U_\infty^3 (1-a)a'\lambda_r^2 r}{\frac{1}{2}\rho U_\infty^3 \pi R^2} = \frac{8(1-a)a'\lambda_r^2 r}{R^2} \\ \frac{dC_P}{d\mu} &= 8(1-a)a'\lambda^2 \mu^3 \end{aligned} \quad (3.20)$$

where $\mu = r/R$.

Knowing how a and a' vary radially, Equation 3.20 can be integrated to determine the overall power coefficient for the disc for a given tip speed ratio λ .

It was argued by Glauert (1935b) that the rotation in the wake required energy that is taken from the flow and is unavailable for extraction; but this can be shown not to be the case. The lift forces on the blades forming the rotor disc are normal to the resultant velocity relative to the blades and so no work is done on or by the fluid. Therefore, Bernoulli's theorem can be applied to the flow across the disc, relative to the disc spinning at angular velocity Ω , to give

$$\begin{aligned} &\frac{1}{2}\rho U_\infty^2 (1-a)^2 + \frac{1}{2}\rho \Omega^2 r^2 + \frac{1}{2}\rho w^2 + p_D^+ \\ &= \frac{1}{2}\rho U_\infty^2 (1-a)^2 + \frac{1}{2}\rho \Omega^2 (1+2a')^2 r^2 + \frac{1}{2}\rho w^2 + p_D^- \end{aligned}$$

Consequently,

$$\Delta p_d = 2\rho \Omega^2 (1+a')a'r^2$$

The pressure drop across the disc clearly has two sources. The first source

$$\Delta p_{d1} = 2\rho \Omega^2 a' r^2 \quad (3.21)$$

is shown to be, from Equation 3.18, the same as that given by Equation 3.9 in the simple momentum theory in which rotation plays no part. The second source is

$$\Delta p_{d2} = 2\rho \Omega^2 a'^2 r^2 \quad (3.22)$$

Δp_{d2} can be shown to be caused by a radial, static pressure gradient in the rotating wake that balances the centrifugal force on the rotating fluid.

$$\frac{dp}{dr} = \rho(2\Omega a')^2 r$$

The kinetic energy per unit volume of the rotating fluid in the wake is also equal to the drop in static pressure of Equation 3.22; and so the two are in balance and there is no loss of available kinetic energy.

The question arises, however, does the pressure drop of Equation 3.22 cause an additional thrust on the rotor disc? The answer is no, because whereas the pressure drop of Equation 3.21 applies only to points immediately downstream of the disc, with a positive axial pressure gradient between the disc and the fully developed wake (Figure 3.2), the pressure drop of Equation 3.22 applies to the entire wake and there is no pressure gradient causing a further slowing down of the axial velocity. If there is no change of axial momentum of the fluid there can be no corresponding force on the rotor because that would violate Newton's third law of motion! There appears to be an anomaly and it is probable that this is caused by the concept of an actuator disc itself because an actuator disc is not physically realisable.

However, the concept does produce useful results and is not to be dismissed.

3.3.3 Maximum power

The values of a and a' which will provide the maximum possible efficiency can be determined by differentiating Equation 3.19 by either factor and putting the result equal to zero.

Whence

$$\frac{da}{da'} = \frac{1-a}{a'} \quad (3.23)$$

From Equation 3.18

$$\frac{da}{da'} = \frac{\lambda_r^2}{1-2a}$$

giving

$$a' \lambda_r^2 = (1-a)(1-2a) \quad (3.24)$$

The combination of Equations 3.18 and 3.21 gives the required values of a and a' which maximise the incremental power coefficient.

$$a = \frac{1}{3}$$

and

$$a' = \frac{a(1-a)}{\lambda_r^2} \quad (3.25)$$

The axial flow induction for maximum power extraction is the same as for the non-rotating wake case, that is, $a = \frac{1}{3}$ and is uniform over the entire disc. On the other hand a' varies with radial position.

From Equation 3.20 the maximum power is

$$C_P = 8 \int_0^1 (1-a) a' \lambda^2 \mu^3 d\mu$$

Substituting for the expressions in Equation 3.23

$$C_P = 8 \int_0^1 (1-a) \frac{a(1-a)}{\lambda^2 \mu^2} \lambda^2 \mu^3 d\mu = 4a(1-a)^2 = \frac{16}{27} \quad (3.26)$$

Which is precisely the same as for the non-rotating wake case.

3.4 Vortex cylinder model of the actuator disc

3.4.1 Introduction

The momentum theory of Section 3.1 uses the concept of the actuator disc across which a pressure drop develops constituting the energy extracted by the rotor. In the rotor disc theory of Section 3.3 the actuator disc is depicted as being swept out by a multiplicity of aerofoil blades each with radially uniform bound circulation $\Delta\Gamma$. From the tip of each blade a helical vortex of strength $\Delta\Gamma$ convects downstream with the local flow velocity. If the number of blades is assumed to be very large but the solidity of the total is finite and small then the accumulation of helical tip vortices will form the surface of a tube. As the number of blades approaches infinity the tube surface will become a continuous tubular vortex sheet (see Figure 3.6).

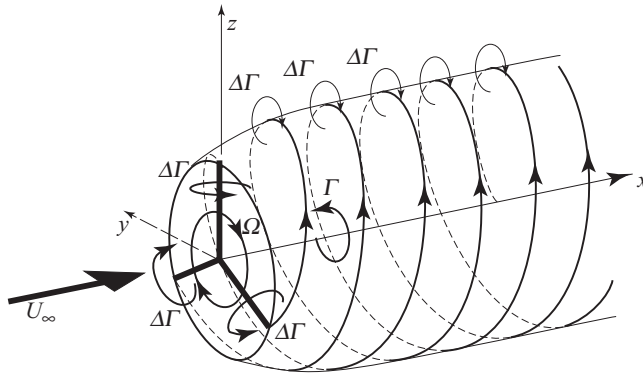


Figure 3.6 Helical vortex wake shed by rotor with three blades each with uniform circulation $\Delta\Gamma$

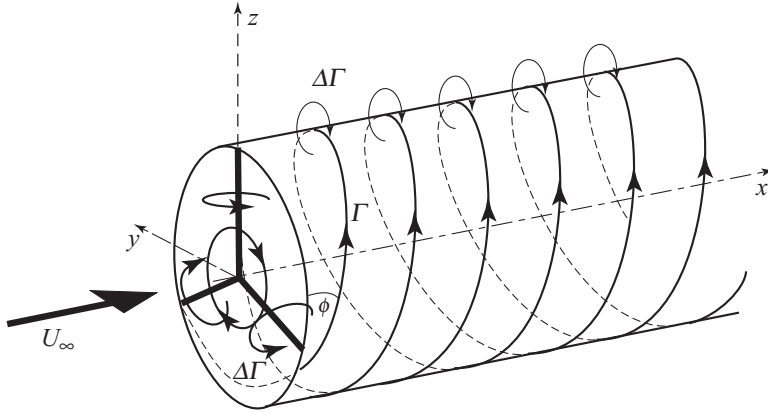


Figure 3.7 Simplified helical vortex wake ignoring wake expansion

From the root of each blade, assuming it reaches to the axis of rotation, a line vortex of strength $\Delta\Gamma$ will extend downstream along the axis of rotation contributing to the total root vortex of strength Γ . The vortex tube will expand in radius as the flow of the wake inside the tube slows down. Vorticity is confined to the surface of the tube, the root vortex and to the bound vortex sheet swept by the multiplicity of blades to form the rotor disc; elsewhere in the wake and everywhere else in the entire flow field the flow is irrotational (see Figure 3.7).

The nature of the tube's expansion cannot be determined by means of the momentum theory and so, as an approximation, the tube is allowed to remain cylindrical. The Biot-Savart law is used to determine the induced velocity at any point in the vicinity of the actuator disc. The cylindrical vortex model allows the whole flow field to be determined and is accurate within the limitations of the non-expanding cylindrical wake.

3.4.2 Vortex cylinder theory

The vortex cylinder has surface vorticity which follows a helical path with a helix angle ϕ_t or, as it has been termed previously, the flow angle at the outer edge of the disc. The strength of the vorticity is $g = d\Gamma/dn$, where n is a direction in the tube surface normal to the direction of $\Delta\Gamma$, and has a component $g_\theta = g \cos \phi_t$ parallel to the rotor disc. Due to g_θ the axial (parallel to the axis of rotor rotation) induced velocity at the rotor plane is *uniform* over the rotor disc and can be determined by means of the Biot-Savart law as

$$u_d = -\frac{g_\theta}{2} = -aU_\infty \quad (3.27)$$

In the far wake the axial induced velocity is also uniform within the cylindrical wake and is

$$u_w = -g_\theta = -2aU_\infty \quad (3.28)$$

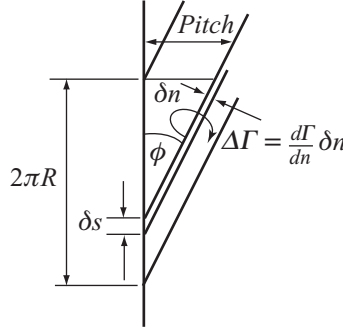


Figure 3.8 The geometry of the vorticity in the cylinder surface

The ratio of the two induced velocities corresponds to that of the simple momentum theory and justifies the assumption of a cylindrical vortex sheet.

3.4.3 Relationship between bound circulation and the induced velocity

The total circulation on all of the multiplicity of blades is Γ which is shed at a uniform rate into the wake in one revolution. So, from Figure 3.8 in which the cylinder has been slit longitudinally and opened out flat,

$$g = \frac{\Gamma}{2\pi R \sin \phi_t} \quad (3.29)$$

hence,

$$g_\theta = \frac{\Gamma \cos \phi_t}{2\pi R \sin \phi_t} = \frac{\Gamma}{2\pi R} \frac{\Omega R}{U_\infty (1 - a)} \quad (3.30)$$

therefore,

$$2aU_\infty = \frac{\Gamma}{2\pi R} \frac{\Omega R}{U_\infty (1 - a)} \quad (3.31)$$

So, the total circulation is related to the induced velocity factors

$$\Gamma = \frac{4\pi U_\infty^2 a(1 - a)}{\Omega} \quad (3.32)$$

3.4.4 Root vortex

Just as a vortex is shed from each blade tip a vortex is also shed from each blade root. If it is assumed that the blades extend to the axis of rotation, obviously not a practical option, then the root vortices will each be a line vortex running axially downstream from the centre of the

disc. The direction of rotation of the all the root vortices will be the same forming a core, or root, vortex of total strength Γ . The root vortex is primarily responsible for inducing the tangential velocity in the wake flow and in particular the tangential velocity on the rotor disc.

On the rotor disc surface the tangential velocity induced by the root vortex, given by the Biot-Savart law, is

$$a' r \Omega = \frac{\Gamma}{4\pi r}$$

so

$$a' = \frac{\Gamma}{4\pi r^2 \Omega} \quad (3.33)$$

This relationship can also be derived from the momentum theory: the rate of change of angular momentum of the air which passes through an annulus of the disc of radius r and radial width δr is equal to the torque increment imposed upon the annulus

$$\delta Q = \rho U_\infty (1 - a) 2\pi r 2a' r^2 \Omega \delta r \quad (3.34)$$

The torque per unit span acting on all the blades is given by the Kutta-Joukowski theorem. The lift per unit radial width, L , is

$$L = \rho (W \times \Gamma)$$

where $(W \times \Gamma)$ is a vector product

$$\delta Q = \rho W \times \Gamma r \sin \phi_t \delta r = \rho \Gamma r U_\infty (1 - a) \delta r \quad (3.35)$$

Equating the two expressions for δQ gives

$$a' = \frac{\Gamma}{4\pi r^2 \Omega}$$

Hence,

$$a' = \frac{U_\infty^2 a(1 - a)}{\lambda_r^2} = \frac{a(1 - a)}{\lambda_r^2}$$

At the outer edge of the disc the tangential induced velocity is

$$a'_t = \frac{a(1 - a)}{\lambda^2} \quad (3.36)$$

Equation 3.36 is exactly the same as Equation 3.23 of Section 3.3.3.

3.4.5 Torque and power

The torque on an annulus of radius r and radial width δr is

$$\begin{aligned}\frac{dQ}{dr}\delta r &= \rho W \Gamma r \sin \phi_i \delta r = \frac{\rho 4\pi r U_\infty^3 a(1-a)^2}{\Omega} \delta r \\ \frac{dQ}{dr} &= \frac{\frac{1}{2}\rho U_\infty^3 2\pi r 4a(1-a)^2}{\Omega}\end{aligned}\quad (3.37)$$

The radial distribution of power is

$$\frac{dP}{dr} = \Omega \frac{dQ}{dr} = \frac{1}{2}\rho U_\infty^3 2\pi r 4a(1-a)^2 \quad (3.38)$$

and, therefore, the total power is

$$P = \frac{1}{2}\rho U_\infty^3 \pi R^2 4a(1-a)^2 \quad (3.39)$$

Power coefficient

$$C_P = 4a(1-a)^2 = 4a'_i(1-a)\lambda^2 \quad (3.40)$$

Again a result that is identical to that predicted by the simple momentum theory.

What is particularly interesting is that the rotational flow in the wake makes no difference at all to the efficiency of the power extraction.

3.4.6 Axial flow field

The induced velocity in the wind-wise (axial) direction can be determined both upstream of the disc and downstream in the developing wake, as well as on the disc itself. The flow field (net velocity) is axi-symmetric and a radial cross-section is shown in Figure 3.9. Both radial and axial distances are divided by the disc radius with the axial distance being measured downstream from the disc and the radial distance being measured from the rotational axis. The velocity is divided by the wind speed.

The axial velocity within the wake is sharply lower than without and is radially uniform at the disc and in the far wake, just as the momentum theory predicts. There is a small acceleration of the flow immediately outside of the wake. The induced velocity on the wake cylinder itself is $\frac{1}{4}a$ at the disc and $\frac{1}{2}a$ in the far wake.

3.4.7 Tangential flow field

The tangential induced velocity is determined not only by the root vortex but also by the component of vorticity $g \sin \phi_i$ on the wake cylinder and the bound vorticity on the rotor disc. At a radial distance equal to half the disc radius, as an example, the axial variation of the three contributions are shown in Figure 3.10.

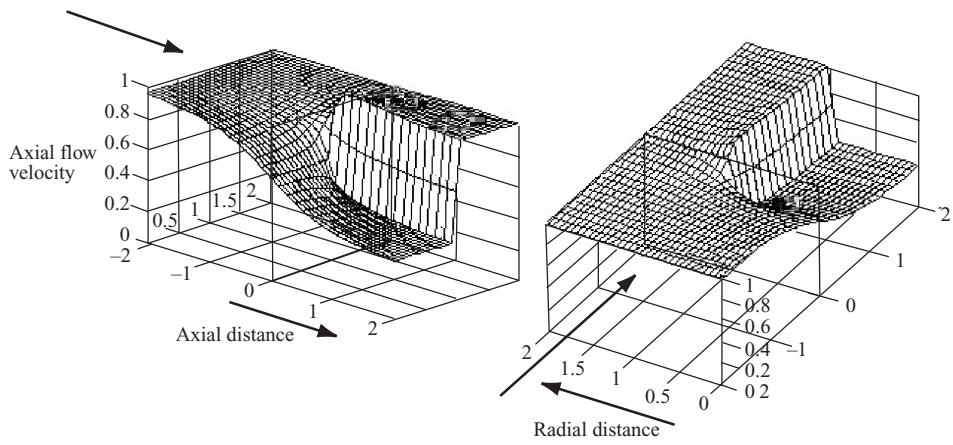


Figure 3.9 The radial and axial variation of axial velocity in the vicinity of an actuator disc, $a = \frac{1}{3}$

The bound vorticity causes rotation in opposite senses upstream and downstream of the disc with a step change across the disc. The upstream rotation, which is in the same sense as the rotor rotation, is nullified by the root vortex, which induces rotation in the opposite sense to that of the rotor. The downstream rotation is in the same sense for both the root vortex and the bound vorticity the stream-wise variations of the two summing to give a uniform velocity in the stream-wise sense. The vorticity located on the surface of the wake cylinder makes a small contribution.

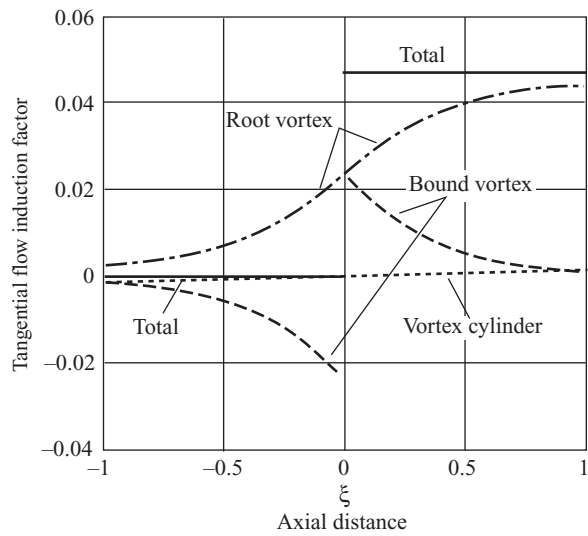


Figure 3.10 The axial variation of tangential velocity in the vicinity of an actuator disc at 50% radius, $a = \frac{1}{3}$, $\lambda = 6$

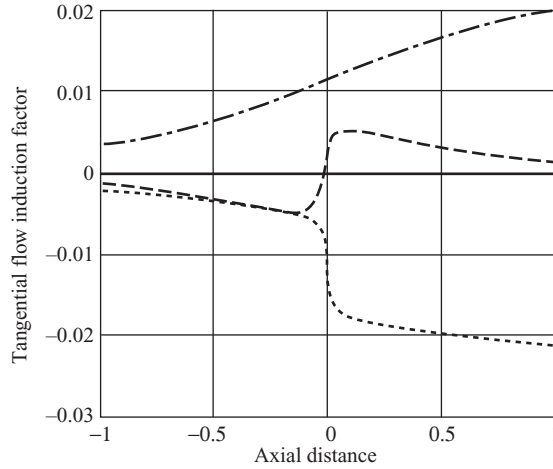


Figure 3.11 The axial variation of tangential velocity in the vicinity of an actuator disc at 101% radius, $a = \frac{1}{3}$, $\lambda = 6$

At the disc itself the bound vorticity induces no rotation, the wake cylinder induces no rotation either and so it is only the root vortex which does induce rotation and that value is half the total induced generally in the wake. It is now clear why only half the rotational velocity is used to determine the flow angle at the disc.

The rotational flow is confined to the wake, that is, inside the cylinder. There is no rotational flow anywhere outside of the wake, neither upstream of the disc or outside the cylinder. The rotational flow within the cylinder falls with increasing radius but is not zero at the outer edge of the wake; therefore, there is an abrupt fall of rotational velocity across the cylindrical surface.

The cylinder itself, therefore, rotates with angular velocity $a'_t \Omega$ and so the rotation of the flow relative to the disc is $(1 + a'_t) \Omega$. It would seem that the flow angle ϕ_t should take the additional rotation into account but it is determined by the flow relative to the moving cylindrical surface for which the relative rotational velocity is just Ω .¹

The contributions of the three vorticity sources to the rotational flow at a radius of 101% of the disc radius are shown in Figure 3.11: the total rotational flow is zero at all axial positions but the individual components are not zero.

3.4.8 Axial thrust

The axial thrust T on the disc can be determined using the Kutta-Joukowski theorem

$$\frac{dT}{dr} = \rho \Gamma V$$

¹ In the first edition of the Wind Energy Handbook the flow angle was determined, mistakenly, by including the rotation of the cylinder in the relative rotational velocity.

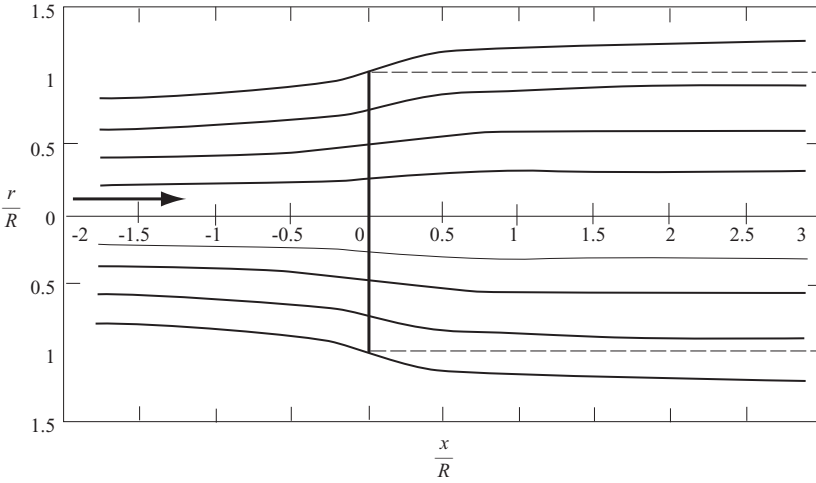


Figure 3.12 Flow field through an actuator disc for $a = \frac{1}{3}$

where V is the tangential velocity component at the disc. If $V = r\Omega$, then, using Equation 3.30

$$\frac{dT}{dr} = \rho 4\pi r U_\infty^2 a(1-a) \quad (3.41)$$

Integration of Equation 3.41 over the entire disc gives the thrust coefficient as

$$C_T = 4a(1 - a) \quad (3.42)$$

That is, the same as for the simple momentum theory and so is in balance with the rate of change of axial momentum. However, if the induced tangential velocity $a'r\Omega$ is included in V , as might be assumed, the resulting thrust becomes infinite, or very much greater than the rate of change of axial momentum, and the necessary balance is lost.

3.4.9 Radial flow field

Although the vortex cylinder model has been simplified by not allowing the cylinder to expand, the theory nevertheless predicts flow expansion. A radial velocity distribution is predicted by the theory as shown in Figure 3.12 which shows a longitudinal section of the flow field through the rotor disc.

The radial velocity, as calculated, is greatest as the flow passes through the rotor disc and, although not shown in Figure 3.12, it is infinite at the edge of the disc. The situation is very similar to the determination of the potential flow field around a flat, solid, circular disc which is normal to the oncoming flow; an infinite radial velocity is predicted at the disc edge for the flow to pass around and continue radially inwards on the downstream side. In practice, there would be insufficient static pressure available to fuel an infinite, or even a very high, velocity and so some discontinuity in the flow must occur. The presence of even the smallest amount of viscosity would produce a thick boundary layer towards the disc edge because

of the high radial velocity. The viscosity in the boundary layer would absorb much of the available energy and dissipate it as heat so that, as the flow accelerated around the disc edge, the maximum velocity attainable would be limited by the static pressure approaching zero. Instead of the flow moving around the edge it would separate from the edge and continue downstream leaving a very low pressure region behind the disc with very low velocity: a stagnant region.

In the case of the permeable rotor disc there would be some flow through the disc, which would behave as described above, but the separation of the flow at the disc edge would produce an additional low pressure in the wake.

The problem of the infinite radial velocity at the rotor disc edge arises because of the assumption of an infinite number of rotor blades. If the theory is modified such that there are only a few blades the infinite radial velocity disappears. However, if, for a given rotor, the tip speed ratio is increased, with a consequent increase of the axial flow induction factor, the radial velocity at the tip rises sharply and the problem of edge separation returns, which is what actually occurs (see Section 3.6).

3.4.10 Conclusions

Despite the exclusion of wake expansion the vortex theory produces results exactly in agreement with the momentum theory and enlightens understanding of the flow through an energy extracting actuator disc. However, the infinite radial velocity predicted at the outer edge of the disc is further evidence that the actuator disc is physically unrealisable.

3.5 Rotor blade theory (blade-element/momentum theory)

3.5.1 Introduction

The aerodynamic lift (and drag) forces on the span-wise elements of radius r and length δr of the several blades of a wind turbine rotor are responsible for the rate of change of axial and angular momentum of all of the air which passes through the annulus swept by the blade elements. In addition, the force on the blade elements caused by the drop in pressure associated with the rotational velocity in the wake must also be provided by the aerodynamic lift and drag. As there is no rotation of the flow approaching the rotor, the reduced pressure on the downwind side of the rotor caused by wake rotation appears as a step pressure drop just as is that which causes the change in axial momentum. Because the wake is still rotating in the far wake the pressure drop that caused the rotation is still present and so does not contribute to the axial momentum change.

3.5.2 Blade element theory

It is assumed that the forces on a blade element can be calculated by means of two-dimensional aerofoil characteristics using an angle of attack determined from the incident resultant velocity in the cross-sectional plane of the element; the velocity component in the span-wise direction is ignored. Three-dimensional effects are also ignored.

The velocity components at a radial position on the blade expressed in terms of the wind speed, the flow factors and the rotational speed of the rotor will determine the angle of attack.

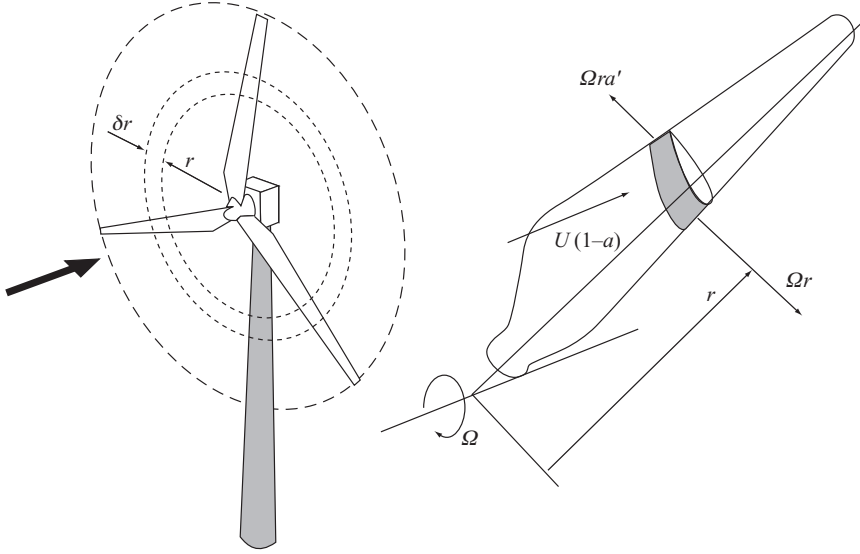


Figure 3.13 A blade element sweeps out an annular ring

Having information about how the aerofoil characteristic coefficients C_l and C_d vary with the angle of attack, the forces on the blades for given values of a and a' can be determined.

Consider a turbine with B blades of tip radius R each with chord c and set pitch angle β measured between the *aerofoil zero lift line* and the plane of the disc. Both the chord length and the pitch angle may vary along the blade span. Let the blades be rotating at angular velocity Ω and let the wind speed be U_∞ . The tangential velocity experienced by the blade element shown in Figure 3.13 is, by the argument offered in Section 3.4.8 is $r\Omega$. However, despite the validity of the reasoning in Section 3.4.8, it appears logical to combine the tangential velocity with that of the wake $a'r\Omega$ giving a net tangential flow velocity experienced by the blade element as $(1 + a')r\Omega$. In all of the literature available, particularly Glauert, it is the combined tangential velocity that is assumed to be correct and so it will be in this text, even though it has been demonstrated to be invalid in the case of the actuator disc. The actuator disc is infinitesimally thin and the change in tangential velocity is abrupt. The rotor blade 'disc', on the other hand has axial depth and the tangential velocity develops in a gradual manner, as demonstrated in Figure 3.5.

Figure 3.14 shows all the velocities and forces relative to the blade chord line at radius r . From Figure 3.14 the resultant relative velocity at the blade is

$$W = \sqrt{U_\infty^2(1-a)^2 + r^2\Omega^2(1+a')^2} \quad (3.43)$$

that acts at an angle ϕ to the plane of rotation such that

$$\sin \phi = \frac{U_\infty(1-a)}{W} \quad \text{and} \quad \cos \phi = \frac{r\Omega(1+a')}{W} \quad (3.44)$$

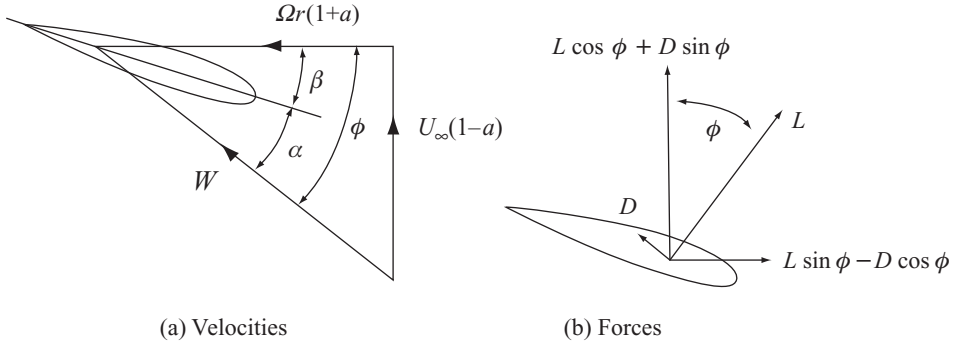


Figure 3.14 Blade element velocities and forces

The angle of attack α is then given by

$$\alpha = \phi - \beta \quad (3.45)$$

The basic assumption of the blade element theory is that the aerodynamic lift and drag forces acting upon an element are the same as those acting on an isolated, identical element at the same angle of attack in two-dimensional flow.

The lift force on a span-wise length δr of each blade, normal to the direction of W , is, therefore,

$$\delta L = \frac{1}{2} \rho W^2 c C_l \delta r$$

and the drag force parallel to W is

$$\delta D = \frac{1}{2} \rho W^2 c C_d \delta r$$

The axial thrust on an annular ring of the actuator disc is

$$\delta T = \delta L \cos \phi + \delta D \sin \phi = \frac{1}{2} \rho W^2 B c (C_l \cos \phi + C_d \sin \phi) \delta r \quad (3.46)$$

The torque on an annular ring is

$$\delta Q = (\delta L \sin \phi - \delta D \cos \phi) r = \frac{1}{2} \rho W^2 B c r (C_l \sin \phi - C_d \cos \phi) \delta r \quad (3.47)$$

where B is the number of blades.

3.5.3 The blade-element/momentum (BEM) theory

The basic assumption of the BEM theory is that the force of a blade element is solely responsible for the change of axial momentum of the air which passes through the annulus

swept by the element. It is, therefore, to be assumed that there is no radial interaction between the flows through contiguous annuli: a condition that is, strictly, only true if the axial flow induction factor does not vary radially. In practice, the axial flow induction factor is seldom uniform, but experimental examination of flow through propeller discs by Lock (1924) shows that the assumption of radial independence is acceptable.

Equating the axial thrust on all blade elements, given Equation 3.46 with the rate of change of axial momentum of the air that passes through the annulus swept out by the elements, given by Equation 3.9 with $A_D = 2\pi r\delta r$

$$\delta T = \frac{1}{2}\rho W^2 Bc(C_l \cos \phi + C_d \sin \phi)\delta r = 2\pi r\delta r\rho U_\infty(1-a)2aU_\infty \quad (3.48)$$

Also equating the torque on the elements, given by Equation 3.47, with the rate of change of angular momentum of the air passing through the swept annulus, given by Equation 3.34,

$$\delta Q = \frac{1}{2}\rho W^2 Bcr(\delta L \sin \phi - \delta D \cos \phi)\delta r = 2\pi r\delta r\rho U_\infty(1-a)2a'r^2\Omega \quad (3.49)$$

If drag is eliminated from the above two equations, to make a comparison with the results of the vortex theory of Section 3, the flow angle ϕ can be determined

$$\tan \phi = \frac{a'r\Omega}{aU_\infty} = \frac{a'}{a} \frac{r}{R} \lambda$$

However, from the velocity triangle at a blade element given by Equation 3.44 the flow angle is also

$$\tan \phi = \frac{1-a}{\lambda_r(1+a')}$$

Equating the two above expressions for $\tan \phi$

$$\frac{a'}{a} \frac{r}{R} \lambda = \frac{1-a}{\lambda_r(1+a')}$$

$$a(1-a) = \lambda_r^2 a'(1+a') \quad (3.50)$$

At the outer edge of the rotor $\mu = 1$ and $a' = a'_t$, so

$$a(1-a) = \lambda^2 a'_t(1+a'_t) \quad (3.50a)$$

Equation 3.50a differs from Equation 3.36 by the additional term $(1+a'_t)$. By the argument put forward in the last paragraph of Section 3.4.8 the inclusion of the additional term would result in an infinite thrust but, as previously stated, the inclusion of the term is the usual practice.

With drag included, the thrust Equation 3.48 can be reduced to

$$\frac{W^2}{U_\infty^2} B \frac{c}{R} (C_l \cos \phi + C_d \sin \phi) = 8\pi a(1-a)\mu \quad (3.51)$$

where the parameter $\mu = r/R$ the torque Equation 3.49 simplifies to

$$\frac{W^2}{U_\infty^2} N \frac{c}{R} (C_l \sin \phi - C_d \cos \phi) = 8\pi \lambda \mu^2 a'(1-a) \quad (3.52)$$

It is convenient to put

$$C_l \cos \phi + C_d \sin \phi = C_x \quad (3.53)$$

and

$$C_l \sin \phi - C_d \cos \phi = C_y$$

Solving Equations 3.51 and 3.52 to obtain values for the flow induction factors a and a' using two-dimensional aerofoil characteristics requires an iterative process for which the following equations, derived from 3.51, 3.52 and 3.53, are convenient; in which the right hand sides are evaluated using existing values of the flow induction factors yielding simple equations for the next iteration of the flow induction factors.

$$\frac{a}{1-a} = \frac{\sigma_r}{4 \sin^2 \phi} C_x \quad (3.54)^2$$

$$\frac{a'}{1+a'} = \frac{\sigma_r C_y}{4 \sin \phi \cos \phi} \quad (3.55)$$

Blade solidity σ is defined as total blade area divided by the rotor disc area and is a primary parameter in determining rotor performance. Chord solidity σ_r is defined as the total blade chord length at a given radius divided by the circumferential length at that radius.

$$\sigma_r = \frac{B}{2\pi r} \frac{c}{R} = \frac{B}{2\pi \mu} \frac{c}{R} \quad (3.56)$$

It is argued by Wilson and Lissaman (1974) that the drag coefficient should not be included in Equations 3.54 and 3.55 because the velocity deficit caused by drag is confined to the narrow wake which flows from the trailing edge of the aerofoil. Furthermore, Wilson and Lissaman reason, the drag based velocity deficit is only a feature of the wake and does not contribute to the velocity deficit upstream of the rotor disc. The basis of the argument for excluding drag

²Equation 3.54 is changed from that given in the first edition because the flow rotation is now properly accounted for.

in the determination of the flow induction factors is that, for attached flow, drag is caused only by skin friction and does not affect the pressure drop across the rotor. Clearly, in stalled flow the drag is overwhelmingly caused by pressure. In attached flow it has been shown by Young and Squire (1938) that the modification to the inviscid pressure distribution around an aerofoil caused by the boundary layer has an effect both on lift and drag. The ratio of pressure drag to total drag at zero angle of attack is approximately the same as the thickness to chord ratio of the aerofoil and increases as the angle of attack increases.

One last point about the blade element-momentum theory: the theory is strictly only applicable if the blades have uniform circulation, that is, if a is uniform. For non-uniform circulation there is a radial interaction and exchange of momentum between flows through adjacent elemental annular rings. It cannot be stated that the only axial force acting on the flow through a given annular ring is that due to the pressure drop across the disc.

However, in practice, it appears that the error involved in relaxing the above constraint is small for tip speed ratios greater than three.

3.5.4 Determination of rotor torque and power

The calculation of torque and power developed by a rotor requires knowledge of the flow induction factors, which are obtained by solving Equations 3.54 and 3.55. The solution is usually carried out iteratively because the two-dimensional aerofoil characteristics are non-linear functions of the angle of attack.

To determine the complete performance characteristic of a rotor, that is, the manner in which the power coefficient varies over a wide range of tip speed ratio, requires the iterative solution.

The iterative procedure is to assume a and a' to be zero initially, determining ϕ , C_l and C_d on that basis, and then to calculate new values of the flow factors using Equations 3.54 and 3.55. The iteration is repeated until convergence is achieved.

From Equation 3.51 the torque developed by the blade elements of span-wise length δr is

$$\delta Q = 4\pi\rho U_\infty \Omega r a' (1 - a) r^2 \delta r$$

If drag, or part of the drag, has been excluded from the determination of the flow induction factors then its effect must be introduced when the torque is calculated, see Equation 3.51,

$$\delta Q = 4\pi\rho U_\infty \Omega r a' (1 - a) r^2 \delta r - \frac{1}{2} \rho W^2 B c C_d \cos \phi r \delta r$$

The complete rotor, therefore, develops a total torque Q .

$$Q = \frac{1}{2} \rho U_\infty^2 \pi R^3 \lambda \int_0^R \mu^2 \left(8a'(1 - a)\mu - \frac{W}{U_\infty} \frac{B}{\pi} \frac{c}{R} C_d (1 + a') \right) d\mu \quad (3.57)$$

The power developed by the rotor is

$$P = Q\Omega$$

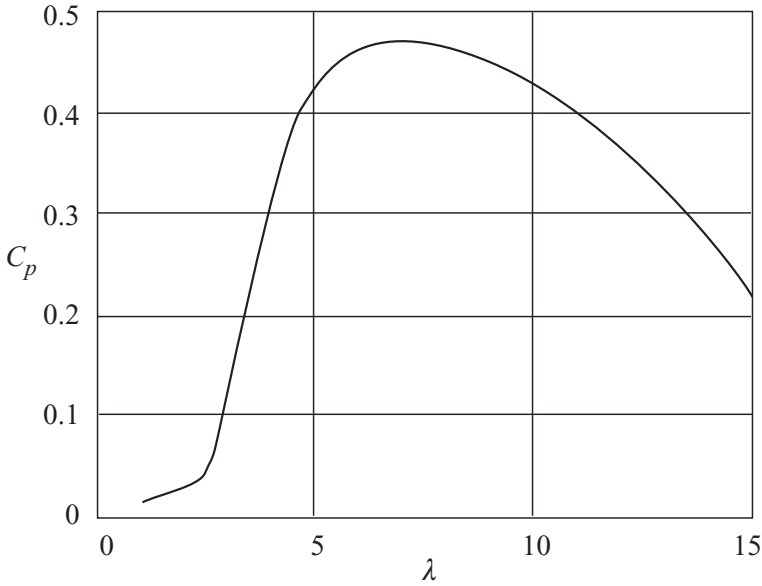


Figure 3.15 Power coefficient – tip speed ratio performance curve

The power coefficient is, therefore,

$$C_P = \frac{P}{\frac{1}{2} \rho U_{\infty}^3 \pi R^2}$$

Solving the blade element-momentum Equations 3.54 and 3.55 for a given, suitable blade geometrical and aerodynamic design yields a series of values for the power and torque coefficients which are functions of the tip speed ratio. A typical performance curve for a modern, high speed wind turbine is shown in Figure 3.15.

The maximum power coefficient occurs at a tip speed ratio for which the axial flow induction factor a , which in general varies with radius, approximates most closely to the Lanchester-Betz limit value of $\frac{1}{3}$. At lower tip speed ratios the axial flow induction factor can be much less than $\frac{1}{3}$ and aerofoil angles of attack are high leading to stalled conditions. For most wind turbines stalling is much more likely to occur at the blade root because, from practical constraints, the built-in pitch angle β of a blade is not large enough in that region. At low tip speed ratios blade stalling is the cause of a significant loss of power, as demonstrated in Figure 3.15. At high tip speed ratios a is high, angles of attack are low and drag begins to predominate. At both high and low tip speed ratios, therefore, drag is high and the general level of a is non-optimum so the power coefficient is low. Clearly, it would be best if a turbine can be operated at all wind speeds at a tip speed ratio close to that which gives the maximum power coefficient.

3.6 Breakdown of the momentum theory

3.6.1 Free-stream/wake mixing

For heavily loaded turbines, when a is high, the momentum theory predicts a reversal of the flow in the wake. Such a situation cannot actually occur, so, what happens is that the wake becomes turbulent and, in doing so, entrains air from outside the wake by a mixing process which re-energises the slow moving air that has passed through the rotor.

A rotor operating at increasingly high tip speed ratios presents a decreasingly permeable disc to the flow. Eventually, when λ is high enough for the axial flow factor to be equal to one, the disc effectively becomes a solid plate.

The flow past a solid disc, because of viscosity, separates at the disc's edge. A boundary layer develops as the flow over the front of the disc spreads out radially and by the time it reaches the edge viscosity has sapped much of the kinetic energy. As the boundary layer flows around the disc edge it accelerates causing a large drop in static pressure (Bernoulli). To flow around the disc edge would require very high velocity and there is insufficient static pressure to provide the necessary kinetic energy. The flow, therefore, separates from the disc and continues in the general stream-wise direction. In the region directly behind the disc there is slow moving, almost stagnant, air at low static pressure of the flow separating at the disc edge. At the front of the disc, at the very centre, the flow is brought to rest and so there is a large increase in static pressure as the kinetic energy is converted to pressure energy. Elsewhere on the front surface the flow moves radially with a velocity, outside the boundary layer, which increases towards the disc edge. The static pressure is generally higher on the front of the disc than on the rear and so the disc experiences a pressure drag force.

A similar process happens with a spinning rotor at high tip speed ratios. The air which does not pass through the rotor disc moves radially outwards and separates at the disc edge causing a low static pressure to develop behind the disc: the drop in static pressure caused by the separation increases as the tip speed ratio rises and the axial flow factor increases. The air which does pass through the rotor emerges into a low pressure region and is moving slowly. There is insufficient kinetic energy to provide the rise in static pressure necessary to achieve the ambient atmospheric pressure that must exist in the far wake. The air can only achieve atmospheric pressure by gaining energy from the mixing process in the turbulent wake. The shear layer in the flow between the free-stream air and the wake air is what becomes of the boundary layer that develops on the front of the disc. The shear layer is unstable and breaks up into the turbulence that causes the mixing and re-energisation of the wake air.

3.6.2 Modification of rotor thrust caused by flow separation

The low static pressure downstream of the rotor disc caused by the separation of the free-stream flow at the edge of the disc and the high static pressure at the stagnation point on the upstream side causes a large thrust on the disc, much larger than that predicted by the momentum theory. Some experimental results reported by Glauert (1926), for a whole rotor, can be seen in Figure 3.16 where the simple expression for the thrust force coefficient, as derived from the momentum theory $C_T = 4a(1 - a)$, is given for comparison.

The thrust (or drag) coefficient for a simple, flat circular plate is given by Hoerner (1965) as 1.17 but, as demonstrated in figure 3.16, the thrust on the rotating disc is higher. It might have been expected that when $a = 1$ the rotor would have the same thrust coefficient as the

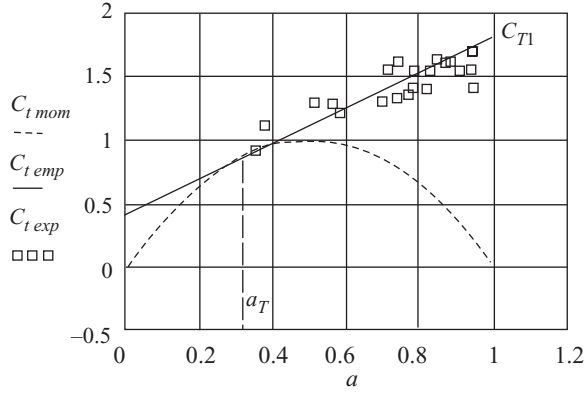


Figure 3.16 Comparison of theoretical and measured values of C_T

circular plate. The principal difference between the circular plate and the rotor is that the latter is rotating and, as Hoerner also describes, this causes energy to be dissipated in a thicker, rotating boundary layer on the upstream surface of the rotor disc giving rise to an even lower pressure on the downstream side.

It would follow from the above arguments that for high values of the axial induction factor most of the pressure drop across the disc is not associated with blade circulation, just as it is absent in the case of the circular plate. Circulation would cause a small pressure drop similar to that given by the momentum theory because it would be determined by the very low axial velocity of the flow which actually permeates the disc.

3.6.3 Empirical determination of thrust coefficient

A suitable straight line through the experimental points would appear to be possible, although Glauert proposed a parabolic curve, and provides an empirical solution to the problem of the thrust on a heavily loaded turbine (a rotor operating at a high value of the axial flow induction factor).

Most authors assume that the entire thrust on the rotor disc is associated with axial momentum change. Therefore, for the empirical line to be useful it must be assumed that it applies not only to the whole rotor but also to each separate stream-tube. Let C_{T1} be the empirical value of C_T when $a = 1$. Then, as the straight line must be a tangent to the momentum theory parabola at the transition point, the equation for the line is

$$C_T = C_{T1} - 4(\sqrt{C_{T1}} - 1)(1 - a) \quad (3.58)$$

and the value of a at the transition point is

$$a_T = 1 + \frac{1}{2}\sqrt{C_{T1}}$$

By inspection, C_{T1} must lie between 1.6 and 2: $C_{T1} = 1.816$ would appear to be the best fit to the experimental data of Figure 3.16, whereas Wilson and Lissaman (1974) favour the lower value of $C_{T1} = 1.6$. Glauert fits a parabolic curve to the data giving much higher values

of C_{T1} at high values of a but he was considering the case of an *airscrew* in the windmill brake state where the angles of attack are negative.

The flow field through the turbine under heavily loaded conditions cannot be modelled easily and the results of this empirical analysis must be regarded as being only approximate at best. They are, nevertheless, better than those predicted by the momentum theory. For most practical designs the value of axial flow induction factor rarely exceeds 0.6 and for a well designed blade will be in the vicinity of 0.33 for much of its operational range.

For values of a greater than a_T it is common to replace the momentum theory thrust in Equation 3.9 with Equation 3.58, in which case Equation 3.54 is replaced by

$$(1 - a)^2 \frac{\sigma_r}{\sin^2 \phi} C_x + 4(\sqrt{C_{T1}} - 1)(1 - a) - C_{T1} = 0 \quad (3.54a)$$

However, as the additional pressure drop is caused by edge flow separation then this course of action is questionable and it may be more appropriate to retain Equation 3.53.

3.7 Blade geometry

3.7.1 Introduction

The purpose of most wind turbines is to extract as much energy from the wind as possible and each component of the turbine has to be optimised for that goal. Optimal blade design is influenced by the mode of operation of the turbine, that is, fixed rotational speed or variable rotational speed and, ideally, the wind distribution at the intended site. In practice engineering compromises are made but it is still necessary to know what would be the best design.

Optimising a blade design means maximising the power output and so a suitable solution to blade element-momentum equations (3.54 and 3.55) is necessary.

3.7.2 Optimal design for variable speed operation

A turbine operating at variable speed can maintain the constant tip speed ratio required for the maximum power coefficient to be developed regardless of wind speed. To develop the maximum possible power coefficient requires a suitable blade geometry the conditions for which will now be derived.

For a chosen tip speed ratio λ the torque developed at each blade station is maximised if

$$8\pi\lambda\mu^2 \frac{d}{da'} a'(1 - a) = 0$$

giving

$$\frac{da}{da'} = \frac{1 - a}{a'} \quad (3.59)$$

From Equations 3.51 and 3.53a a relationship between the flow induction factors can be obtained.

Dividing Equations 3.51 and 3.53a

$$\frac{\frac{C_l}{C_d} \tan \phi - 1}{\frac{C_l}{C_d} + \tan \phi} = \frac{\lambda \mu a'(1 - a)}{a(1 - a) + (a' \lambda \mu)^2} \quad (3.60)$$

The flow angle ϕ is given by

$$\tan \phi = \frac{1 - a}{\lambda \mu (1 + a')} \quad (3.61)$$

Substituting Equation 3.61 into Equation 3.60 gives

$$\frac{\frac{C_l}{C_d} \frac{1 - a}{\lambda \mu (1 + a')} - 1}{\frac{C_l}{C_d} + \frac{1 - a}{\lambda \mu (1 + a')}} = \frac{\lambda \mu a'(1 - a)}{a(1 - a) + (a' \lambda \mu)^2}$$

Simplifying,

$$\begin{aligned} & \left[\frac{C_l}{C_d} (1 - a) - \lambda \mu (1 + a') \right] [a(1 - a) + (a' \lambda \mu)^2] \\ &= \left[\lambda \mu (1 + a') \frac{C_l}{C_d} + (1 - a) \right] \lambda \mu a' (1 - a) \end{aligned} \quad (3.62)$$

At this stage the process is made easier to follow if drag is ignored, Equation 3.62 then reduces to

$$a(1 - a) - \lambda^2 \mu^2 a' = 0 \quad (3.62a)$$

Differentiating Equation 3.62a with respect to a' gives

$$(1 - 2a) \frac{da}{da'} - \lambda^2 \mu^2 = 0 \quad (3.63)$$

and substituting Equation 3.59 into 3.63

$$(1 - 2a)(1 - a) - \lambda^2 \mu^2 a' = 0 \quad (3.64)$$

Equations 3.62a and 3.64, together, give the flow induction factors for optimised operation

$$a = \frac{1}{3} \quad \text{and} \quad a' = \frac{a(1 - a)}{\lambda^2 \mu^2} \quad (3.65)$$

that agree exactly with the momentum theory prediction because no losses, such as aerodynamic drag, have been included and the number of blades is assumed to be large: every fluid

particle which passes through the rotor disc interacts with a blade resulting in a uniform axial velocity over the area of the disc.

To achieve the optimum conditions the blade design has to be specific and can be determined from either of the fundamental Equations 3.51 and 3.53. Choosing Equation 3.53, because it is the simpler, and ignoring the drag the torque developed in optimised operation is

$$\delta Q = 4\pi\rho U_\infty \Omega r a'(1-a)r^2 \delta r = 4\pi\rho \frac{U_\infty^3}{\Omega} a(1-a)^2 r \delta r$$

The component of the lift per unit span in the tangential direction is, therefore,

$$L \sin \phi = 4\pi\rho \frac{U_\infty^3}{\Omega} a(1-a)^2$$

By the Kutta-Joukowski theorem the lift per unit span is

$$L = \rho W \Gamma$$

where Γ is the sum of the individual blade circulations and W is the component of incident velocity mutually perpendicular to both Γ and L .

Consequently,

$$\rho W \Gamma \sin \phi = \rho \Gamma U_\infty (1-a) = 4\pi\rho \frac{U_\infty^3}{\Omega} a(1-a)^2 \quad (3.66)$$

so

$$\Gamma = 4\pi \frac{U_\infty^2}{\Omega} a(1-a) \quad (3.67)$$

The circulation is therefore uniform along the blade span and this is a condition for optimised operation.

To determine the blade geometry, that is, how should the chord size vary along the blade and what pitch angle β distribution is necessary, we must return to Equation 3.53a.

$$\frac{W^2}{U_\infty^2} B \frac{c}{R} C_l \sin \phi = 8\pi\lambda\mu^2 a'(1-a)$$

substituting for $\sin \phi$ gives

$$\frac{W}{U_\infty} B \frac{c}{R} C_l (1-a) = 8\pi\lambda\mu^2 a'(1-a) \quad (3.68)$$

From which is derived

$$\frac{B}{2\pi} \frac{c}{R} = \frac{4\lambda\mu^2 a'}{\frac{W}{U_\infty} C_l}$$

The only unknown on the right hand side of the above equation is the value of the lift coefficient C_l and so it is common to include it on the left side of the equation with the chord solidity as a blade geometry parameter. The lift coefficient can be chosen as that value which corresponds to the maximum lift/drag ratio C_l/C_d as this will minimise drag losses: even though drag has been ignored in the determination of the optimum flow induction factors and blade geometry it cannot be ignored in the calculation of torque and power. Blade geometry also depends upon the tip speed ratio λ so it is also included in the blade geometry parameter.

Hence,

$$\sigma_r \lambda C_l = \frac{B}{2\pi} \frac{c}{R} \lambda C_l = \frac{4\lambda^2 \mu^2 a'}{\sqrt{(1-a)^2 + (\lambda\mu(1+a'))^2}} \quad (3.69)$$

Introducing the optimum conditions of Equation 3.65

$$\sigma_r \lambda C_l = \frac{\frac{8}{9}}{\sqrt{\left[1 - \frac{1}{3}\right]^2 + \lambda^2 \mu^2 \left[1 + \frac{2}{9\lambda^2 \mu^2}\right]^2}} \quad (3.69a)$$

The parameter $\lambda\mu$ is called the local speed ratio and is equal to the tip speed ratio where $\mu = 1$.

If, for a given design, C_l is held constant then Figure 3.17 shows the blade plan-form for increasing tip speed ratio. A high design tip speed ratio would require a long, slender blade (high aspect ratio) whilst a low design tip speed ratio would need a short, fat blade. The design tip speed ratio is that at which optimum performance is achieved. Operating a rotor at other than the design tip speed ratio gives a less than optimum performance, even in ideal drag free conditions.

In off-optimum operation the axial inflow factor is not uniformly equal to $\frac{1}{3}$, in fact it is not uniform at all.

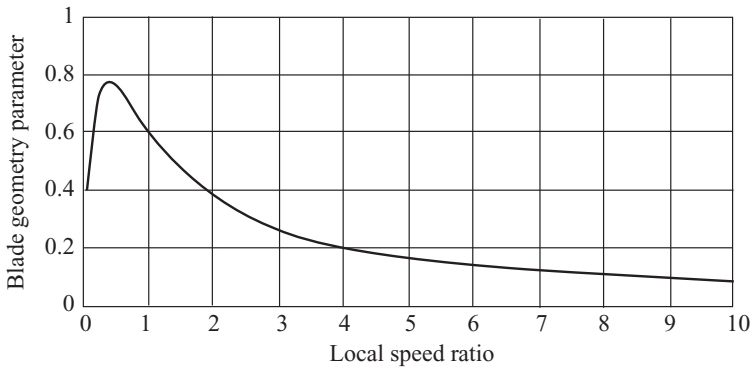


Figure 3.17 Variation of blade geometry parameter with local speed ratio

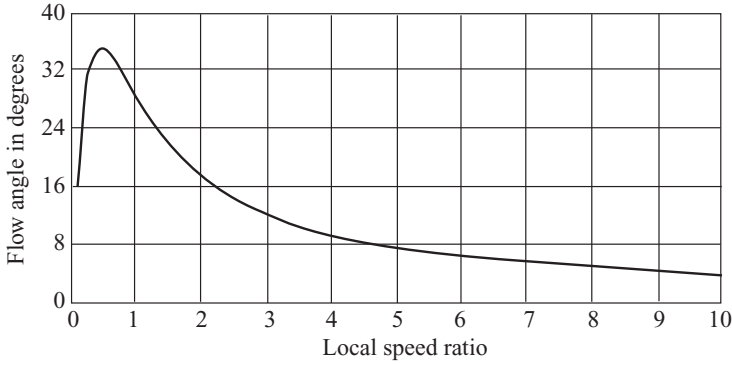


Figure 3.18 Variation of inflow angle with local speed ratio

The local inflow angle ϕ at each blade station also varies along the blade span as shown in Equation 3.70 and Figure 3.18.

$$\tan \phi = \left[\frac{1 - a}{\lambda \mu (1 + a')} \right] \quad (3.70)$$

which, for optimum operation, is

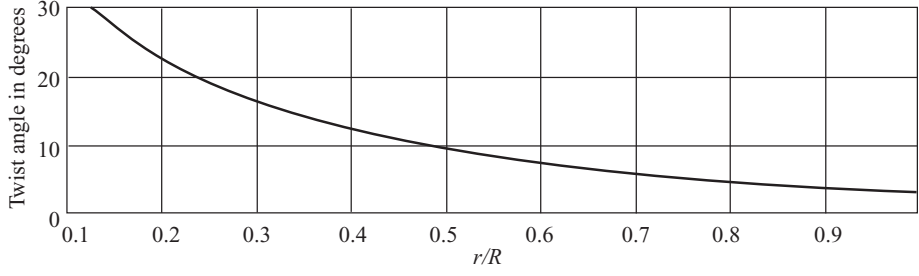
$$\tan \phi = \left[\frac{1 - \frac{1}{3}}{\lambda \mu \left(1 + \frac{2}{9\lambda^2 \mu^2} \right)} \right] \quad (3.70a)$$

Close to the blade root the inflow angle is large which could cause the blade to stall in that region. If the lift coefficient is to be held constant such that drag is minimised everywhere then the angle of attack α also needs to be uniform at the appropriate value. For a prescribed angle of attack variation the design pitch angle $\beta = \phi - \alpha$ of the blade must vary accordingly.

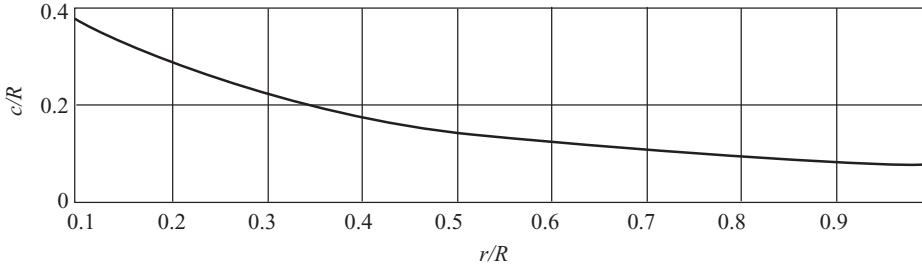
As an example, suppose that the blade aerofoil is NACA 4412, popular for hand-built wind turbines because the bottom (high pressure) side of the profile is almost flat which facilitates manufacture. At a Reynolds number of about $5 \cdot 10^5$ the maximum lift/drag ratio occurs at a lift coefficient of about 0.7 and an angle of attack of about 3° . Assuming that both C_l and α are to be held constant along each blade and there are to be three blades operating at a tip speed ratio of 6 then the blade design in plan-form and pitch (twist) variation are shown in Figures 3.19a,b, respectively.

3.7.3 A simple blade design

The blade design of Figure 3.19 is efficient but complex to build and, therefore, costly. Suppose the plan-form was prescribed to have a uniform taper such that the outer part of the blade corresponds closely to Figure 3.19a. A straight line drawn through the 70% and 90% span points as shown in Figure 3.20 not only simplifies the plan-form but removes a lot of material close to the root.



(a)



(b)

Figure 3.19 Optimum blade design for three blades and $\lambda = 6$

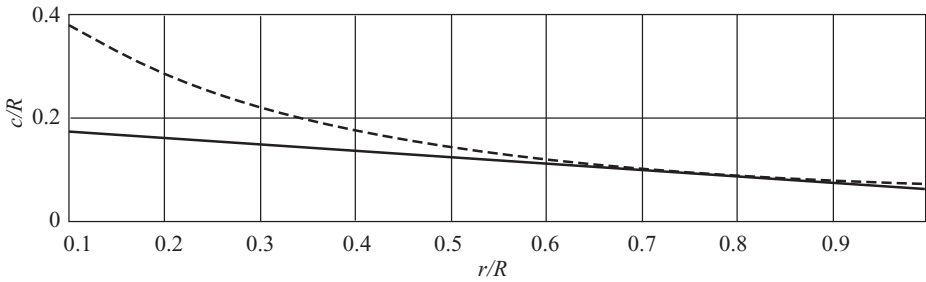
The expression for the new plan-form is

$$c_u = \frac{8}{9 \cdot 0.8\lambda} \left(2 - \frac{\lambda\mu}{0.8\lambda} \right) \frac{2\pi}{C_l \lambda B} \quad (3.71)$$

The 0.8 in Equation 3.71 refers to the 80% point, midway between the target points.

Equations 3.69a and 3.71 can be combined to give the required span-wise variation of C_l for optimal operation.

$$C_l = \frac{8}{9} \frac{1}{\frac{Bc_u\lambda}{2\pi} \sqrt{\left(1 - \frac{1}{3}\right)^2 + \lambda^2\mu^2 \left(1 + \frac{2}{9\lambda^2\mu^2}\right)^2}}$$


Figure 3.20 Uniform taper blade design for optimal operation

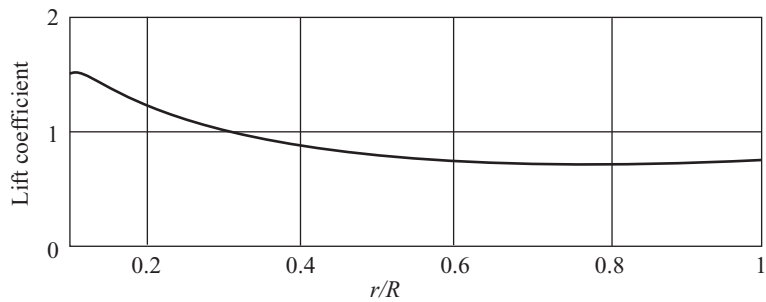


Figure 3.21 Span-wise distribution of the lift coefficient required for the linear taper blade

Close to the blade root the lift coefficient approaches the stalled condition and drag is high but the penalty is small because the adverse torque is small in that region (see Figure 3.21).

Assuming that stall does not occur and that for the aerofoil in question, which has a 4% camber (this approximates to a zero lift angle of attack of -4°) the lift coefficient is given approximately by

$$C_l = 0.1(\alpha + 4^\circ)$$

where α is in degrees and 0.1 is a good approximation to the gradient of the C_l vs. α for most aerofoils, so

$$\alpha = \frac{C_l}{0.1} - 4^\circ$$

The blade twist distribution can now be determined from Equations 3.70a and 3.45.

The twist angle close to the root is still high but lower than for the constant C_l blade (see Figure 3.22).

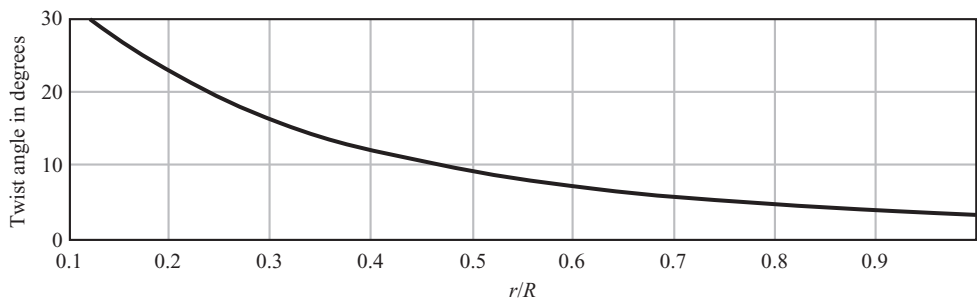


Figure 3.22 Span-wise distribution of the twist required for the linear taper blade

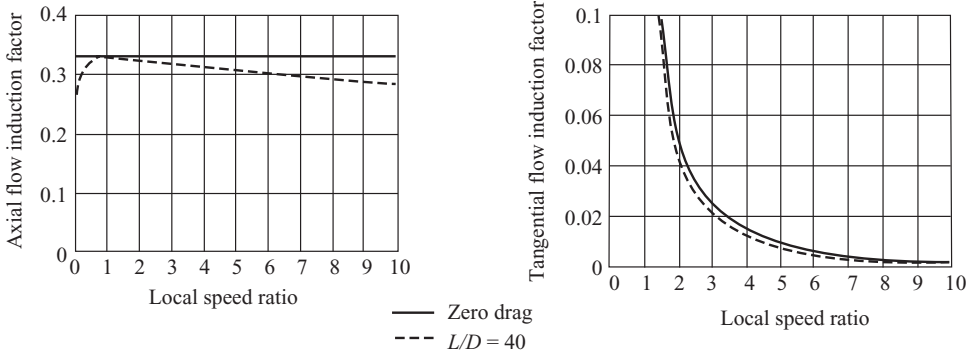


Figure 3.23 Radial variation of the flow induction factors with and without drag

3.7.4 Effects of drag on optimal blade design

If, despite the views of Wilson and Lissaman (1974) (see Section 3.5.3), the effects of drag are included in the determination of the flow induction factors we must return to Equation 3.48 and follow the same procedure as described for the drag free case.

In the current context the effects of drag are dependent upon the magnitude of the lift/drag ratio which, in turn, depends on the aerofoil profile but largely on Reynolds number and on the surface roughness of the blade. A high value of lift/drag ratio would be about 150, whereas a low value would be about 40.

Unfortunately, with the inclusion of drag, the algebra of the analysis is complex. Polynomial equations have to be solved for both a and a' . The details of the analysis are left for the reader to discover.

In the presence of drag, the axial flow induction factor for optimal operation is not uniform over the disc as it is in the hypothetical drag free situation. However, the departure of the axial flow distribution from uniformity is not great, even when the lift/drag ratio is low provided the flow around a blade remains attached.

The radial variation of the axial and tangential flow induction factors is shown in Figure 3.23 for zero drag and for a lift/drag ratio of 40. The tangential flow induction factor is lower in the presence of drag than without because the blade drags the fluid around in the direction of rotation, opposing the general rotational reaction to the shaft torque.

From the torque/angular momentum Equation 3.53 the blade geometry parameter becomes

$$\frac{B}{2\pi} \frac{c}{R} \lambda C_l = \frac{4\lambda^2 \mu^2 a' (1 - a)}{\frac{W}{U_\infty} \left[(1 - a) - \frac{C_l}{C_d} \lambda \mu (1 + a') \right]} \quad (3.72)$$

Figure 3.24 compares the blade geometry parameter distributions for zero drag and a lift/drag ratio of 40 and, as is evident, drag has very little effect on blade optimal design.

A similar result is apparent in Figure 3.25 for the inflow angle, Figure 3.25, distribution where drag also has little influence.

As far as blade design for optimal operation is concerned drag can be ignored, greatly simplifying the process.

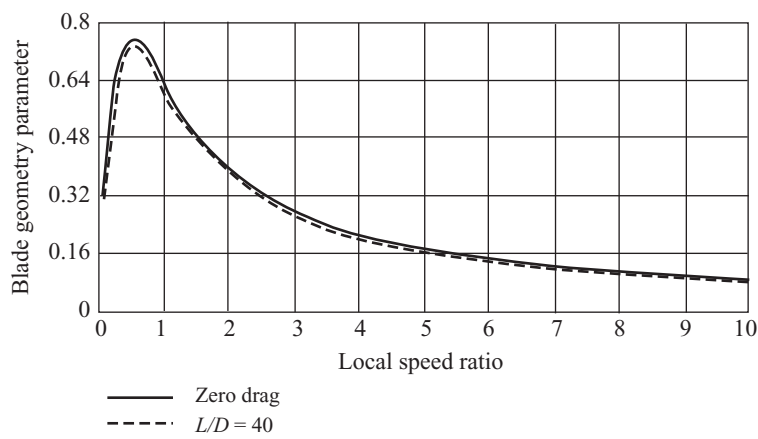


Figure 3.24 Span-wise variation of the blade geometry parameter with and without drag

The results of Equation 3.57 showing the maximum power coefficients for a range of design tip speed ratios and several lift/drag ratios is shown in Figure 3.26. The flow induction factors have been determined without drag (Equations 3.54 and 3.55) but the torque has been calculated using Equation 3.57, which includes drag. The losses caused by drag are significant and increase with increasing design tip speed ratio. As will be shown later, when tip-losses are also taken into account, the losses at low tip speed ratios are even greater.

3.7.5 Optimal blade design for constant speed operation

If the rotational speed of a turbine is maintained at a constant level then the tip speed ratio is continuously changing and a blade optimised for a fixed tip speed ratio would not be appropriate.

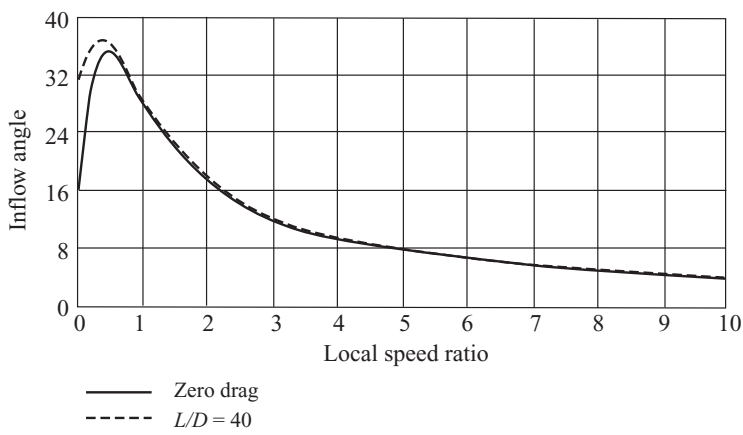


Figure 3.25 Variation of inflow angle with local speed ratio with and without drag.

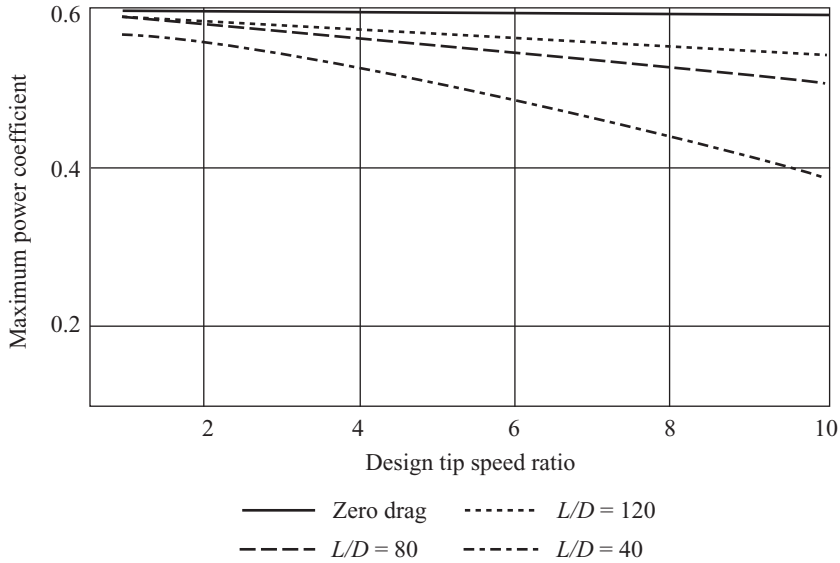


Figure 3.26 The variation of maximum C_p with design λ for various lift/drag ratios.

No simple technique is available for the optimal design of a blade operating at constant rotational speed. A non-linear programming method could be applied by maximising energy capture at a site with a specified wind speed distribution. Alternatively, a design tip speed ratio could be chosen which corresponds to the wind speed at the specified site which contains the most energy or, more practically, the pitch angle for the whole blade can be adjusted to maximise energy capture.

3.8 The effects of a discrete number of blades

3.8.1 Introduction

The analysis described in all prior sections assumes that there is a sufficient number of blades on the rotor for every fluid particle passing through the rotor disc to interact with a blade, that is, that all fluid particles undergo the same loss of momentum. With a small number of blades some fluid particles will interact with them but most will pass between the blades and, clearly, the loss of momentum by a particle will depend on its proximity to a blade as the particle passes through the rotor disc. The axial induced velocity will, therefore, at any instant, vary around the disc, the average value determining the overall axial momentum of the flow and the larger value local to a blade determining the forces on the blade.

3.8.2 Tip-losses

If the axial flow induction factor a is large at the blade position then, by Equation 3.44, the inflow angle ϕ will be small and the lift force will be almost normal to the rotor plane. The component of the lift force in the tangential direction will be small and so will be its

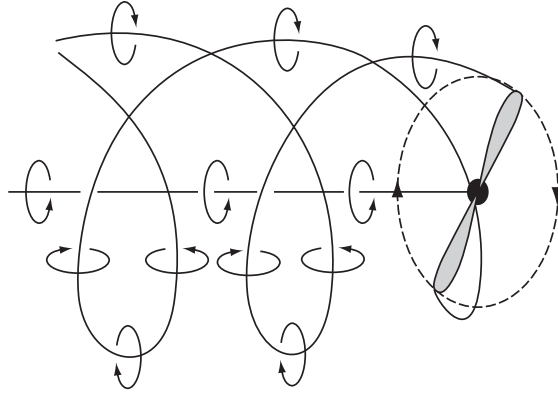


Figure 3.27 Helical trailing tip vortices of a horizontal axis turbine wake

contribution to the torque. A reduced torque means reduced power and this reduction is known as tip-loss because the effect occurs only at the outermost parts of the blades.

In order to account for tip-losses, the manner in which the axial flow induction factor varies azimuthally needs to be known but, unfortunately, this requirement is beyond the abilities of the blade element-momentum theory.

Just as a vortex trails from the tip of an aircraft wing so does a vortex trail from the tip of a wind turbine blade. Because the blade tip follows a circular path it leaves a trailing vortex as a helical structure which convects downstream with the wake velocity. For a two blade rotor, unlike an aircraft wing, the bound circulations on the two blades shown in Figure 3.27 are opposite in sign and so combine to shed a straight line vortex along the rotational axis with strength equal to the blade circulation times the number of blades.

For a single vortex to be shed from the blade tip the circulation strength along the blade span must be uniform and, as has been shown, uniform circulation is a requirement for optimised operation. However, the uniform circulation requirement assumes that the axial flow induction factor is uniform across the disc and, as has been argued above, with discrete blades rather than a uniform disc the flow factor is not uniform.

In the case of Figure 3.27, close to the blade tips the tip vortex causes very high values of the flow factor a such that, locally, the net flow past the blade is in the upstream direction. The average value of a , azimuthally, is radially uniform which means that if high values occur in the vicinity of the blades then low values occur elsewhere. The azimuthal variation of a for a number of radial positions is shown in Figure 3.28 for a three blade rotor operating at a tip speed ratio of six. The calculation for Figure 3.28 assumes a discrete vortex for each blade with a constant pitch and constant radius helix.

At a particular radial position the ratio of the azimuthal average of a to the value at the azimuthal position where the blade is centred is shown in Figure 3.29, being unity for most of the blade span and only near the tip does it begin to fall to zero. The ratio is called ‘the tip-loss factor’.

From Equation 3.20 and in the absence of tip-loss and drag the contribution of each blade element to the overall power coefficient is

$$\delta C_P = 8\lambda^2 \mu^3 a'(1-a)\delta\mu \quad (3.73)$$

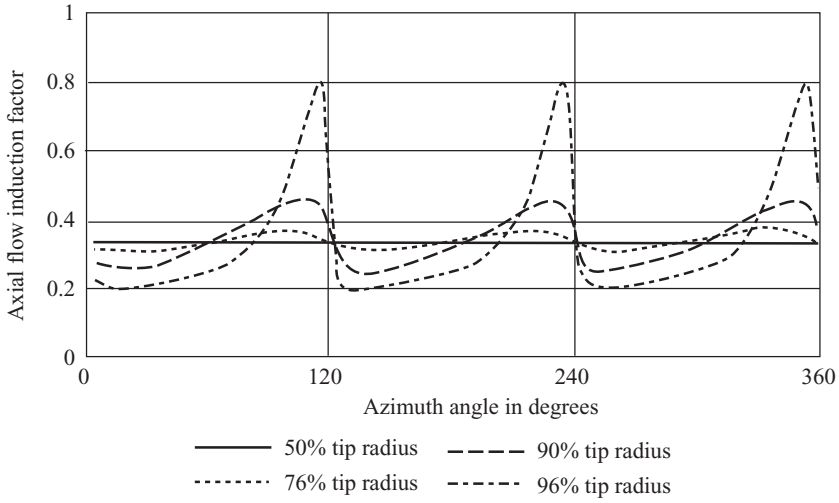


Figure 3.28 Azimuthal variation of a for various radial positions for a three blade rotor with uniform blade circulation operating at a tip speed ratio of six. The blades are at 120° , 240° and 360° .

Substituting for a' from Equation 3.25 gives

$$\delta C_P = 8\mu a(1 - a)^2 \delta\mu \quad (3.74)$$

Whereas, from the Kutta-Joukowski theorem, the circulation Γ on the blade, which is uniform, provides a torque per unit span of

$$\frac{dQ}{dr} = \rho(W \times \Gamma) \sin \phi_r r$$

where the angle ϕ_r is determined by the flow velocity local to the blade.

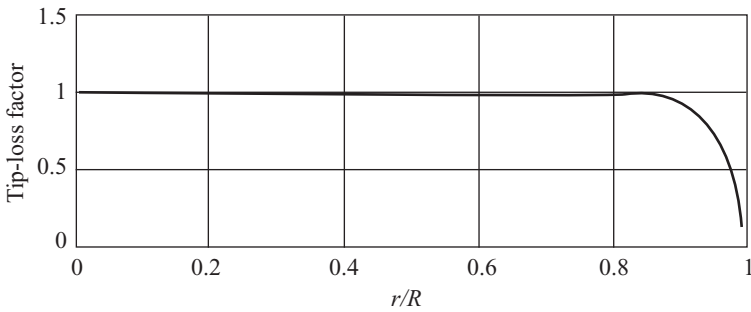


Figure 3.29 Span-wise variation of the tip-loss factor for a blade with uniform circulation

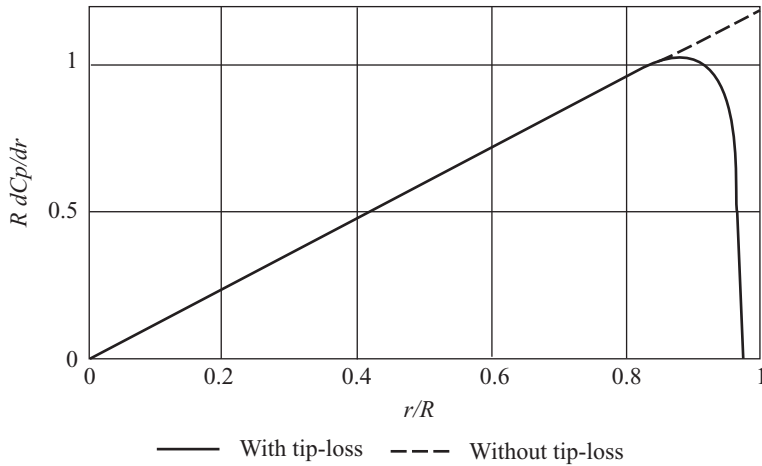


Figure 3.30 Span-wise variation of power extraction in the presence of tip-loss for a blade with uniform circulation on a three-blade turbine operating at a tip speed ratio of six

The strength of the total circulation for all three blades is given by Equation 3.67 and so, in the presence of tip-loss, the increment of power coefficient from a blade element is

$$\delta C_P = 8\mu a(1 - a)(1 - a_r)\delta\mu \quad (3.75)$$

where $a = \frac{1}{3}$ is the average axial flow induction factor and a_r is the value local to the blade, as described in Figure 3.33. In Equation 3.75 the term $(1 - a)$ refers to the mass flow through the annulus swept by a blade element and the term $(1 - a_r)$ refers to the loss of momentum experienced by the flow.

The results from Equations 3.73 and 3.74 are plotted in Figure 3.30 and clearly show the effect of tip-loss. Equation 3.70 assumes that $a = \frac{1}{3}$ uniformly over the whole disc, Equation 3.72 recognises that a is not uniform. The azimuthally averaged value of a is equal to $\frac{1}{3}$ at every radial position but the azimuth variation gives rise to the tip-loss. The blade does not extract energy from the flow efficiently because a varies. Imagine the disc comprising a myriad of elemental discs, each with its own independent stream-tube, and not all of them operating at the Lanchester-Betz limit. Note that the power loss to the wind is exactly the same as that extracted by the blades, there is no effective drag associated with tip-loss.

With uniform circulation the azimuthal average value of a is also radially uniform, but that implies a discontinuity of axial velocity at the wake boundary with a corresponding discontinuity in pressure. Whereas such discontinuities are acceptable in the idealised actuator disc situation, they will not occur in practice with a finite number of blades. If it is assumed that a is zero outside of the wake then a must fall to zero in a regular fashion towards the blade tips and, consequently, the bound circulation must also fall to zero. The manner in which the circulation varies at the tip will be governed by the blade tip design, that is, the chord and pitch variation, and there will be a certain design which will minimise the tip-loss.

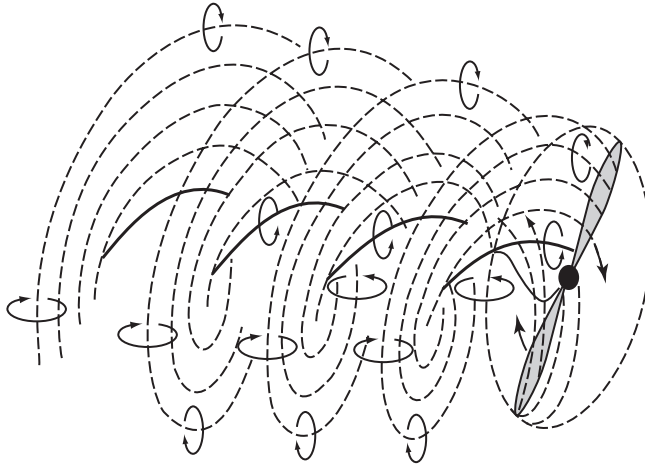


Figure 3.31 A helicoidal vortex sheet wake for a two-blade rotor

If the circulation varies along the blade span vorticity is shed into the wake in a continuous fashion from the trailing edge.

Therefore, each blade sheds a helicoidal sheet of vorticity, as shown in Figure 3.31, rather than a single helical vortex as shown in Figure 3.27. The helicoidal sheets convect with the wake velocity and so there can be no flow across the sheets, which can therefore be regarded as impermeable. The intensity of the vortex sheets is equal to the rate of change of bound circulation along the blade span and so increases rapidly towards the blade tips. There is flow around the blade tips because of the pressure difference between the blade surfaces, which means that on the upwind surface of the blades the flow moves towards the tips and on the downwind surface the flow move towards the root. The flows from either surface leaving the trailing edge of a blade will not be parallel to one another and will form a surface of discontinuity of velocity in a radial sense within the wake; the axial velocity components will be equal. The surface of discontinuity is called a vortex sheet. A similar phenomenon occurs with aircraft wings and a textbook of aircraft aerodynamics will explain it in greater detail.

A deeper understanding of the mechanism of tip-loss can be obtained by following the path of air particles. An air particle approaches the spinning rotor, ‘senses’ high pressure ahead and slows down accordingly. The high pressure on the upwind side of the rotor blades is effectively smeared around the whole disc. Slowing down also causes the particle to move outwards to maintain the mass flow rate. When the particle reaches the rotor plane it will either be close to a blade or not and its axial velocity will be affected accordingly, as shown in Figure 3.28. If the particle passes through the rotor plane close to blade then it will also be strongly affected by the blade’s pressure field.

As a particle passes close to and in front of a blade it will leave the trailing edge having accelerated in the tangential direction; it will then pass downstream, on the upwind side of the vortex sheet being shed from the trailing edge and so will also be moving radially outwards. The particle, therefore, migrates outward to the edge of the vortex sheet around which it is swept on to the downwind side and migrates inward with a radial velocity which

reduces to zero at a radial point on the sheet where the shed vorticity is zero. The particle then continues downstream with the velocity of the axial and tangential velocities of the vortex sheet.

A second particle which passes a blade close to the downwind, low pressure, surface is accelerated tangentially in the opposite direction to the blade motion and then slows down, leaving the trailing edge with the same axial and tangential velocity components as the first particle but on the downwind side of the vortex sheet so it will have, in addition, a radially inwards velocity. The second particle will, depending on its radial position, migrate inwards until the radial velocity becomes zero.

A third particle which passes between two blades will be moving axially at a greater velocity than the first two particles, will not be strongly affected by the pressure fields of the blades but, because of the solid blockage presented by the blades (see Figure 3.5) will be directed into a helical path. Being faster than the vortex sheet ahead, the particle will begin to catch it up and as it does so the influence of the vortex sheet will move it outwards, around the edge of the sheet and then inwards, just like the first particle. Unlike the first particle, however, the third particle will still be moving faster than the vortex sheet and so will move axially away from the sheet, approaching the next sheet downstream and repeating the motion around the edge of that sheet. The particle will proceed downstream overtaking and hopping around each vortex sheet in turn.

The third particle does not lose as much axial momentum as particles one and two and is, therefore, affected by the so-called tip loss. The affect is greater the closer the third particle is to the edge of the rotor disc as it passes through the disc.

A fourth particle passes between the blades but at a radial position, closer to the axis of rotation, where its axial velocity is equal to that of the vortex sheets. If the particle passes midway, say, between two blades then it remains midway between the two corresponding vortex sheets as it moves downstream and does not undergo any radial motion other than the general expansion caused by the slowing down of the flow.

The fourth particle is totally unaffected by the fact that there is a finite number of blades and follows the same progress as if it were passing through a uniform actuator disc.

The axial flow induction factor varies, therefore, not only azimuthally but also radially, is a function of both r and θ . The azimuthally averaged value of $a(r) = a_b(r)f(r)$, where $f(r)$ is known as the tip-loss factor, has a value of unity inboard (particle four) and falls to zero at the edge of the rotor disc. The value $a_b(r)$ is the level of axial flow induction factor that occurs locally at a blade element and is the velocity with which the vortex sheet convects downstream. If $a_b(r)$ can be held radially uniform then the vortex sheets will be radially flat, as shown in Figure 3.31, but if $a_b(r)$ is not uniform the vortex sheets will warp.

In the application of the blade element-momentum theory it is argued that the rate of change of axial momentum is determined by the azimuthally averaged value of axial flow induction factor, whereas the blade forces are determined by the value of the flow factor which occurs locally at the blade element, that experienced by the first and second particles.

The mass flow rate through an annulus = $\rho U_\infty(1 - a_b(r)f(r))2\pi r\delta r$.

The azimuthally averaged overall change of axial velocity = $2a_b(r)f(r)U_\infty$.

The rate of change of axial momentum = $4\pi r\rho U_\infty^2(1 - a_b(r)f(r))a_b(r)f(r)\delta r$.

The blade element forces are $\frac{1}{2}\rho W^2 BcC_l$ and $\frac{1}{2}\rho W^2 BcC_d$ where W and C_l are determined using $a_b(r)$.

The pressure force caused by the rotation of the wake is also calculated using an azimuthally averaged value of the tangential flow induction factor $2a'_b(r)f(r)$.

3.8.3 Prandtl's approximation for the tip-loss factor

The function for the tip-loss factor $f(r)$ is shown in Figure 3.29 for a blade with uniform circulation operating at a tip speed ratio of six and is not readily obtained by analytical means for any desired tip speed ratio. Sidney Goldstein (1929) did analyse the tip-loss problem for application to propellers and achieved a solution in terms of Bessel functions, but neither that nor the Biot-Savart solution used above is suitable for inclusion in the blade element-momentum theory. Fortunately, in 1919, Ludwig Prandtl, reported by Betz (1919), had already developed an ingenious approximate solution which does yield a relatively simple analytical formula for the tip-loss function.

Prandtl's approximation was inspired by the fact that, being impermeable (particles one and three in the above description pass around the outer edge of a sheet but not through it), the vortex sheets could be replaced by material sheets which, provided they move with the velocity dictated by the wake, would have no effect upon the wake flow. The theory applies only to the developed wake. In order to simplify his analysis Prandtl replaced the helicoidal sheets with a succession of discs, moving with the uniform, central wake velocity $U_\infty(1 - a)$ and separated by the same distance as the normal distance between the vortex sheets. Conceptually, the discs, travelling axially with velocity $U_\infty(1 - a)$ would encounter the unattenuated free-stream velocity U_∞ at their outer edges. The fast flowing free-stream air would tend to weave in and out between successive discs. The wider apart successive discs the deeper, radially, the free-stream air would penetrate. Taking any line parallel to the rotor axis at a radius r , somewhat smaller than the wake radius (rotor radius), the average axial velocity along that line would be greater than $U_\infty(1 - a)$ and less than U_∞ . Let the average velocity be $U_\infty(1 - af(r))$, where $f(r)$ is the tip-loss function, has a value less than unity and falls to zero at the wake boundary. At a distance from the wake edge the free-stream fails to penetrate and there is little or no difference between the induced velocity at the blade and that in the wake, that is, $f(r) = 1$.

A particle path, as shown in Figure 3.32, is very similar to that described for particle three, above, and may be interpreted as that of the *average* particle passing through the rotor disc at

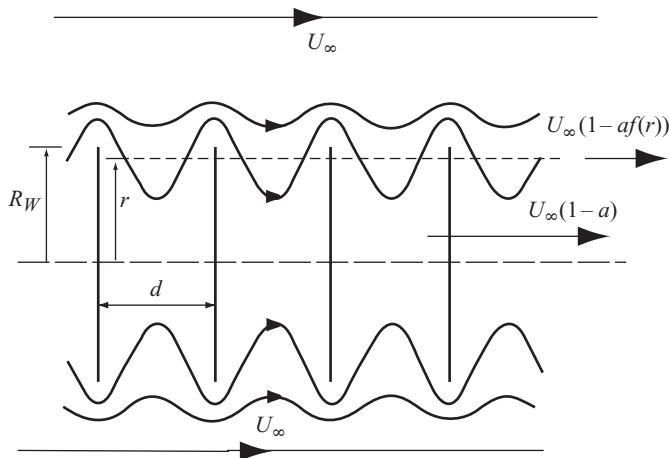


Figure 3.32 Prandtl's wake-disc model to account for tip-losses

a given radius in the actual situation: the azimuthal variations of particle velocities at various radii are shown in Figure 3.28 and a ‘Prandtl particle’ would have a velocity equal to the average of a variation. Figure 3.32 depicts the developed wake.

Prandtl’s approximation defines quite well the downstream behaviour of particle three above, which passes the rotor plane between two blades.

The mathematical detail of Prandtl’s analysis (see Glauert, 1935a) is beyond the scope of this text but, unlike Goldstein’s theory, the result can be expressed in closed solution form; the tip-loss factor is given by

$$f(r) = \frac{2}{\pi} \cos^{-1} \left(e^{-\pi \left(\frac{R_w - r}{d} \right)} \right) \quad (3.76)$$

$R_w - r$ is a distance measured from the wake edge. Distance d between the discs should be that of the distance travelled by particle three between successive vortex sheets. Glauert (1935a) takes d as being the normal distance between successive helicoidal vortex sheets.

The helix angle of the vortex sheets is the flow angle ϕ_s and so with B sheets intertwining from B blades

$$d = \frac{2\pi R_w}{B} \sin \phi_s = \frac{2\pi R_w}{B} \frac{U_\infty(1-a)}{W_s} \quad (3.77)$$

Prandtl’s model has no wake rotation but the discs may spin at the rotor speed without affecting the flow at all, as it is inviscid, thus, a' is zero and W_s is the resultant velocity (not including the radial velocity) at the edge of a disc. Glauert (1935a) argues that $R_w/W_s \approx r/W$, which is more convenient to use.

$$W = \sqrt{[U_\infty(1-a)]^2 + (r\Omega)^2}$$

so

$$\pi \left(\frac{R_w - r}{d} \right) = \frac{B}{2} \left(\frac{R - r}{r} \right) \sqrt{1 + \frac{(r\Omega)^2}{[U_\infty(1-a)]^2}}$$

and

$$f(\mu) = \frac{2}{\pi} \cos^{-1} \left(e^{-\frac{B}{2} \left(\frac{1-\mu}{\mu} \right) \sqrt{1 + \frac{(\lambda\mu)^2}{(1-a)^2}}} \right) \quad (3.78)$$

The Prandtl tip-loss factor for a three-blade rotor operating at a tip speed ratio of six is compared with the tip-loss factor of the helical vortex wake in Figure 3.33.

It should be pointed out that the vortex theory of Figure 3.28 also predicts that the tip-loss factor should be applied to the tangential flow induction factor.

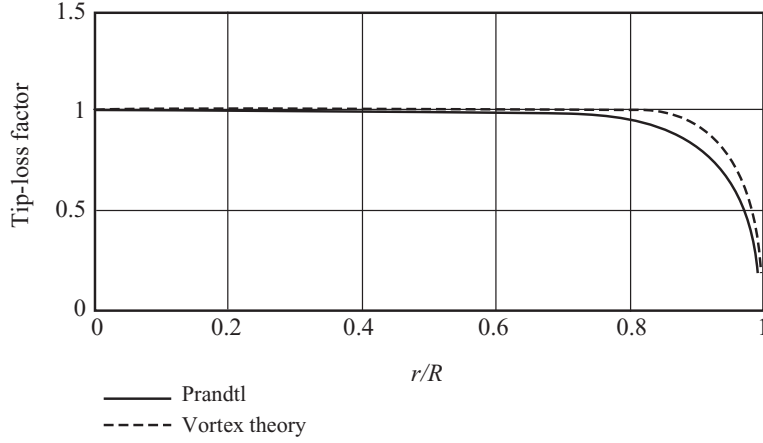


Figure 3.33 Comparison of the Prandtl tip-loss factor with that predicted by a vortex theory for a three blade turbine optimised for a tip speed ratio of six

It is now useful to know what is the variation of circulation along the blade. For the previous analysis, which disregarded tip-losses, the blade circulation was uniform (Equation 3.65).

Following the same procedure from which Equation 3.66 was developed:

$$\rho(W \times \Gamma) \sin \phi = \rho \Gamma U_{\infty}(1 - a_b(r)) = 4\pi \frac{U_{\infty}^3}{\Omega} a(r)(1 - a(r))^2$$

Recall that $a_b(r)$ is the flow factor local to the blade at radius r , which is equal to a , and $a(r)$ is the average value of the flow factor at radius r .

Therefore,

$$\Gamma(r) = \frac{4\pi}{\lambda(1 - a)} af(r)(1 - af(r))^2 \quad (3.79)$$

$\Gamma(r)$ is the total circulation for all blades and is shown in Figure 3.34; and, as can be seen, it is almost uniform except near to the tip. The dashed vertical line shows the effective blade length (radius) $R_{ef} = 0.975$ if the circulation is assumed to be uniform at the level that pertains at the inboard section of the blade.

The Prandtl tip-loss factor appears to offer an acceptable, simple solution to a complex problem; not only does it account for the effects of discrete blades but it also allows the induction factors to fall to zero at the edge of the rotor disc.

3.8.4 Blade root losses

At the root of a blade the circulation must fall to zero as it does at the blade tip, and so it can be presumed that a similar process occurs. The blade root will be at some distance from the rotor axis and the airflow through the disc inside the blade root radius will be at the free-stream velocity. Actually, the vortex theory of Section 3.4 can be extended to show that the flow

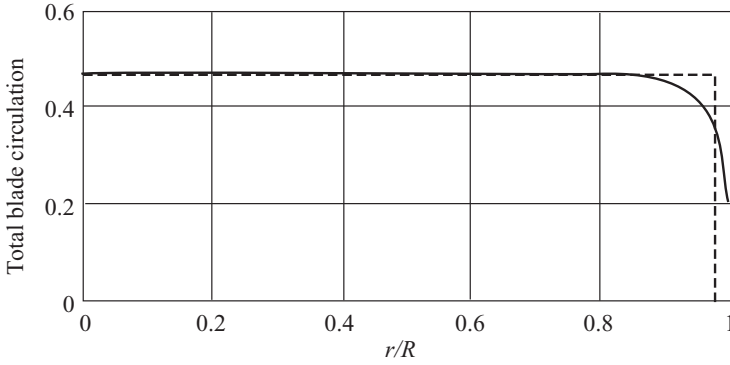


Figure 3.34 Span-wise variation of blade circulation for a three-blade turbine optimised for a tip speed ratio of six

through the root disc is somewhat higher than the free-stream velocity. It is usual, therefore, to apply the Prandtl tip loss function at the blade root as well as at the tip (see Figure 3.35).

If μ_R is the normalised root radius then the root loss factor can be determined by modifying the tip loss factor of Equation 3.78.

$$f_R(\mu) = \frac{2}{\pi} \cos^{-1} \left(e^{-\frac{B}{2} \left(\frac{\mu - \mu_R}{\mu} \right)} \sqrt{1 + \frac{(\lambda\mu)^2}{(1-a)^2}} \right) \quad (3.80)$$

If Equation 3.78 is now termed $f_T(r)$ the complete tip/root loss factor is

$$f(\mu) = f_T(\mu) f_R(\mu) \quad (3.81)$$

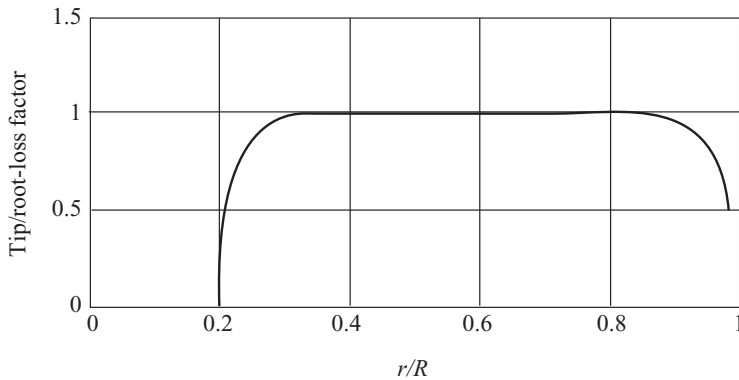


Figure 3.35 Span-wise variation of combined tip/root loss factor for a three-blade turbine optimised for a tip speed ratio of six and with a blade root at 20% span

3.8.5 Effect of tip-loss on optimum blade design and power

With no tip-loss the optimum axial flow induction factor is uniformly $\frac{1}{3}$ over the whole swept rotor. The presence of tip-loss changes the optimum value of the average value of a which reduces to zero at the edge of the wake, but, local to the blade tends to increase in the tip region.

If $a(r)$ is taken as the azimuthal average at radius r then locally, at the blade at that radius, the flow factor will be $a(r)/f(r)$. The inflow angle ϕ , at the blade is then, from Equation 3.61,

$$\tan \phi = \frac{1}{\lambda\mu} \left(\frac{1 - \frac{a}{f}}{1 + \frac{a'}{f}} \right) \quad (3.82)$$

but Equation 3.60, which is the ratio of the non-dimensional rate of change of angular momentum to the non-dimensional rate of change of axial momentum, is not changed because it deals with the whole flow through the disc and so uses average values. If drag is ignored for the present, Equation 3.61 becomes

$$\tan \phi = \frac{\lambda\mu a'(1-a)}{a(1-a) + (a'\lambda\mu)^2} \quad (3.83)$$

Hence,

$$\frac{(1-a)\lambda\mu a'}{a(1-a) + (a'\lambda\mu)^2} = \frac{\left(1 - \frac{a}{f}\right)}{\lambda\mu \left(1 + \frac{a'}{f}\right)}$$

which becomes

$$\lambda^2\mu^2 \frac{(f-1)}{f} a'^2 - \lambda^2\mu^2(1-a)a' + a(1-a) \left(1 - \frac{a}{f}\right) = 0 \quad (3.84)$$

A great simplification can be made to Equation 3.84 by ignoring the first term because, clearly, it disappears for much of the blade, where $f = 1$, and for the tip region the value of a'^2 is very small. For tip speed ratios greater than three neglecting the first term makes negligible difference to the result.

$$\lambda^2\mu^2 a' = a \left(1 - \frac{a}{f}\right) \quad (3.85)$$

As before, Equation 3.59 still applies

$$\frac{da}{da'} = \frac{1-a}{a'}$$

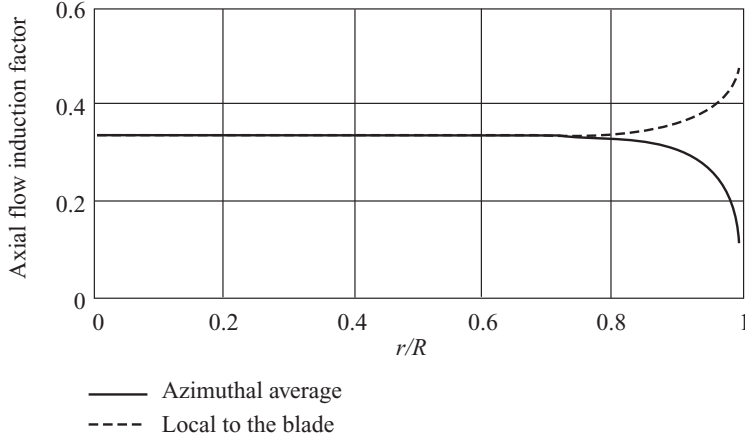


Figure 3.36 Axial flow factor variation with radius for a three blade turbine optimised for a tip speed ratio of six

From 3.85

$$\frac{da'}{da} = \frac{1}{\lambda^2 \mu^2} \left(1 - 2 \frac{a}{f} \right)$$

Consequently,

$$(1 - a) \left(1 - 2 \frac{a}{f} \right) = \lambda^2 \mu^2 a'$$

Which, combined with Equation 3.85, gives

$$a^2 - \frac{2}{3}(f + 1)a + \frac{1}{3}f = 0$$

so

$$a = \frac{1}{3} + \frac{1}{3}f - \frac{1}{3}\sqrt{1 - f + f^2} \quad (3.86)$$

The radial variation of the average value of a , as given by Equation 3.86, and the value local to the blade a/f , is shown in Figure 3.36. An exact solution would also have the local induced velocity falling to zero at the blade tip.

Clearly, the required blade design for optimal operation would be a little different to that which corresponds to the Prandtl tip-loss factor because a/f , the local flow factor does not fall to zero at the blade tip. The use of the Prandtl tip-loss factor leads to an approximation, but that was recognised from the outset.

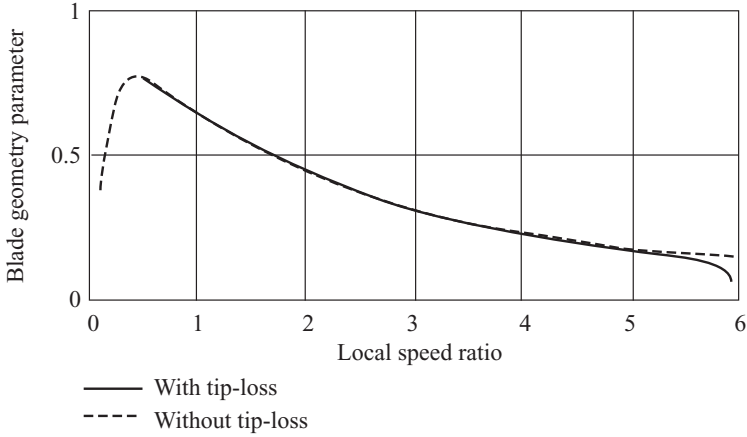


Figure 3.37 Variation of blade geometry parameter with local speed ratio, with and without tip-loss for a three-blade rotor with a design tip speed ratio of six

The blade design, which gives optimum power output, can now be determined by adapting Equations 3.68 and 3.69 accordingly:

$$\sigma_r \lambda C_l = \frac{4\lambda^2 \mu^2 a'}{\sqrt{\left(1 - \frac{a}{f}\right)^2 + \left[\lambda \mu \left(1 + \frac{a'}{f}\right)\right]^2}} \left(\frac{1 - a}{1 - \frac{a}{f}} \right)$$

Introducing Equation 3.85 gives

$$\sigma_r \lambda C_l = \frac{4a(1 - a)}{\sqrt{\left(1 - \frac{a}{f}\right)^2 + \left[\lambda \mu \left(1 + \frac{a \left(1 - \frac{a}{f}\right)}{\lambda^2 \mu^2 f}\right)\right]^2}} \quad (3.87)$$

The blade geometry parameter given by Equation 3.85 is shown in Figure 3.37 compared with the design which excludes tip-loss. As can be seen, only in the tip region is there any difference between the two designs.

Similarly, the inflow angle distribution, shown in Figure 3.38, can be determined by suitably modifying Equation 3.68

$$\tan \phi = \frac{1 - \frac{a}{f}}{\lambda \mu \left(1 + \frac{a \left(1 - \frac{a}{f}\right)}{\lambda^2 \mu^2 f}\right)} \quad (3.88)$$

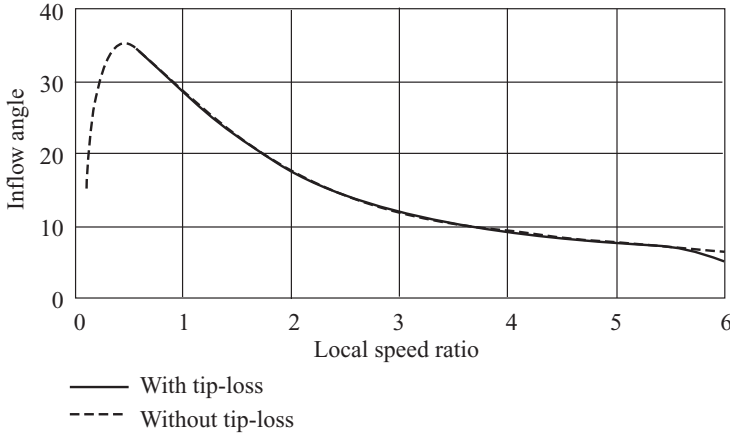


Figure 3.38 Variation of inflow angle with local speed ratio, with and without tip-loss for a three-blade rotor with a design tip speed ratio of six

Again, the effects of tip-loss are confined to the blade tip.

The power coefficient for an optimised rotor, operating at the design tip speed ratio, without drag and tip-losses, is equal to the Lanchester-Betz limit 0.593; but with tip-loss there is obviously a reduced optimum power coefficient. Equation 3.20 determines the power coefficient, see Figure 3.39.

$$C_P = \frac{P}{\frac{1}{2}\rho U_\infty^3 \pi R^2} = 8\lambda^2 \int_0^1 a'(1-a)\mu^3 d\mu \quad (3.89)$$

for which a' and a are obtained from Equations 3.85 and 3.86.

The maximum power coefficient that can be achieved in the presence of both drag and tip-loss is significantly less than the Lanchester-Betz limit at all tip speed ratios. As is shown in Figure 3.40, drag reduces the power coefficient at high tip speed ratios but the effect of tip-loss is most significant at low tip speed ratios because the separation of the helicoidal vortex sheets is large.

3.8.6 Incorporation of tip-loss for non-optimal operation

The blade element-momentum Equations 3.52, 3.52a and 3.53 are used to determine the flow induction factors for non-optimal operation. With tip-loss included, the BEM equations have to be modified. The necessary modification depends upon whether the azimuthally averaged values of the flow factors are to be determined or the maximum (local to a blade element) values. If the former alternative is chosen then, in the momentum terms the flow factors remain unmodified but in the blade element terms the flow factors must appear as the average values divided by the tip-loss factor. Choosing to determine the maximum values of the flow factors means that they are not modified in the blade element terms but are multiplied by the

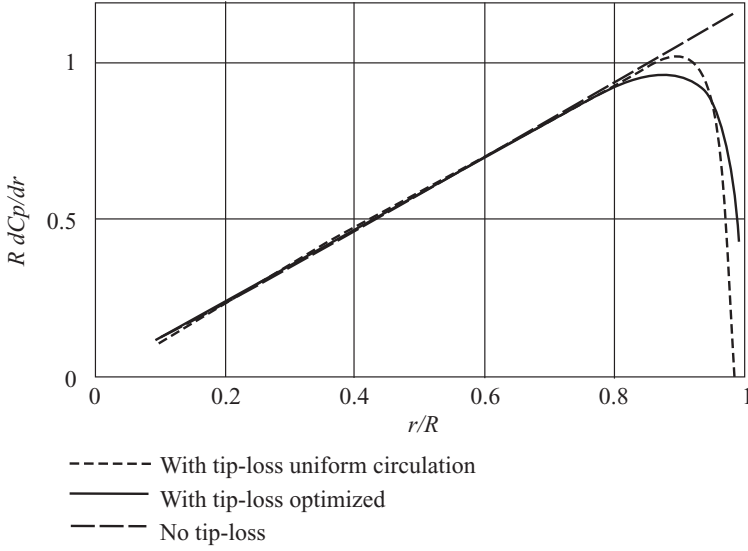


Figure 3.39 Span-wise variation of power extraction in the presence of tip-loss for three blades with uniform circulation and of optimised design for a tip speed ratio of six

tip-loss factor in the momentum terms. The latter choice allows the simplest modification of Equations 3.54 and 3.55.

$$\frac{af}{1-a} = \frac{\sigma_r}{4 \sin^2 \phi} C_x \frac{1-a}{1-af} \quad (3.54b)$$

$$\frac{a'f}{1+a'} = \frac{\sigma_r}{4 \sin \phi \cos \phi} C_y \frac{1-a}{1-af} \quad (3.55a)$$

There remains the problem of the breakdown of the momentum theory when wake mixing occurs. The helicoidal vortex sheets may not exist and so Prandtl's approximation is not physically appropriate. Nevertheless, particles which pass between blades will no doubt still lose less momentum than those which interact with a blade and so the application of a tip-loss factor is necessary. Prandtl's approximation is the only practical method available and so is commonly used. In view of the manner in which the experimental results of Figure 3.16 were gathered, it is the average value of a which should determine at which stage the momentum theory breaks down.

3.8.7 Alternative explanation for tip-loss

The flow approaching the rotor is expanding because it is slowing down and so is not axial, that is, it is not parallel to the rotation axis, or the undisturbed flow direction. Consequently, there is a radial flow velocity component at the upwind side of the rotor that arises because there is a radial pressure gradient with lower pressure (atmospheric) in the tip region than in the inner region. The change of radial momentum at a point on the rotor disc is balanced by the equal and opposite radial momentum at the diametrically opposite point. The magnitude

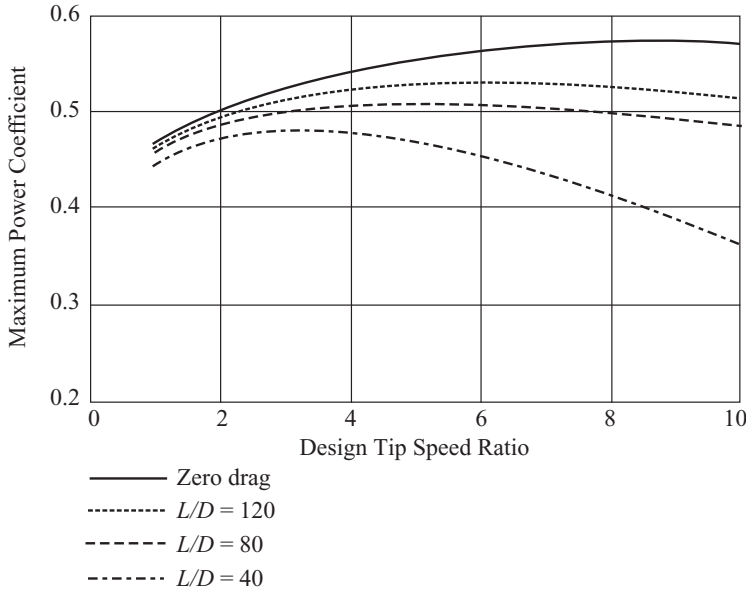


Figure 3.40 The variation of maximum C_p with design λ for various lift/drag ratios and including tip-losses for a three-bladed rotor

of the radial velocity increases with radius and so its effects will be greatest at the tip region. The kinetic energy associated with the radial flow is not available for energy capture because it does not influence the aerodynamic force on the blade. The axial component of velocity in the approaching flow is, therefore, depleted but not by an axial force.

For a well designed blade the tip chord length will be zero and so would not exert any axial force at all on the airflow, which means that the static pressure will be atmospheric and the resultant approach velocity will be equal in magnitude to the undisturbed wind speed. But, at the tip, there will be a relatively large radial flow velocity component causing a significant reduction in the axial flow component that is not associated with a corresponding axial force caused by a rotor blade. If, for example, the axial velocity component is $\frac{2}{3}$ of the undisturbed wind speed then the angle of inclination of the wind velocity at the tip will be

$$\cos^{-1}\left(\frac{2}{3}\right) = 48.19 \text{ degrees}$$

that is, $a = \frac{1}{3}$ but there is no associated axial force.

If a is redefined as a reduction in the stream-wise flow velocity, as shown in Figure 3.41, then at the blade tip a would be zero but would steadily rise towards the central section of the blade, reach a maximum and then fall to zero once more at the blade root.

Applying the same arguments as for the simple momentum theory of Section 3.2.1, Equation 3.7 can be modified thus,

$$\frac{1}{2}\rho(U_\infty^2 - U_w^2)A_D = (U_\infty - U_w)\rho A_D U_\infty(1 - a)\cos\psi \quad (3.7a)$$

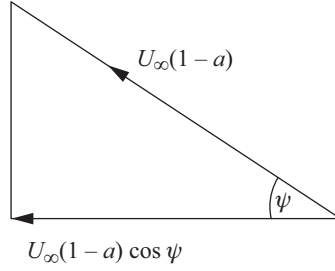


Figure 3.41 The flow geometry at the rotor disc with flow expansion

and the axial velocity in the wake becomes

$$U_W = U_\infty(1 - 2a \cos \psi) \quad (3.8a)$$

where ψ is the angle of the resultant stream-wise flow direction to the axial direction at the rotor disc.

The retarding force on the air flowing through an elemental area of the disc δA_D , from Equation 3.9, is

$$\delta T = (p_D^+ - p_D^-) \delta A_D = 2\rho \delta A_D U_\infty^2 a(1-a) \cos^2 \psi \quad (3.9a)$$

and the power extracted from that air is

$$\delta Power = \delta T U_D = 2\rho A_D U_\infty^3 a(1-a)^2 \cos^3 \psi \quad (3.10a)$$

Both a and ψ will vary radially and will change according to how the circulation on the disc varies radially. Disc circulation, or the bound vorticity on the disc, must also rise and fall from blade root to blade tip, as shown in Figure 3.42.

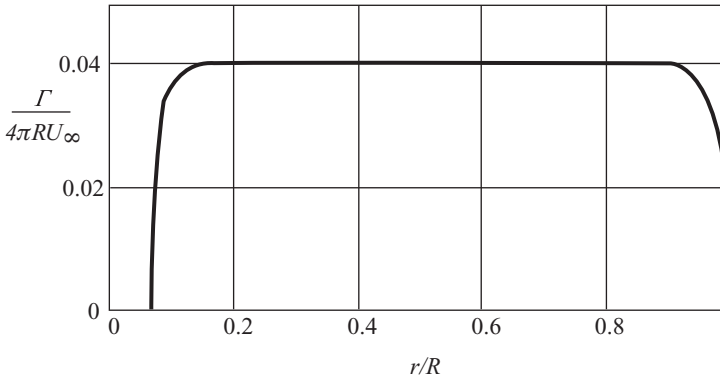


Figure 3.42 The variation of circulation along the length of a blade

Using just the momentum theory it is not possible to determine the manner of the variation of a and ψ but it is clear that the integration with respect to radius r of Equation 3.10a would result in a value for the optimised power coefficient that would be less than the Lanchester-Betz limit.

The process of flow expansion used here takes place entirely in the region of flow that is upstream of the rotor disc and shows that there is a consequent loss of efficiency in the tip and root regions of a blade. The tip/root-loss mechanism described by Prandtl, on the other hand, takes place entirely in the flow region downstream of the rotor disc so the two loss processes are distinct.

3.9 Stall delay

A phenomenon first noticed on propellers by Himmelskamp (1945) is that of lift coefficients being attained at the inboard section of a rotating blade which are significantly in excess of the maximum value possible in two-dimensional static tests. In other words the angle of attack at which stall occurs is greater for a rotating blade than for the same blade tested statically. The power output of a rotor is measurably increased by the stall delay phenomenon and, if included, improves the comparison of theoretical prediction with measured output. It is noticed that the effect is greater near the blade root and decreases with radius.

The reason for the stall delay has been the cause of much discussion, but a convincing physical process has not yet been established. What is agreed is that, for whatever reason, the adverse pressure gradient experienced by the flow passing over the downwind surface of the blade is reduced by the blade's rotation. The adverse pressure gradient slows down the flow as it approaches the trailing edge of the blade after the velocity peak reached close to the leading edge. In the boundary layer viscosity also slows down the flow and the combination of the two effects, if sufficiently large, can bring the boundary layer flow to a standstill (relative to the blade surface) or even cause a reversal of flow direction. When flow reversal takes place the flow separates from the blade surface and stall occurs, giving rise to loss of lift and a dramatic increase in pressure drag.

Aerodynamic analyses (Wood, 1991; Snel *et al.*, 1993) of rotating blades using computational fluid dynamic techniques, which include the effects of viscosity, do show a decrease in the adverse pressure gradient but it is not obvious from these numerical calculations as to what exactly is occurring physically.

It is also agreed that the parameter that influences stall delay predominantly is the local blade solidity $c(r)/r$.

The evidence which does exist shows that for attached flow conditions, below what would otherwise be the static (non-rotating) stall angle of attack, there is little difference between two-dimensional flow conditions and rotating conditions. When stall does occur, however, the air in the separated region, which is moving very slowly with respect to the blade surface, is rotating with the blade and so is subject to centrifugal force causing it to flow radially outwards. Prior to stalling taking place, centrifugal forces on the fluid in the boundary layer, again causing radial flow, may reduce the displacement thickness and so increase the resistance to separation.

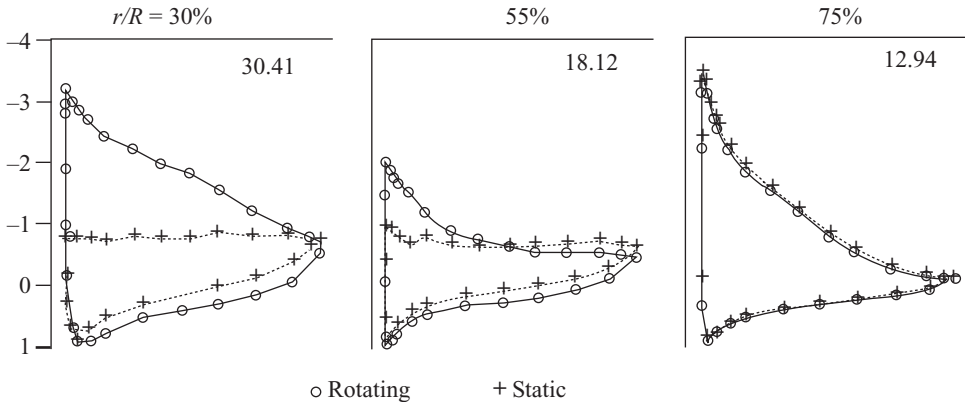


Figure 3.43 Pressure measurements on the surface of a wind turbine blade while rotating and while static by Ronsten (1991)

Blade surface pressures have been measured by Ronsten (1991) on a blade while static and while rotating. Figure 4.43 shows the comparison of surface pressure coefficients for similar angles of attack in the static and rotating conditions (tip speed ratio of 4.32) for three span-wise locations. At the 30% span location the estimated angle of attack at 30.41° is well above the static stall level which is demonstrated by the static pressure coefficient distribution. The rotating pressure coefficient distribution at 30% span shows a high leading edge suction pressure peak with a uniform pressure recovery slope over the rear section of the upper surface of the chord. The gradual slope of the pressure recovery indicates a reduced adverse pressure gradient with the effect on the boundary layer that it is less likely to separate. The level of the leading edge suction peak, however, is very much less than it would be if, in the non-rotating situation, it were possible for flow still to be attached at 30.41°.

The situation at the 55% span-wise location is similar to that at 30%; the static pressures indicate that the section has stalled but the rotating pressures show leading edge suction peak which is small but significant. At the 75% span location there is almost no difference between static and rotating blade pressure coefficient distributions at an angle of attack of 12.94°, which is below the static stall level: the leading edge suction pressure peak is little higher than that at 30% span, much higher than that at 55%, but the pressure recovery slope is much steeper. The flow appears to be attached at the 30% and 55% span locations on the rotating blade, but the suction pressures are too low for that actually to be the case, so stall appears to be greatly delayed and the low adverse pressure gradient shown by the gentle slope of the pressure recovery appears to indicate the reason for the delay. At 30% span the ratio $c/r = 0.374$, $c/r = 0.161$ at 55% span and at the 75% location $c/r = 0.093$. The increased lift also occurs in the post stall region and is attributed to the radial flow in the separated flow regions.

Snel *et al.* (1993) have proposed a simple, empirical modification to the usually available two dimensional, static aerofoil lift coefficient data which fits the measured lift coefficients by Ronsten (1991) and computed results using a three-dimensional computational fluid dynamics code.

Table 3.1 Summary of Ronsten’s measurements of lift coefficient and lift coefficients corrected to rotating conditions using Equation 3.90

$r/R \cdot 100$	30%	55%	75%
c/r	0.374	0.161	0.093
Angle of attack Δ	30.41°	18.12°	12.94°
C_l static (measured)	0.8	0.74	1.3
C_l rotating (measured)	1.83	0.93	1.3
C_l rotating (Snel)	1.87	0.84	1.3

If the linear part of the static, 2-D, $C_l - \alpha$ curve is extended beyond the stall then let ΔC_l be the difference between the two curves. Then the correction to the 2-D curve to account for the rotational, 3-D, effects is $3 (c/r)^2 \Delta C_l$.

$$C_{l,3D} = C_{l,2D} + 3 \left(\frac{c}{r}\right)^2 \Delta C_l \tag{3.90}$$

Table 3.1 compares the measured static ($C_{l,2D}$) ($C_{l_{2D}}$) and rotating ($C_{l,3D}$) lift coefficients with the calculated values for the rotating values using Snel’s correction of Equation 3.90. The correction is quite good and is very simple to apply. An example of the correction is given by Snel in (1993) and is shown in Figure 3.44.

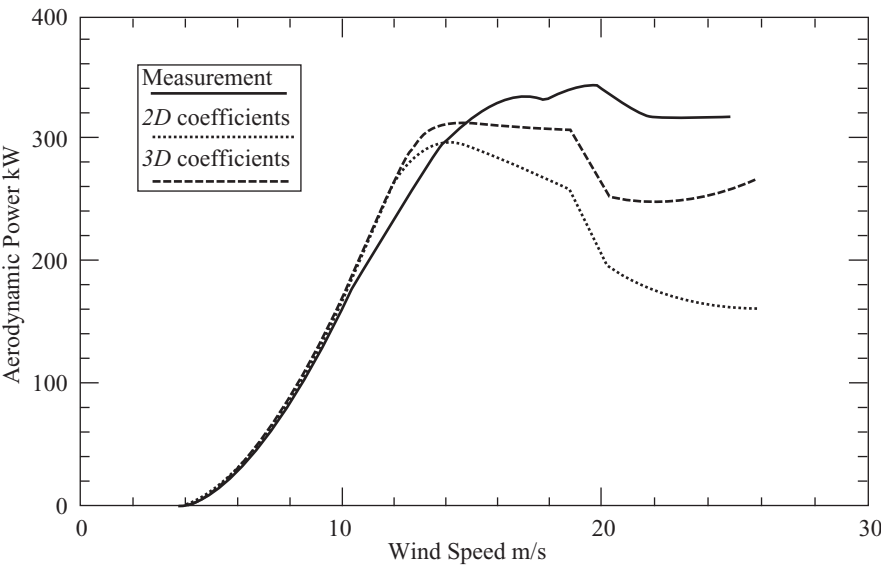


Figure 3.44 A comparison of measured and Snel’s predicted power curves for a NORDTANK 300 kW turbine

Table 3.2 Blade design of a 17 m-diameter rotor

Radius r mm	$\mu = \frac{r}{R}$	Chord c mm	Pitch β deg	Thickness/Chord ratio of blade %
1700	0.20	1085	15.0	24.6
2125	0.25	1045	12.1	22.5
2150	0.30	1005	9.5	20.7
2975	0.35	965	7.6	19.5
3400	0.40	925	6.1	18.7
3825	0.45	885	4.9	18.1
4250	0.50	845	3.9	17.6
4675	0.55	805	3.1	17.1
5100	0.60	765	2.4	16.6
5525	0.65	725	1.9	16.1
5950	0.70	685	1.5	15.6
6375	0.75	645	1.2	15.1
6800	0.80	605	0.9	14.6
6375	0.85	565	0.6	14.1
7225	0.90	525	0.4	13.6
8075	0.95	485	0.2	13.1
8500	1.00	445	0.0	12.6

3.10 Calculated results for an actual turbine

The blade design of a turbine operating at constant uniform rotational speed and fixed pitch is given in Table 3.2 below and the aerofoil characteristics are shown in Figure 3.45.

Using the above data the results shown in Figure 3.46 were obtained.

The blade is designed for optimum performance at a tip speed ratio of about six and, ideally, the angle of attack uniform along the span at the level for which the lift/drag ratio is a maximum, about 7° for the aerofoil concerned. At the lowest tip speed ratio shown in Figure 3.44 the entire blade is stalled and for a rotational speed of 60 rpm the corresponding wind speed will be 26 m/s, which is the cut-out speed. For the highest tip speed ratio shown the corresponding wind speed will be 4.5 m/s, the cut-in speed. Maximum power is developed at a tip speed ratio of four at a wind speed of 13 m/s and, clearly, much of the blade is stalled.

The axial flow induction factor is not uniform along the span at any tip speed ratio, indicating that the blade design is an engineering compromise, but at the tip speed ratio of six there is value a little higher than $\frac{1}{3}$. The flow factors shown in Figure 3.47 are those local to the blade and so average value of axial flow factor will be close to $\frac{1}{3}$ at a tip speed ratio of six.

Generally, the axial flow factor increases with tip speed ratio while the tangential flow factor decreases with tip speed ratio. The angular velocity of the wake increases sharply with decreasing radius because it is determined by the root vortex.

The importance of the outboard section of the blade is clearly demonstrated in Figure 3.48. The dramatic effect of stall is shown in the difference in torque distribution between the

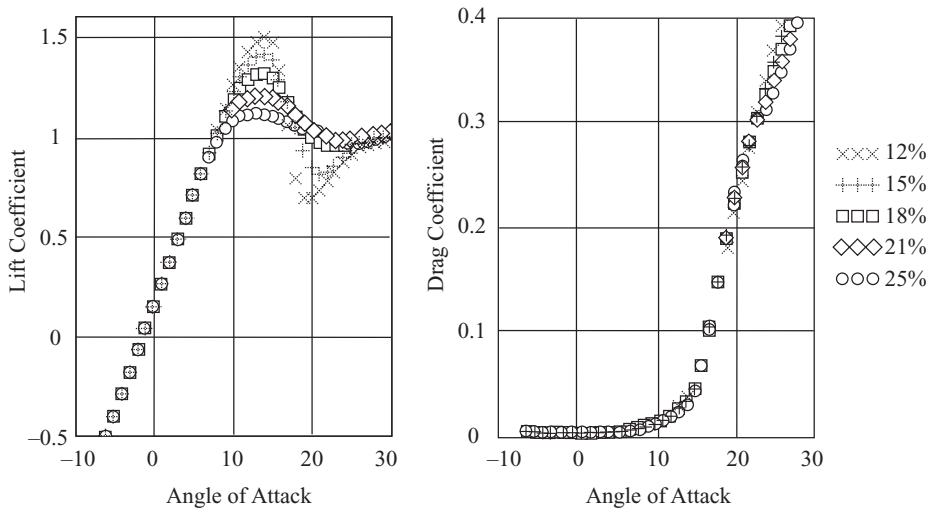


Figure 3.45 The aerodynamic characteristics of the NACA632XX aerofoil series The complete $C_P - \lambda$ curve for the design is given in Figure 3.15.

tip speed ratio of four and the tip speed ratio of two. Note, also, the flat distribution of torque at the high tip speed ratio of 12; this is caused by the effect of drag which reduces torque as the square of the local speed ratio and with the low angle of attack at $\lambda = 12$ drag causes a significant loss of power.

Although the blade axial force *coefficient* increases with tip speed ratio it must be remembered that the actual thrust force increases with wind speed as is demonstrated in Figure 3.49.

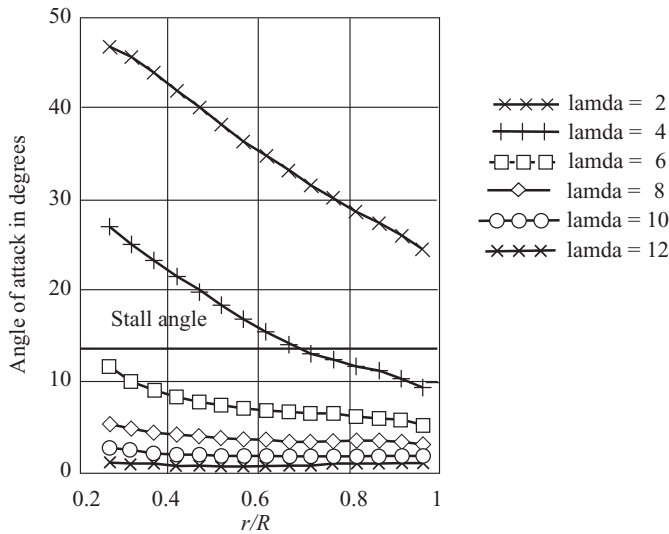


Figure 3.46 Angle of attack distribution for a range of tip speed ratios

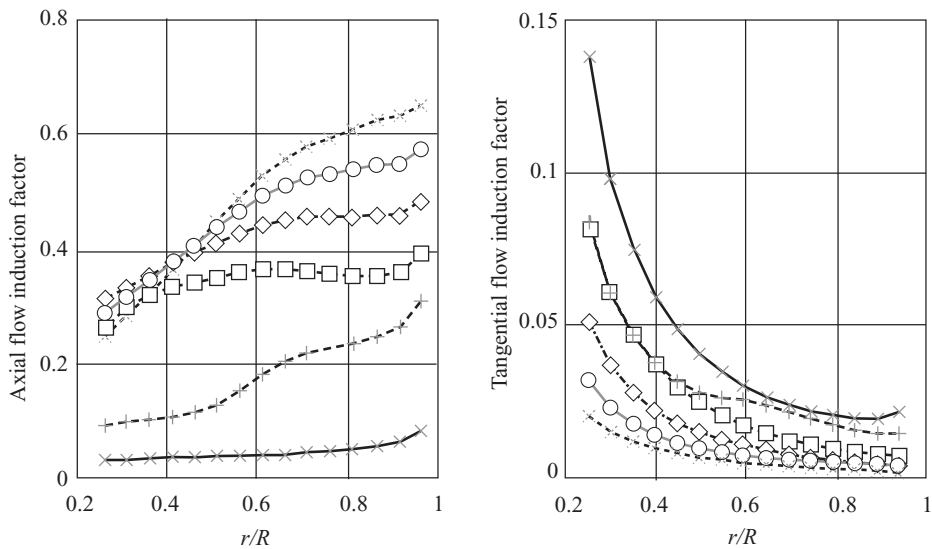


Figure 3.47 Distribution of the flow induction factors for a range of tip speed ratios

3.11 The performance curves

3.11.1 Introduction

The performance of a wind turbine can be characterised by the manner in which the three main indicators, power, torque and thrust, vary with wind speed. The power determines the

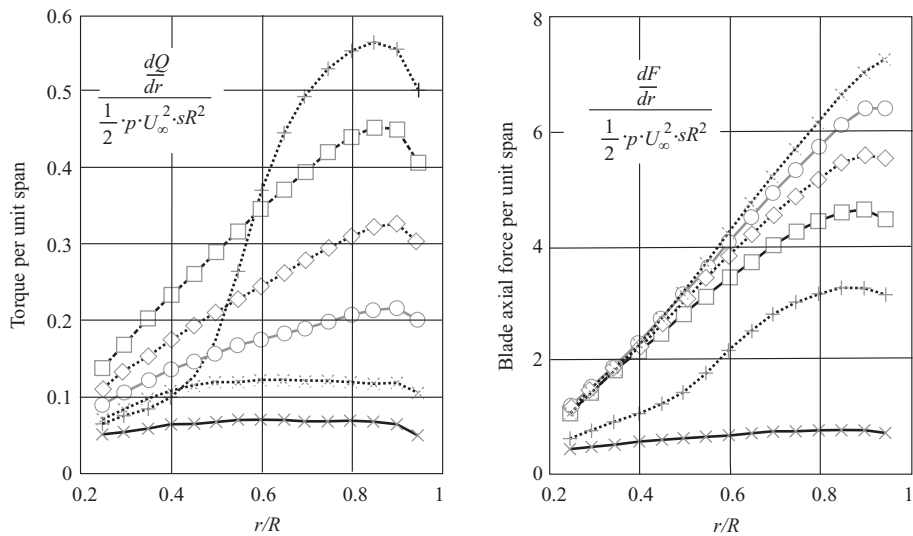


Figure 3.48 Distribution of blade loads for a range of tip speed ratios

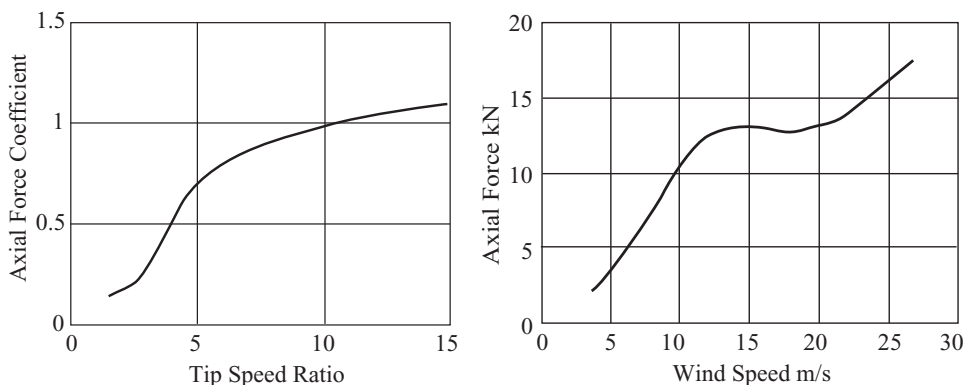


Figure 3.49 Axial force coefficient and the variation of the actual force with wind speed

amount of energy captured by the rotor, the torque developed determines the size of the gear box and must be matched by whatever generator is being driven by the rotor. The rotor thrust has great influence on the structural design of the tower. It is usually convenient to express the performance by means of non-dimensional, characteristic performance curves from which the actual performance can be determined regardless of how the turbine is operated, for example, at constant rotational speed or some regime of variable rotor speed. Assuming that the aerodynamic performance of the rotor blades does not deteriorate the non-dimensional aerodynamic performance of the rotor will depend upon the tip speed ratio and, if appropriate, the pitch setting of the blades. It is usual, therefore, to display the power, torque and thrust coefficients as functions of tip speed ratio.

3.11.2 The $C_P - \lambda$ performance curve

The theory described in this chapter gives the wind turbine designer a means of examining how the power developed by a turbine is governed by the various design parameters. The usual method of presenting power performance is the non-dimensional $C_P - \lambda$ curve and the curve for a typical, modern, three-blade turbine is shown in Figure 3.50.

The first point to notice is that the maximum value of C_P is only 0.47, achieved at a tip speed ratio of seven, which is much less than the Betz limit for that tip speed ratio. The discrepancy is caused, in this case, by drag and tip-losses but the stall also reduces the C_P at low values of the tip speed ratio.

Even with no losses included in the analysis the Lanchester-Betz limit is not reached because the blade design is not perfect (see Figure 3.51).

3.11.3 The effect of solidity on performance

At this stage, the other principal parameter to consider is the solidity, defined as total blade area divided by the swept area. For the three-blade machine, above, the solidity is 0.0345 but this can be altered readily by varying the number of blades as shown in Figure 3.52.

The solidity could also have been changed by changing the blade chord.

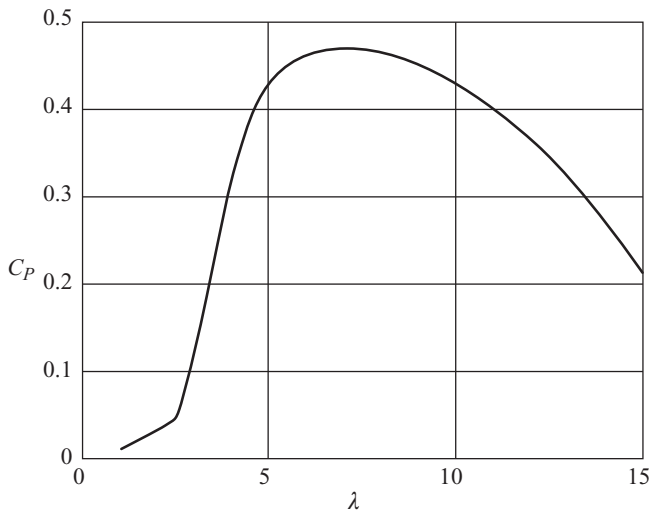


Figure 3.50 $C_P - \lambda$ performance curve for a modern three-blade turbine

The main effects to observe of changing solidity are:

- 1. Low solidity produces a broad, flat curve which means that the C_P will change very little over a wide tip speed ratio range, but the maximum C_P is low because the drag losses are high (drag losses are roughly proportional to the cube of the tip speed ratio).
- 2. High solidity produces a narrow performance curve with a sharp peak making the turbine very sensitive to tip speed ratio changes and, if the solidity is too high, has a relatively low maximum C_P . The reduction in C_P max is caused by stall losses.

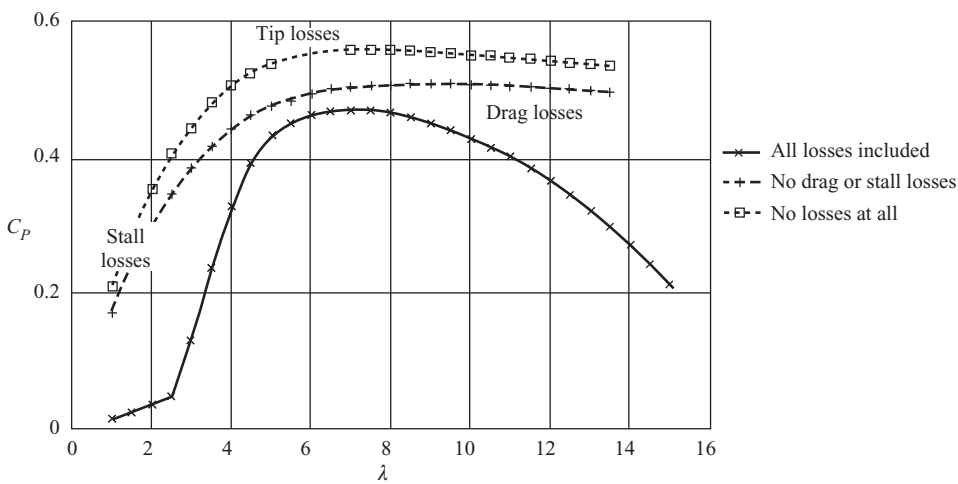


Figure 3.51 $C_P - \lambda$ performance curve for a modern three-blade turbine showing losses

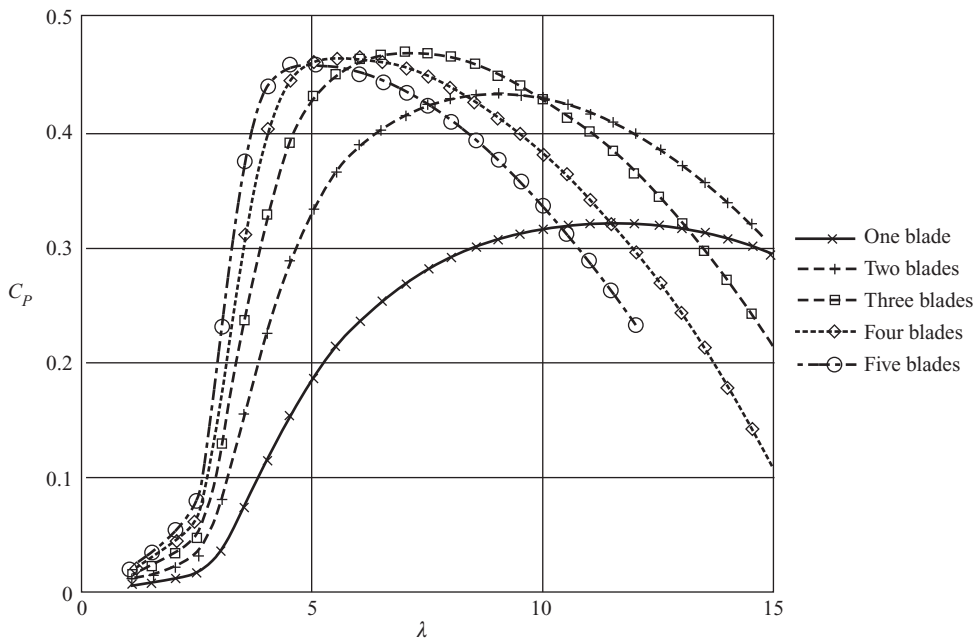


Figure 3.52 Effect of changing solidity

3. An optimum solidity appears to be achieved with three blades, but two blades might be an acceptable alternative because although the maximum C_P is a little lower the spread of the peak is wider and that might result in a larger energy capture.

It might be argued that a good solution would be to have a large number of blades of small individual solidity but this greatly increases production costs and results in blades which are structurally weak and very flexible.

There are applications which require turbines of relatively high solidity, one is the directly driven water pump and the other is the very small turbine used for battery charging. In both cases it is the high starting torque (high torque at very low tip speed ratios) which is of importance and this also allows small amounts of power to be developed at very low wind speeds, ideal for trickle charging batteries.

3.11.4 The $C_Q - \lambda$ curve

The torque coefficient is derived from the power coefficient simply by dividing by the tip speed ratio and so it does not give any additional information about the turbine’s performance. The principal use of the $C_Q - \lambda$ curve is for torque assessment purposes when the rotor is connected to a gear box and generator.

Figure 3.53 shows how the torque developed by a turbine rises with increasing solidity. For modern high speed turbines designed for electricity generation, as low a torque as possible is desirable in order to reduce gearbox costs. On the other hand the multi-bladed, high solidity turbine, developed in the nineteenth century for water pumping, rotates slowly and has a very high starting torque coefficient necessary for overcoming the torque required to start a positive displacement pump.

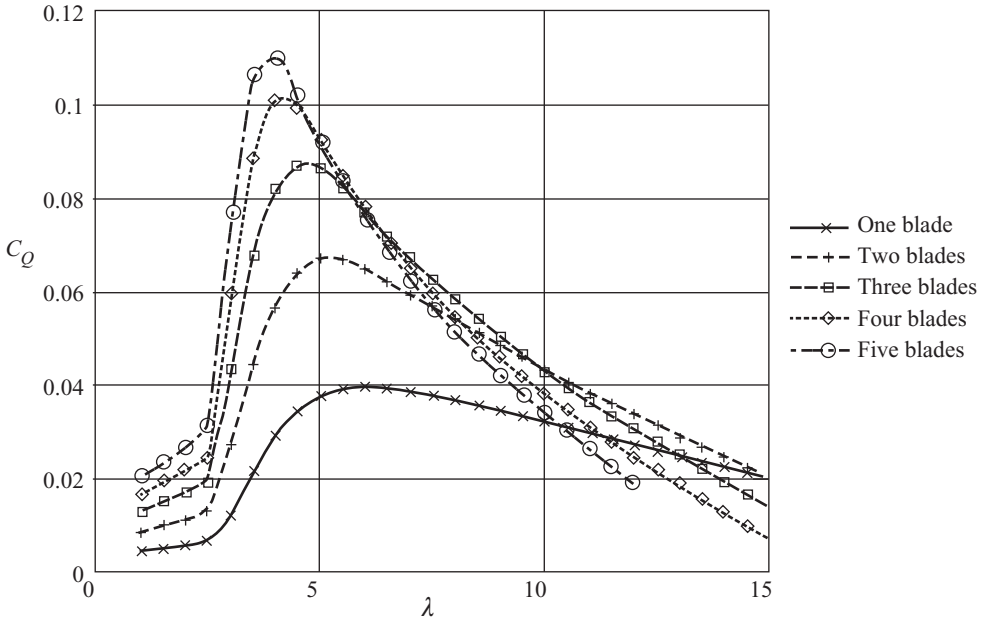


Figure 3.53 The effect of solidity on torque

The peak of the torque curve is caused by stall and occurs at a lower tip speed ratio than the peak of the power curve.

3.11.5 The $C_T - \lambda$ curve

The thrust force on the rotor is directly applied to the tower on which the rotor is supported and so considerably influences the structural design of the tower.

Generally, the thrust on the rotor increases with increasing solidity, as shown in Figure 3.54.

3.12 Constant rotational speed operation

3.12.1 Introduction

The majority of wind turbines currently installed generate electricity. Whether or not these turbines are grid connected they need to produce an electricity supply which is of constant frequency, or else many common appliances will not function properly. Consequently, a mode favoured in the early years of wind turbine development is operation at constant rotational speed. Connected to the grid a constant speed turbine is automatically controlled, whereas a stand-alone machine needs to have speed control and a means of dumping excess power.

3.12.2 The $K_P - 1/\lambda$ curve

An alternative performance curve can be produced for a turbine controlled at constant speed. The $C_P - \lambda$ curve shows, non-dimensionally, how the power would vary with rotational speed

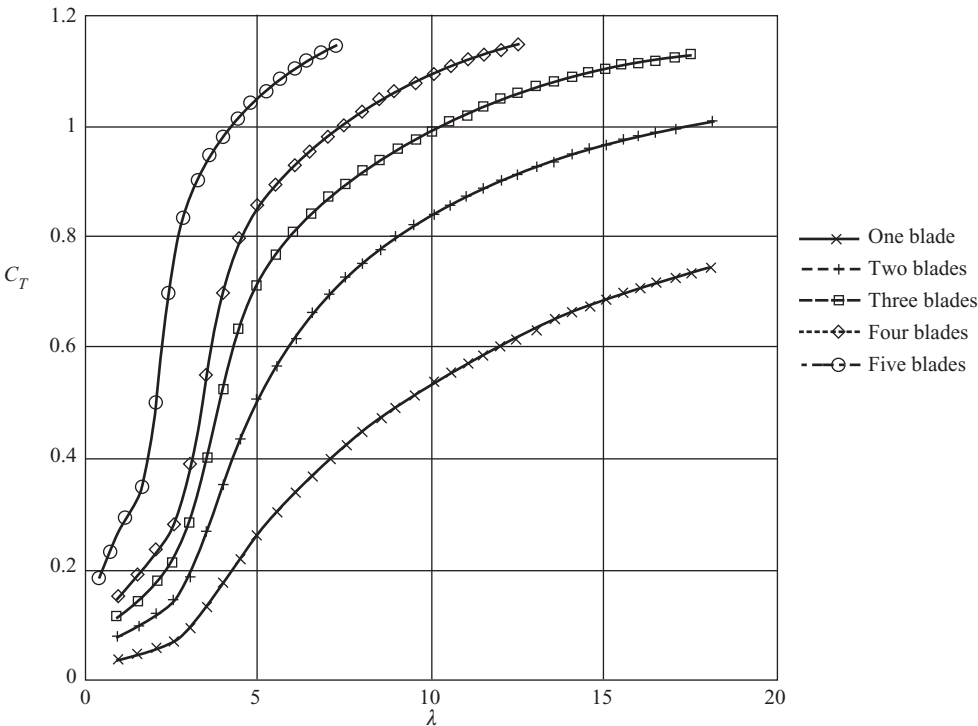


Figure 3.54 The effect of solidity on thrust

if the wind speed was held constant. The $K_P - 1/\lambda$ curve describes, again non-dimensionally, how the power would change with wind speed when constant rotational speed is enforced. K_P is defined as

$$K_P = \frac{\text{Power}}{\left(\frac{1}{2}\right) \cdot \rho \cdot (\Omega \cdot R)^3 A_d} = \frac{C_P}{\lambda^3} \tag{3.91}$$

The $C_P - \lambda$ and $K_P - 1/\lambda$ curves for a typical fixed pitch wind turbine are shown in Figure 3.55. The $K_P - 1/\lambda$ curve, as stated above, has the same form as the power–wind speed characteristic of the turbine. The efficiency of the turbine (given by the $C_P - \lambda$ curve) varies greatly with wind speed, a disadvantage of constant speed operation, but it should be designed such that the maximum efficiencies are achieved at wind speeds where there is the most energy available.

3.12.3 Stall regulation

An important feature of this $K_P - 1/\lambda$ curve is that the power, initially, falls off once stall has occurred and then gradually increases with wind speed. This feature provides an element of passive power output regulation, ensuring that the generator is not overloaded as the wind speed increases. Ideally, the power should rise with wind speed to the maximum value and

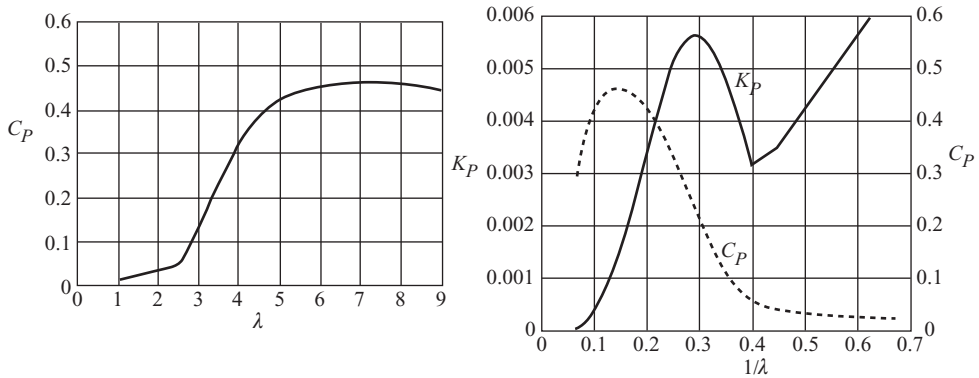


Figure 3.55 Non-dimensional performance curves for constant speed operation

then remain constant regardless of the increase in wind speed: this is called perfect stall regulation. However, stall regulated turbines do not exhibit the ideal, passive stall behaviour.

Stall regulation provides the simplest means of controlling the maximum power generated by a turbine to suit the sizes of the installed generator and gearbox. The principal advantage of stall control is simplicity but there are significant disadvantages. The power versus wind speed curve is fixed by the aerodynamic characteristics of the blades, in particular the stalling behaviour. The post stall power output of a turbine varies very unsteadily and in a manner which, so far, defies prediction, see Figure 3.60, for example. The stalled blade also exhibits low vibration damping because the flow about the blade is unattached to the low pressure surface and blade vibration velocity has little effect on the aerodynamic forces. The low damping can give rise to large vibration displacement amplitudes which will inevitably be accompanied by large bending moments and stresses, causing fatigue damage. When parked in high, turbulent winds the fixed pitch, stationary blade may well be subject to large aerodynamic loads which cannot be alleviated by adjusting (feathering) the blade pitch angle. Consequently, the blades of a fixed pitch, stall regulated turbine must be very strong, involving an appropriate cost penalty.

3.12.4 Effect of rotational speed change

The power output of a turbine running at constant speed is strongly governed by the chosen, operational rotational speed. If a low rotation speed is used the power reaches a maximum at a low wind speed and consequently it is very low. To extract energy at wind speeds higher than the stall peak the turbine must operate in a stalled condition and so is very inefficient. Conversely, a turbine operating at a high speed will extract a great deal of power at high wind speeds but at moderate wind speeds it will be operating inefficiently because of the high drag losses. Figure 3.56 demonstrates the sensitivity to rotation speed of the power output – a 33% increase in rpm from 45 to 60 results in a 150% increase in peak power, reflecting the increased wind speed at which peak power occurs at 60 rpm.

At low wind speeds, on the other hand, there is a marked fall in power with increasing rotational speed as shown in Figure 3.57. In fact, the higher power available at low wind speeds if a lower rotational speed is adopted has led to two speed turbines being built. Operating at

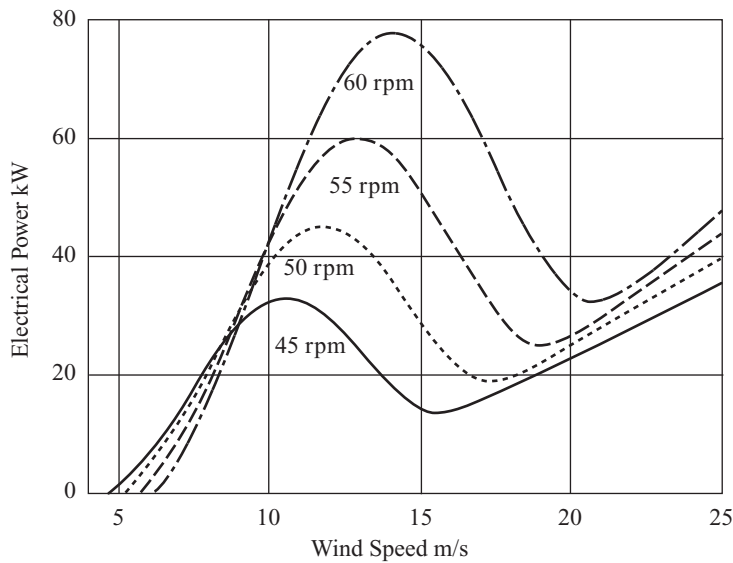


Figure 3.56 Effect on extracted power of rotational speed

one fixed speed which maximises energy capture at wind speeds at, or above, the average level will result in a rather high cut-in wind speed, the lowest wind speed at which generation is possible. Employing a lower rotational speed at low wind speeds reduces the cut-in wind speed and increases energy capture. The increased energy capture is, of course, offset by the cost of the extra machinery.

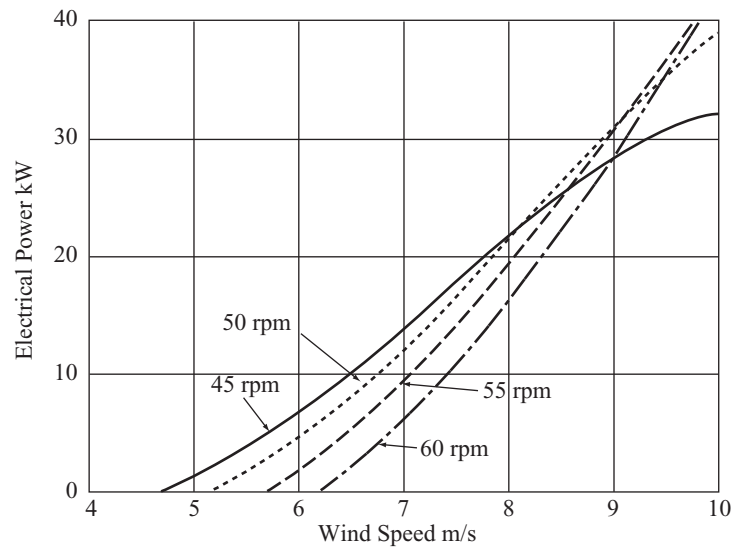


Figure 3.57 Effect on extracted power of rotational speed at low wind speeds

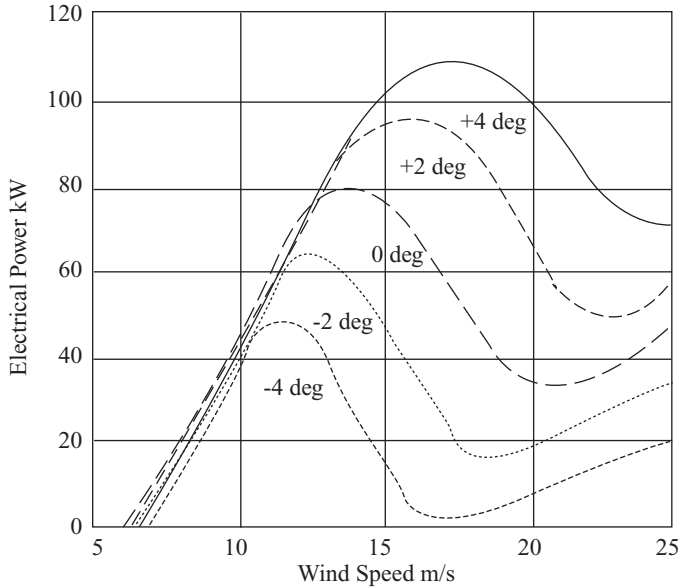


Figure 3.58 Effect on extracted power of blade pitch set angle

3.12.5 Effect of blade pitch angle change

Another parameter which affects the power output is the pitch setting angle of the blades β_s . Blade designs almost always involve twist but the blade can be set at the root with an overall pitch angle. The effects of a few degrees of pitch are shown in Figure 3.58.

Small changes in pitch setting angle can have a dramatic effect on the power output. Positive pitch angle settings increase the design pitch angle and so decrease the angle of incidence. Conversely, negative pitch angle settings increase the angle of incidence and may cause stalling to occur as shown in Figure 3.58. A turbine rotor designed to operate optimally at a given set of wind conditions can be suited to other conditions by appropriate adjustments of blade pitch angle and rotational speed.

3.13 Pitch regulation

3.13.1 Introduction

Many of the shortcomings of fixed pitch/passive stall regulation can be overcome by providing active pitch angle control. Figure 3.58 shows the sensitivity of power output to pitch angle changes.

The most important application of pitch control is for power regulation, but pitch control has other advantages. By adopting a large positive pitch angle a large starting torque can be generated as a rotor begins to turn. A 90° pitch angle is usually used when the rotor is stationary because this will minimise forces on the blades such that they will not sustain damage in high winds. At 90° of positive pitch the blade is said to be ‘feathered’. The blades

need not be as strong, therefore, as for a stall-regulated turbine, which reduces blade costs. Only a small change of pitch angle is needed to provide an assisted start-up.

The principle disadvantages of pitch control are lower reliability and cost but the latter is offset by lower blade costs.

Power regulation can be achieved either by pitching to promote stalling or pitching to feather, which reduces the lift force on the blades by reducing the angle of attack.

3.13.2 Pitching to stall

Figure 3.58 shows the power curves for a turbine rated at 60 kW, which is achieved at 12 m/s. At wind speeds below the rated level the blade pitch angle is kept at 0°. As rated power is reached only a small negative pitch angle, initially of about 2°, is necessary to promote stalling and so to limit the power to the rated level. As the wind speed increases small adjustments in both the positive and negative directions are all that are needed to maintain constant power.

The small size of the pitch angle adjustments make pitching to stall very attractive to designers, but the blades have the same damping and fatigue problems as fixed pitch turbines.

3.13.3 Pitching to feather

By increasing the pitch angle as rated power is reached the angle of attack can be reduced. A reduced angle of attack will reduce the lift force and the torque. The flow around the blade remains attached. Figure 3.59 is for the same turbine as Figure 3.58 but only the 0° power curve is shown below the rated level. Above the rated level fragments of power curves for higher pitch angles are shown as they cross the rated power line: the crossing points give the necessary pitch angles to maintain rated power at the corresponding wind speeds. As can be seen in Figure 3.57, the required pitch angles increase progressively with wind speed and are generally much larger than is needed for the pitching to stall method. In gusty conditions

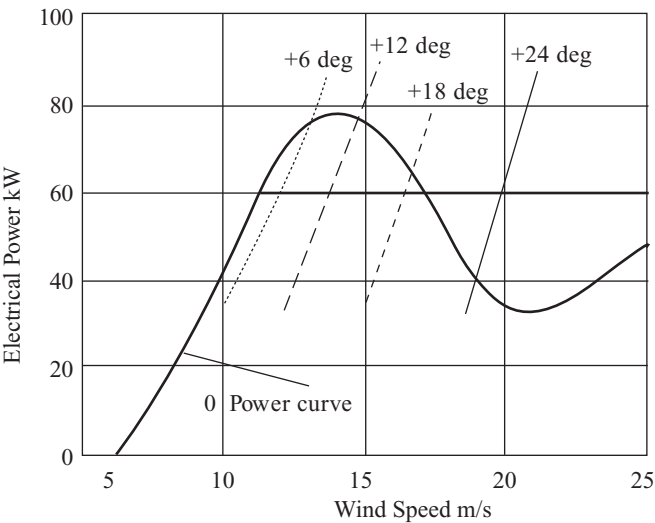


Figure 3.59 Pitching to feather power regulation requires large changes of pitch angle

large pitch excursions are needed to maintain constant power and the inertia of the blades will limit the speed of the control system's response.

Because the blades remain un-stalled if large gusts occur at wind speeds above the rated level large changes of angle of attack will take place with associated large changes in lift. Gust loads on the blades can, therefore, be more severe than for stalled blades.

The advantages of the pitching to feather method are that the flow around the blade remains attached, and so well understood, and provides good, positive damping. Feathered blade parking and assisted starting are also available.

Pitching to feather has been the preferred pitch control option mainly because the blade loads can be predicted with more confidence than for stalled blades.

3.14 Comparison of measured with theoretical performance

The turbine considered in this section is run at constant rotational speed. More detail about this method of operation will be discussed in the next section but the main feature is that there is, theoretically, a unique power output for a given wind speed.

When the turbine was under test the chosen rotational speed was 44 rpm. Energy output and wind speed were measured over one minute time intervals and the average power and wind speed determined. The test was continued until a sufficient range of wind speeds had been covered. The 16 minute average results were then sorted in 'bins' 0.5 m/s of wind speed wide and a fairly smooth power versus wind speed curve was obtained, as shown in Figure 3.60.

The turbine has a diameter of 17 m and would be expected to produce rather more power than shown above if operated at a higher rotational speed.

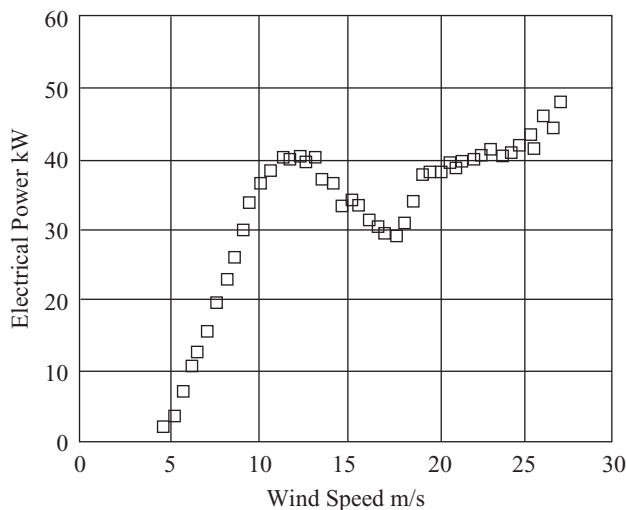


Figure 3.60 Power versus wind speed curve from the binned measurements of a three-blade stall regulated turbine

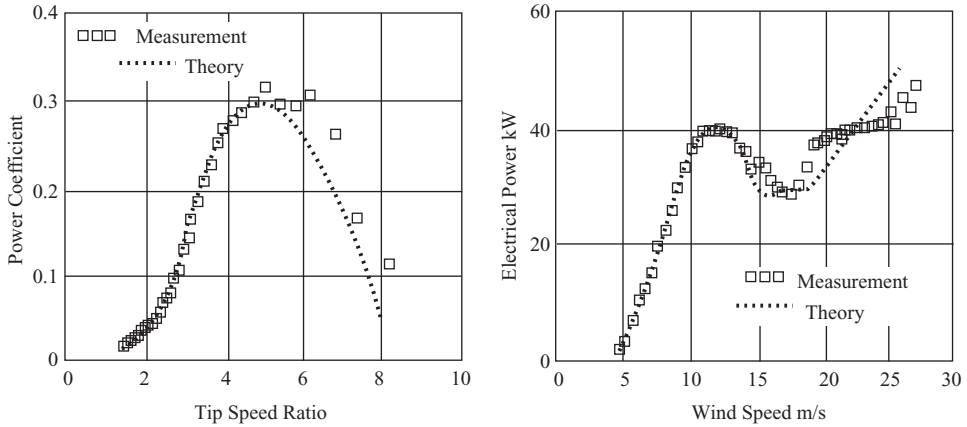


Figure 3.61 Comparison of measured and theoretical performance curves

From the data in Figure 3.58 the $C_p - \lambda$ curve can be derived. The tip speed of the blades is $44 \times \pi/30 \text{ rad/s} \times 8.5 \text{ m} = 39.2 \text{ m/s}$, the swept area is $\pi \times 8.5^2 = 227 \text{ m}^2$ and the air density was measured (from air pressure and temperature readings) at 1.19 kg/m^3 . Therefore,

$$\lambda = \frac{39.2}{\text{wind speed}} \quad \text{and} \quad C_p = \frac{\text{Power} \cdot \lambda^3}{\frac{1}{2} \cdot 1.19 \cdot 39.2^3 \cdot 227}$$

The mechanical and electrical losses were estimated at 5.62 kW and this value was used to adjust the theoretical values of C_p . The resulting comparison of measured and theoretical results is shown in Figure 3.61.

This comparison looks reasonable and shows that the theory is reliable but the quality of the theoretical predictions really relies upon the quality of the aerofoil data. The blade and aerofoil design are the same as given in Section 3.10.

One last point should be made before classifying the theory as complete: it would be as well to look at the raw, one minute average data, before it was reduced down by a binning process and is shown in Figure 3.62. In the post stall region there seems to be a much more complex process taking place than the simple theory predicts and this could be caused by unsteady aerodynamic effects.

3.15 Variable speed operation

If the speed of the rotor can be continuously adjusted such that the tip speed ratio remains constant at the level which gives the maximum C_p , then the efficiency of the turbine will be significantly increased. The pitch angle is kept constant and the generator torque is adjusted to increase speed. Pitch control regulation is required in conditions above the rated wind speed when the rotational speed is kept constant.

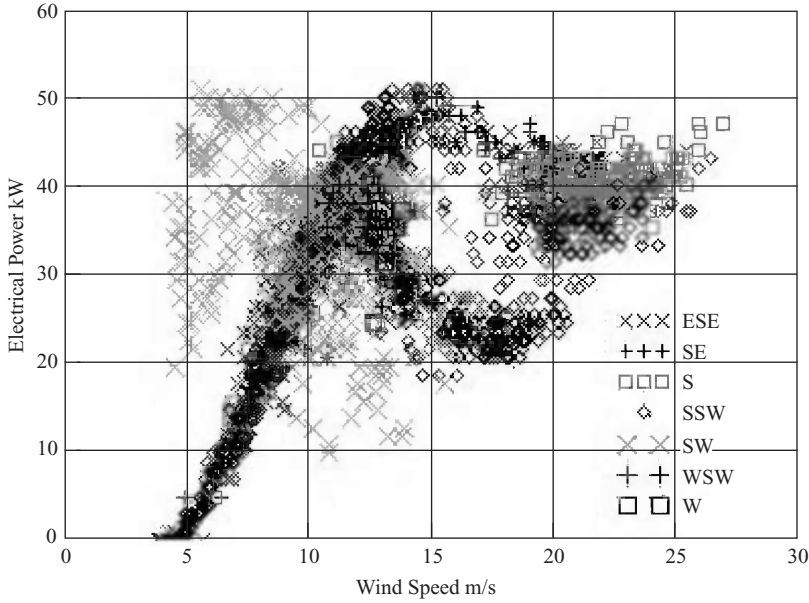


Figure 3.62 Measured raw results of a three-blade wind turbine

3.16 Estimation of energy capture

The quantity of energy that can be captured by a wind turbine depends upon the power versus wind speed characteristic of the turbine and the wind speed distribution at the turbine site.

The wind speed distribution at a site can be represented by the Weibull function: the probability that the wind speed will exceed a value U is

$$F(U) = e^{-\left(\frac{U}{c}\right)^k} \quad (3.92)$$

where c , called the scale factor, is a characteristic speed related to the average wind speed at the site by

$$c = \frac{\bar{U}}{\Gamma\left(1 + \frac{1}{k}\right)} \quad (3.93)$$

(Γ being the gamma function and k a shape parameter). Let $U/\bar{U} = u$ a normalised wind speed.

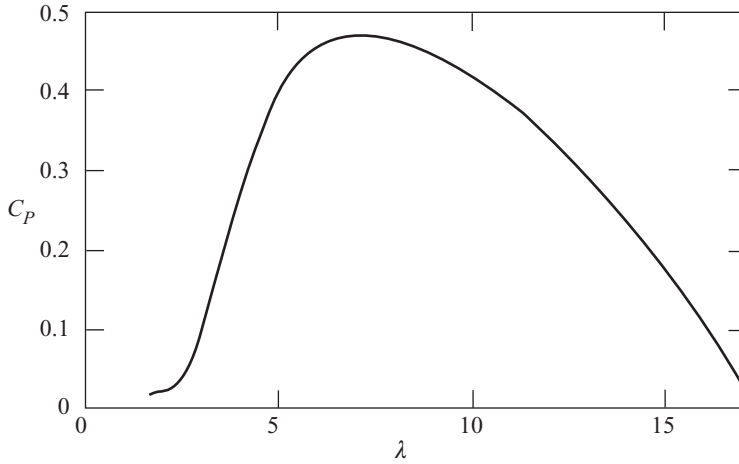


Figure 3.63 $C_P - \lambda$ curve for a design tip speed ratio of seven

The wind speed distribution density is then the modulus of the derivative of Equation 3.92 with respect to u .

$$f(u) = k \left(\frac{\bar{U}}{c} \right)^k u^{k-1} e^{-\left(\frac{\bar{U}}{c} u \right)^k} \quad (3.94)$$

That is, the probability that the wind speed lies between u and $u + \delta u$ is $f(u)\delta u$.

The performance curve shown in Figure 3.63 is for a turbine designed with an optimum tip speed ratio of seven.

As an example, assume that the turbine is stall regulated and operates at a fixed rotational speed at a site where the average wind speed is 6 m/s and the Weibull shape factor $k = 1.8$ then, from Equation 3.93, the scale factor $c = 6.75$ m/s.

Figure 3.64 shows the $K_P - 1/\lambda$ curve for the turbine: from inspection of that curve the tip speed ratio at which stall (maximum power) occurs is 3.7 and the corresponding C_P is 0.22.

The required maximum electrical power of the machine is 500 kW, the transmission loss is 10 kW, the mean generator efficiency is 90% and the availability of the turbine (amount of time for which it is available to operate when maintenance and repair time is taken into account) is 98%.

The maximum rotor shaft power (aerodynamic power) is then

$$P_s = (500 + 10)/0.9 = 567 \text{ kW}$$

The wind speed at which maximum power is developed is 13 m/s; therefore, the rotor swept area must be, assuming an air density of 1.225 kg/m^3 ,

$$567000/(1/2 \times 1.225 \times 13^3 \times 0.22) = 1.92 \times 10^3 \text{ m}^2$$

The rotor radius is, therefore, 24.6 m.

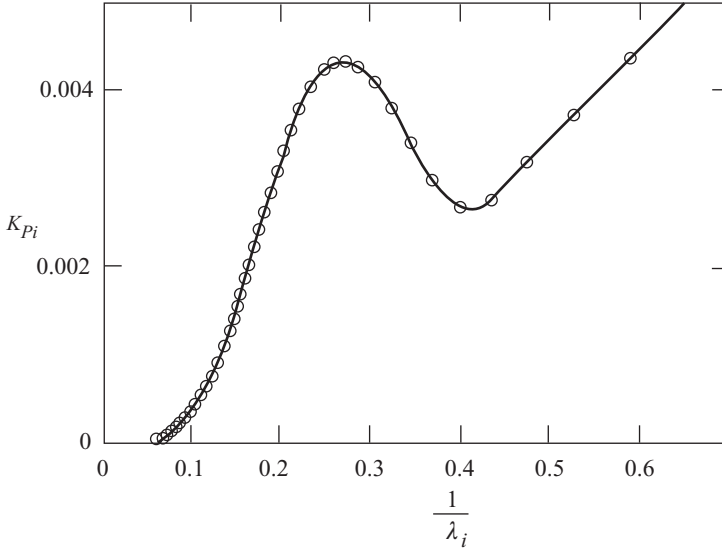


Figure 3.64 $K_P - 1/\lambda$ curve for the example fixed speed, stall regulated turbine

The tip speed of the rotor will be $3.7 \times 13 \text{ m/s} = 48.1 \text{ m/s}$ and so the rotational speed will be $48.1/24.6 \text{ rad/s} = 1.96 \text{ rad/s}$, which is $1.96 \times 60/2\pi \text{ rev/min} = 18.7 \text{ rev/min}$.

The power versus wind speed curve for the turbine can then be obtained from Figure 3.64 and is shown in Figure 3.65.

$$\begin{aligned} \text{Power (electrical)} &= \left(K_P \times \frac{1}{2} \times 1.225 \text{ kg/m}^3 \times (48.1 \text{ m/s})^3 \right. \\ &\quad \left. \times 1.92 \times 10^3 \text{ m}^2 - 10 \times 1000 \text{ W} \right) \times 0.9 \\ \text{Wind speed} &= 48.1 \text{ m/s}/\lambda \end{aligned}$$

The resulting power curve is shown in Figure 3.65.

To determine the energy capture of the turbine over a time period T multiply the power by $f(u) \times T$ (Equation 3.95); because $w(u)$ is the proportion of time T spent at a normalised wind speed u .

$$f(u) \cdot \delta u = \frac{\delta T}{T} \quad (3.95)$$

and

$$\int_0^\infty f(u) \cdot du = 1 \quad (3.96)$$

Plot against u (Figure 3.66) and integrate over the operational wind speed range of the turbine to give the energy capture.

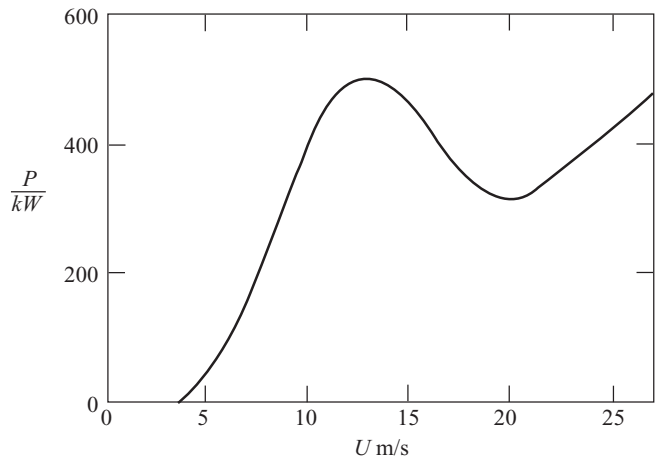


Figure 3.65 Power versus wind speed for the example fixed speed, stall regulated turbine

The operational speed range will be between the cut-in speed and the cut-out speed. The cut-in speed is determined by the transmission losses: at what wind speed does the turbine begin to generate power? The cut-in speed is usually chosen to be somewhat higher than the zero power speed, in the present case, say 4 m/s.

The cut-out speed is chosen to protect the turbine from high loads, usually about 25 m/s. The total energy captured (E) by the turbine in a time period T is

$$T \int_{\frac{U_{ci}}{\bar{U}}}^{\frac{U_{co}}{\bar{U}}} P(u)f(u)du = E \tag{3.97}$$

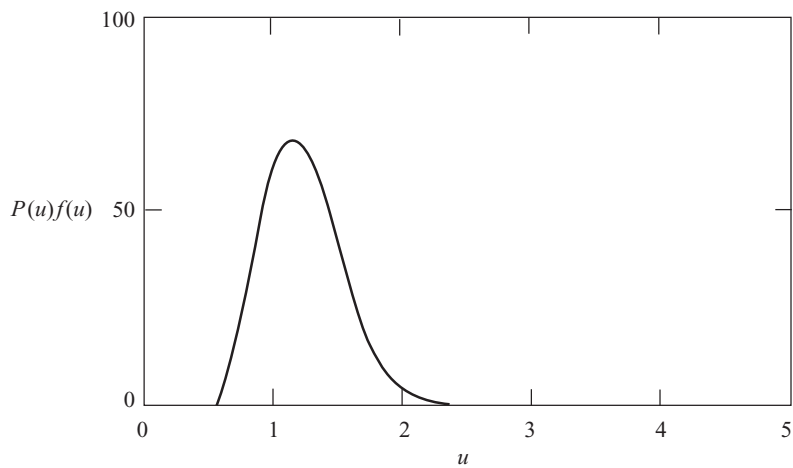


Figure 3.66 Energy capture curve

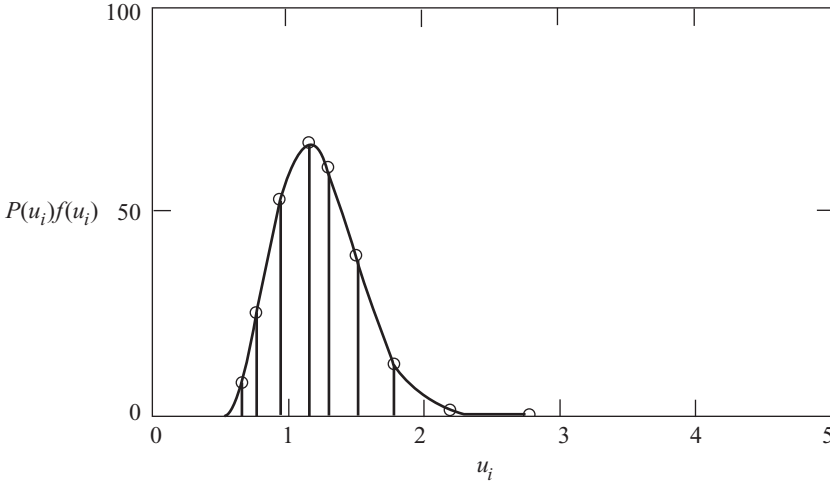


Figure 3.67 Energy capture curve for numerical integration

which is the area under the curve of Figure 3.66 times the time T . Unfortunately, the integral does not have a closed mathematical form in general and so a numerical integration is required, such as the trapezoidal rule or, for better accuracy, Simpson's rule.

For time period of one year $T = 365 \times 24$ hours then for the ten data points shown in Figure 3.67 the energy capture will be, using the trapezoidal rule,

$$E = 0.98 \cdot T \sum_{i=1}^9 (P_{i+1}f(u_{i+1}) + P_i f(u_i)) \frac{(U_{i+1} - U_i)}{2\bar{U}} \quad (3.98)$$

where

$$E = 4.5413 \cdot 10^8 \cdot kWh$$

Even though the upper limit of integration $u_{co} = 4.17$ is greater than highest value of u shown in Figure 3.67 it is clear that almost no energy is captured between those speeds.

A turbine which has pitch control would be able to capture more energy but at the expense of providing the control system and the concomitant reduction in reliability. A turbine operating at variable speed (constant tip speed ratio) until maximum power is reached and then at constant speed with pitch control at higher wind speeds would capture the maximum possible amount of energy in a given time. The power curve for such a machine is shown in Figure 3.68.

The annual energy capture would be, see Figure 3.68,

$$E = 4.8138 \cdot 10^8 \cdot kWh$$

which is a 6% increase in energy capture compared with the fixed speed, stall regulated machine. Whether or not the increase in energy captured is economically worthwhile is a matter for debate. However, variable speed operation has a number of other advantages – see Section 6.9.3.

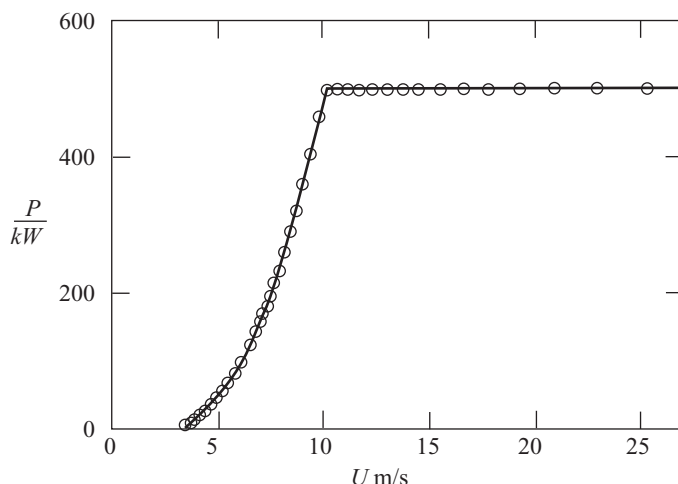


Figure 3.68 Power vs. wind speed variable speed, pitch regulated turbine

3.17 Wind turbine aerofoil design

3.17.1 Introduction

For many years the wind turbine industry relied on aeronautical experience for the aerodynamic design of turbine blades but it became clear that aerofoil sections that were right for aircraft wings were not necessarily right for wind turbine blades.

There were reports of turbines in the 1970s having to be regularly hosed with water to clear accumulated debris on the blades in order to restore power levels that had fallen dramatically. An aerofoil that was tolerant to leading edge roughness was required.

Most early turbines of rated power greater than about 50 kW operated at constant rotational speed and relied upon passive stall for power control. With most aircraft aerofoil sections the stall produced a sudden sharp loss of power output that was not recovered until the wind speed increased. The stalling resulted in significant losses of energy capture. Thus, another requirement for a wind turbine aerofoil was a gentle stall.

A popular range of aerofoil designs was, and still is, the NACA 6 digit series, an example of which is discussed in Section 3.9. Although more tolerant to leading edge grime the NACA 6 digit series is no better than the NACA 4 digit series described in Appendix A3. The main reason for the popularity of the NACA aerofoils is because high quality experimental data is available from tests that were carried out in the 1930s in the pressurised wind tunnel built by NACA (National Advisory Committee for Aeronautics, superseded by NASA in 1959). The NACA technical reports are available free of charge on the NASA website.

3.17.2 The NREL aerofoils

The development of special-purpose aerofoils for horizontal-axis wind turbines began in 1984 jointly between the National Renewable Energy Laboratory (NREL), formerly the Solar Energy Research Institute (SERI), and Airfoils, Incorporated, Tangler and Somers (1995). Since that time nine aerofoil families have been designed for various size rotors with a

Table 3.3 Summary of the NREL aerofoils and their applications

Diameter	Type	Aerofoil thickness	Primary	Tip	Root
3–10 m	Variable speed	Thick	—	S822	S823
	Variable pitch				
10–20 m	Variable speed	Thin	S802	S802	S804
	Variable pitch			S803	
10–20 m	Stall regulated	Thin	S805	S806	S807
			S805A	S806A	S808
10–20 m	Stall regulated	Thick	S819	S820	S821
20–30 m	Stall regulated	Thick	S809	S810	S811
20–30 m	Stall regulated	Thick	S812	S813	S814
					S815
20–40 m	Variable speed	—	S825	S826	S814
	Variable pitch				S815
30–50 m	Stall regulated	Thick	S816	S817	S818
40–50 m	Stall regulated	Thick	S827	S828	S818
40–50 m	Variable speed	Thick	S830	S831	S818
	Variable pitch			S832	

principal requirement that they have a maximum lift coefficient that is maintained in the presence of leading edge surface roughness.

The primary design tool was based on the work of Eppler (1990, 1993) who developed a method of determining the nature of the two-dimensional viscous flow around an aerofoil of any profile. The Eppler method includes flow separation at the stall and has proved to be very successful.

In addition, the several different aerofoil families have been designed for stall-regulated, variable-pitch and variable-rpm wind turbines.

For stall-regulated rotors, improved post-stall power control is achieved through the design of aerofoils for the outer sections of a blade that limit the maximum lift coefficient. The same aerofoils have a relatively high thickness to chord ratio in order to accommodate over-speed control devices.

For variable-pitch and variable-rpm rotors, outer section aerofoils have a high maximum lift coefficient allowing low blade solidity.

Generally, aerofoil cross-sections with a high thickness to chord ratio, structural designs of high stiffness and strength without causing a large weight penalty and aerofoils of low thickness result in less drag.

Annual energy capture improvements that are claimed for the NREL airfoil families are of the order of 23%–35% for stall-regulated turbines, 8%–20% for variable-pitch turbines, and 8%–10% for variable-rpm turbines. The improvement for stall-regulated turbines has been verified in field tests.

The aerofoil shape co-ordinates for some of the NREL aerofoils are available on the website of the National Wind Technology Center (NWTC) at Golden Colorado; measured aerofoil data for some aerofoils is also available. A licence must be purchased for information about those aerofoils that are restricted.

Some of the NREL large blade aerofoil profiles are illustrated in Figure 3.69.

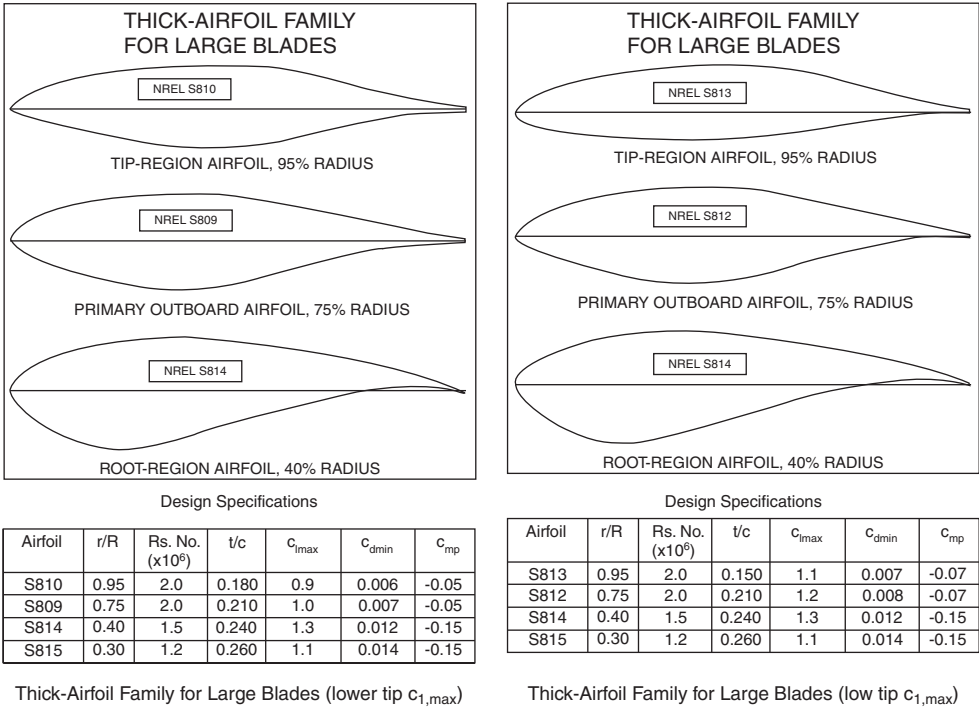


Figure 3.69 NREL aerofoil profiles for large blades. Reproduced from Tangler and Somers (1995)

3.17.3 The Risø aerofoils

The Risø National Laboratory in Denmark have also developed families of aerofoil designs for wind turbines with similar objectives to the NREL series, Fugslang and Bak (2004). Although the aerodynamic design techniques of the two laboratories were different there is, perhaps not surprisingly, a significant similarity between the actual designs.

The design tools for the Risø aerofoils were the X-FOIL code developed by Drela, a development of the work of Eppler (1990, 1993) and the Ellipsys-2D CFD code developed at the Technical University of Denmark by Sørensen (1995).

Three families of aerofoils have been developed at Risø, Risø-A, Risø-P and Risø-B. See Tables 3.4–3.6. The Risø-A family, Figure 3.70 and Table 3.4, was designed in the 1990s and was intended for stall controlled turbines, however, sensitivity to surface roughness was found to be higher than expected in field tests. The Risø-A family of aerofoil profiles are illustrated in Figure 3.70.

The Risø-P family of just four aerofoils, Figure 3.71 and Table 3.5, was designed to replace the corresponding profiles in the Risø-A series for use on variable pitch and variable speed rotors. The Risø-P family of aerofoil profiles are illustrated in Figure 3.71.

The Risø-B family was designed as six separate aerofoils with an extended range of thickness to chord ratio from 15–36%, see Figure 3.72 and Table 3.6. The aerofoils, generally, have high maximum lift coefficients for use on multi-megawatt size rotors with low solidity, flexible blades having variable speed pitch control. The Risø-B family of aerofoil profiles are illustrated in Figure 3.72.

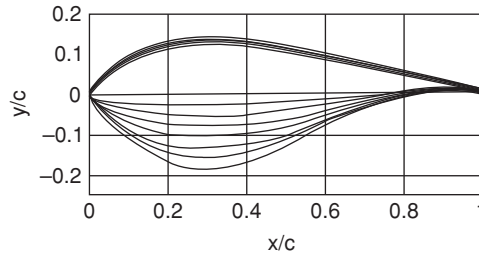


Figure 3.70 The Risø-A series of aerofoil profiles

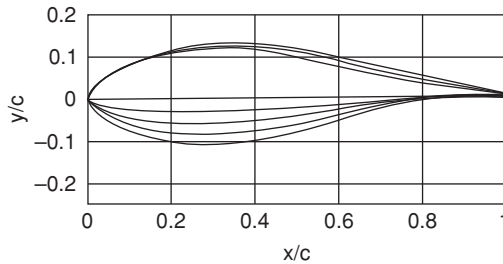


Figure 3.71 The Risø-P series of aerofoil profiles

In Tables 3.4–3.7 the ‘design c_l ’ is the value of the lift coefficient that corresponds to the maximum lift to drag ratio. It is a design feature of the Risø aerofoils that the design c_l is high so that a blade will be most efficient at low solidity.

3.17.4 The Delft aerofoils

The Delft University of Technology in the Netherlands has also developed a number of aerofoils for wind turbine rotors (Timmer and van Rooij, 2003), see Figure 3.73 and Table 3.7. As with the NREL and Risø aerofoils, the principal feature driving the designs was surface roughness insensitivity but more emphasis was placed upon seeking designs for thick aerofoils in order to gain a structural advantage. The Delft University series of aerofoil profiles are illustrated in Figure 3.73.

Table 3.4 The principal characteristics of the Risø-A series

Aerofoil	Max t/c %	x/c at max t/c	y/c at T.E.	$Re \times$ 10^{-6}	α°	c_l Max	Design α	Design C_l	Max C_l/c_d
Risø-A1- 15	15	0.325	0.0025	3.00	−4.0	1.50	6.0	1.13	168
Risø-A1- 18	18	0.336	0.0025	3.00	−3.6	1.53	6.0	1.15	167
Risø-A1- 21	21	0.298	0.005	3.00	−3.3	1.45	7.0	1.15	161
Risø-A1- 24	24	0.302	0.01	2.75	−3.4	1.48	7.0	1.19	157
Risø-A1- 27	27	0.303	0.01	2.75	−3.2	1.44	7.0	1.15	N/A
Risø-A1- 30	30	0.300	0.01	2.50	−2.7	1.35	7.0	1.05	N/A
Risø-A1- 33	30	0.304	0.01	2.50	−1.6	1.20	7.0	0.93	N/A

Table 3.5 The principal characteristics of the Risø-P series

Aerofoil	Max t/c %	x/c at max t/c	y/c at T.E.	Re × 10 ⁻⁶	α	c _l Max	Design α	Design C _l	Max C _l /c _d
Risø-P- 15	15	0.328	0.0025	3.00	-3.5	1.49	6.0	1.12	173
Risø-P- 18	18	0.328	0.0025	3.00	-3.7	1.50	6.0	1.15	170
Risø-P- 21	21	0.323	0.005	3.00	-3.5	1.48	6.0	1.14	159
Risø-P- 24	24	0.320	0.01	2.75	-3.7	1.48	6.0	1.17	156

Table 3.6 The principal characteristics of the Risø-B series

Aerofoil	Max t/c %	x/c at max t/c	y/c at T.E.	Re × 10 ⁻⁶	α	c _l Max	Design α	Design C _l	Max C _l /c _d
Risø-B1- 15	15	0.278	0.006	6.00	-4.1	1.92	6.0	1.21	157
Risø-B1- 18	18	0.279	0.004	6.00	-4.0	1.87	6.0	1.19	166
Risø-B1- 21	21	0.278	0.005	6.00	-3.6	1.83	6.0	1.16	139
Risø-B1- 24	24	0.270	0.007	6.00	-3.1	1.76	6.0	1.15	120
Risø-B1- 30	30	0.270	0.01	6.00	-2.1	1.61	5.0	0.90	N/A
Risø-B1- 36	36	0.270	0.012	6.00	-1.3	1.15	5.0	0.90	N/A

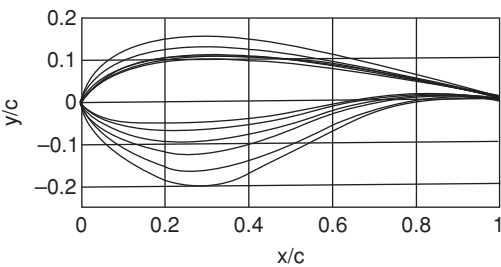


Figure 3.72 The Risø-B series of aerofoil profiles

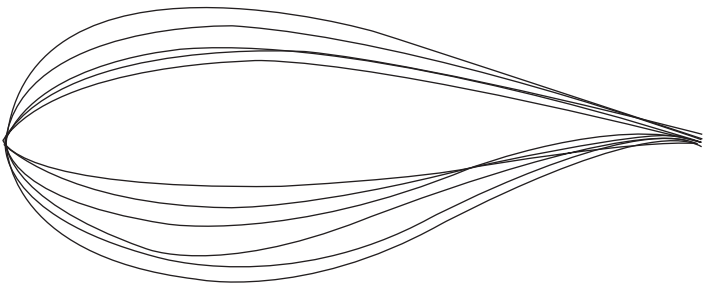


Figure 3.73 The DU series of aerofoil profiles

Table 3.7 The principal characteristics of the DU series

Aerofoil	Max t/c %	x/c at max t/c	y/c at T.E.	Re $\times$ 10 ⁻⁶	α	c _l Max	Design α	Design C _l	Max C _l /c _d
DU 96-W-180	18	0.3	0.0018	3.00	-2.7	1.26	6.59	1.07	145
DU 00-W-212	21.2	0.3	0.0023	3.00	-2.7	1.29	6.5	1.06	132
DU 91-W2-250	25	0.3	0.0054	3.00	-3.2	1.37	6.68	1.24	137
DU 97-W-300	30	0.3	0.0048	3.00	-2.2	1.56	9.3	1.39	98
DU 00-W-350	35	0.3	0.01	3.00	-2.0	1.39	7.0	1.13	81
DU 00-W-401	40.1	0.3	0.01	3.00	-3.0	1.04	5.0	0.82	54

The design tool for the Delft aerofoils was the R-FOIL code a modification made at Delft of the X-FOIL code to include the effects of stall delay.

The two thickest aerofoils have not been tested in a wind tunnel and the characteristics have been determined by calculation.

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Appendix A3 lift and drag of aerofoils

The forces acting on a body immersed in a fluid moving fluid can be resolved into stream-wise (drag) and normal (lift) components. Neglecting buoyancy, the fluid mechanics which give

rise to lift and drag are associated with the boundary layer of slow moving fluid close to the body's surface. The fluid force on the surface of the body can be either parallel to the surface (viscous or skin friction force) or normal to the surface (pressure force).

For a thorough understanding of the phenomena of lift and drag an aerodynamics text should be consulted but for the purposes of wind turbine aerodynamics the basic results are given below.

A3.1 Definition of drag

The drag on a body immersed in an oncoming flow is defined as the force on the body in a direction parallel to the flow direction.

In a very slowly moving fluid the drag on a body may be directly attributable to the viscous, frictional shear stresses set up in the fluid due to the fact that, at the body wall, there is no relative motion.

This type of flow is known as Stokes flow after Sir George Stokes.

A century and a half before Stokes, Isaac Newton showed that the shear stress τ at a boundary wall, or between two layers of fluid moving relative to one another, is proportional to the transverse velocity gradient at the boundary, or between the two layers.

$$\tau = \mu \frac{du}{dy} \quad (\text{A3.1})$$

where the constant of proportionality is μ the fluid viscosity.

Using Newton's theory, Stokes determined the drag force on a sphere in creeping flow (Figure A3.1).

$$\text{Drag} = 3\pi\mu U d \quad (\text{A3.2})$$

where d is the sphere diameter and U is the general flow velocity.

The inviscid flow pattern around a cylinder, Figure A3.2, appears very similar to that of creeping flow but the nature of the flow is very different indeed. By definition inviscid flow causes no viscous drag but it also causes no pressure drag, that is, drag caused by

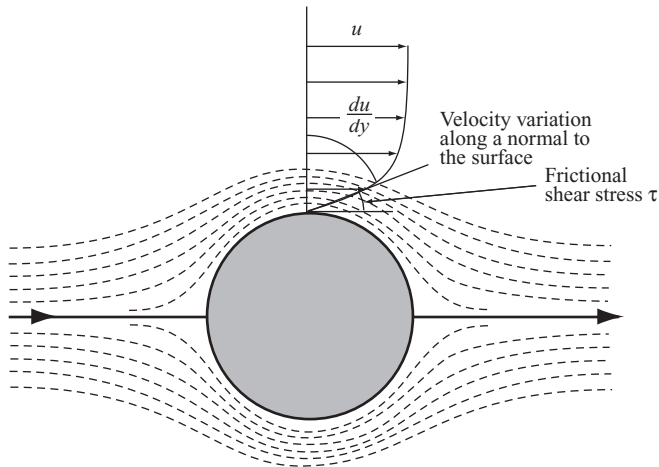


Figure A3.1 Creeping flow past a circular cylinder

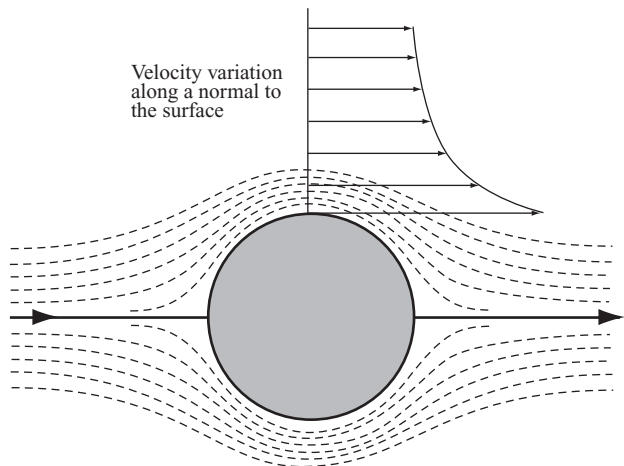


Figure A3.2 Inviscid flow pattern around a cylinder

a pressure force aggregated over the whole surface area. The pressure distribution for the inviscid flow past a cylinder is shown in Figure A3.3, where the atmospheric pressure p_∞ has been subtracted from the pressure around the surface. The symmetry of the pressure distribution fore and aft shows clearly that no pressure drag arises. At the nose of the body the flow is brought exactly to rest and this is called the stagnation point. Another stagnation point occurs at the rear of the body.

In a real fluid, when the viscosity is low and the velocity is relatively high, the drag force that exists is due primarily to an asymmetric pressure distribution, fore and aft, Figure A3.5.

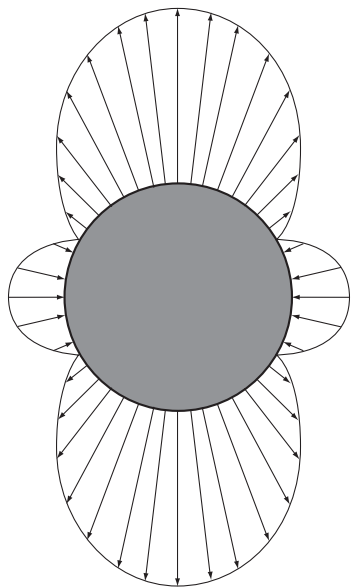


Figure A3.3 Inviscid flow pressure distribution around a cylinder

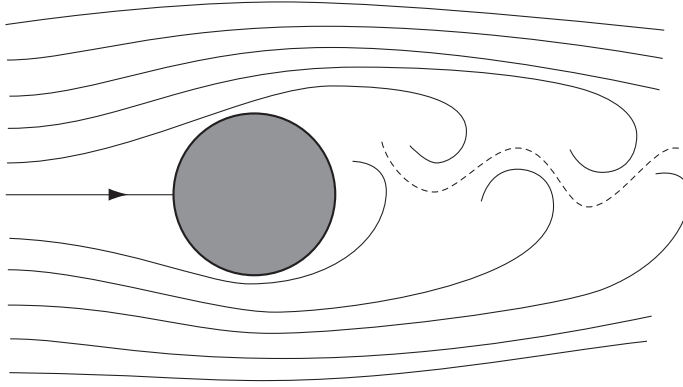


Figure A3.4 Separated flow pattern around a cylinder

This is caused by the fact that the fluid does not follow the boundary of the body but separates from it leaving low pressure, stagnant fluid in the wake, Figure A3.4. On the upstream side the flow remains attached and the pressure is high.

A3.2 Drag coefficient

If Stokes' drag equation (Equation A3.2) for the sphere is re-arranged, giving

$$Drag = 3\pi\mu Ud = \left(24 \left(\frac{\mu}{\rho Ud}\right)\right) \left(\frac{1}{2}\rho U^2\right) \left(\frac{\pi d^2}{4}\right) \quad (A3.3)$$

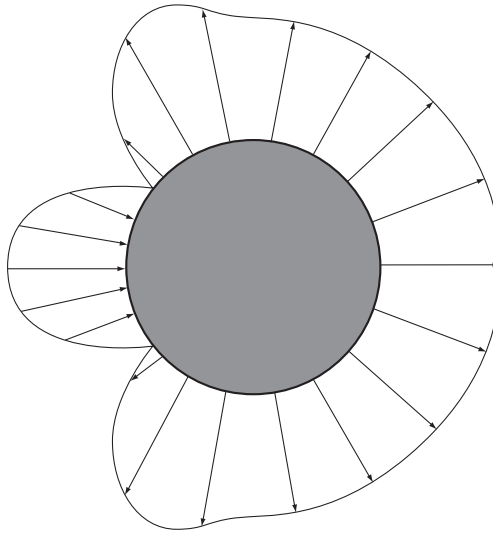


Figure A3.5 Separated flow pressure distribution around a cylinder

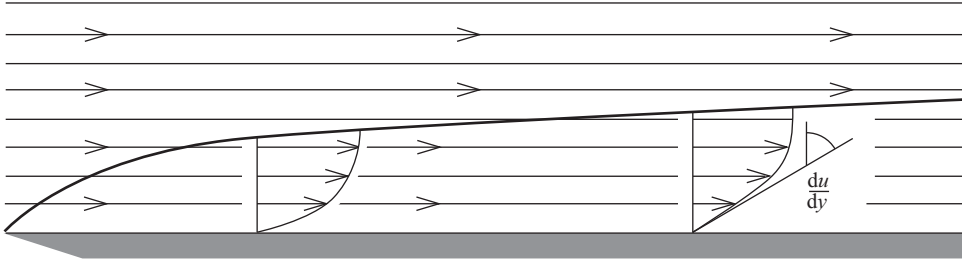


Figure A3.6 Boundary layer showing the velocity profile

it is then in the standard form of drag coefficient (C_D) $\times$ dynamic pressure ($\frac{1}{2}\rho U^2$) $\times$ frontal area (A). The drag coefficient is then defined as

$$C_D = \frac{\text{Drag}}{\frac{1}{2}\rho U^2 A} \quad (\text{A3.4})$$

Note that, $\rho U d / \mu$ is known as the Reynolds number (Re) and represents the ratio of the inertia force acting on a unit volume of fluid, as it is accelerated by a pressure gradient, and the viscous force on the same volume of fluid which is resisting the motion of the fluid. For high Reynolds numbers viscous forces are low and vice versa. The drag coefficient term in Equation A3.3 is $C_D = 24/Re$ and is clearly a function only of the Reynolds number: this turns out to be valid for all bodies in incompressible flow but the functional relationship is not usually as simple as in the above case. However, it can be stated, generally, that C_D falls with increasing Reynolds number.

A3.3 The boundary layer

The reason for the separated flow at the higher Reynolds numbers is the existence of a thin boundary layer of slow moving fluid, close to the body surface, within which viscous forces predominate. See Figure A3.6. Outside of this layer the flow behaves almost inviscidly. The drag on the body caused directly by viscosity is quite small but the effect on the flow pattern is profound.

The drag on an aerofoil can be attributed both to pressure and viscous sources and the drag coefficient varies significantly with both angle of attack and Reynolds number.

A3.4 Boundary layer separation

Referring to Figure A3.3, the inviscid flow pressure distribution around a cylinder, fore and aft the pressure is high; above and below the pressure is low. The fluid on the downstream side is slowing down against an adverse pressure gradient and, at the wall boundary, it slows down exactly to a standstill at the rear stagnation point.

In the real flow the fluid the boundary layer (Figure A3.6), which has already been slowed down by viscosity, comes to a halt well before the stagnation point is reached and the flow begins to reverse under the action of the adverse pressure. At this point, where the pressure is

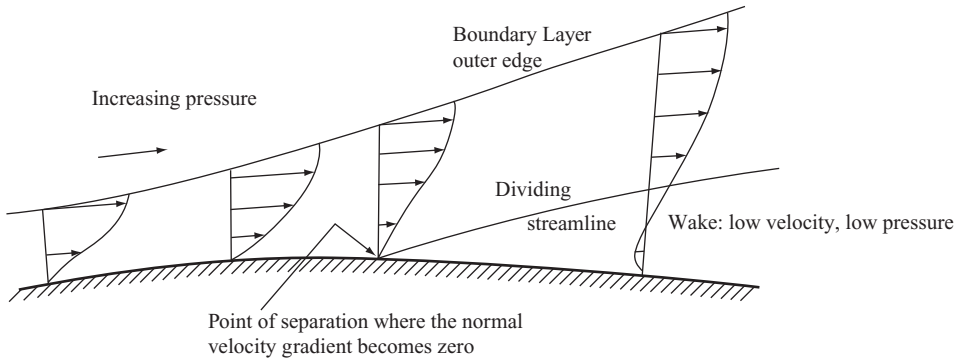


Figure A3.7 Separation of a boundary layer

still low, the boundary layer separates from the body surface forming a wake of stagnant, low pressure fluid (Figure A3.7), the resulting pressure distribution is thereby dramatically altered as shown in Figure A3.5. The high pressure acting at and around the forward stagnation point is no longer balanced by the high pressure at the rear and so a drag-wise pressure force is exerted.

A3.5 Laminar and turbulent boundary layers

A boundary layer grows in thickness from the forward stagnation point, or leading edge. Initially, the flow in the layer is ordered and smooth (laminar) but, at a critical distance l from the stagnation point, characterised by $Re_{crit} = \rho U l / \mu$, the flow begins to become turbulent, (Figure A3.8). This turbulence causes mixing of the boundary layer with the faster moving fluid outside resulting in re-energisation and delaying of the point of separation. The result is to reduce the pressure drag, because the low pressure stagnant rear area is reduced, to increase the viscous (frictional) drag, because the velocity gradient at the surface is increased, and increase the boundary layer thickness.

The coefficient of drag, therefore, varies with Re in a complex fashion, Figure A3.9. For small bodies at low speeds the critical Re is never reached and so separation takes place early. For large bodies, or high speeds, turbulence develops quickly and separation is delayed.

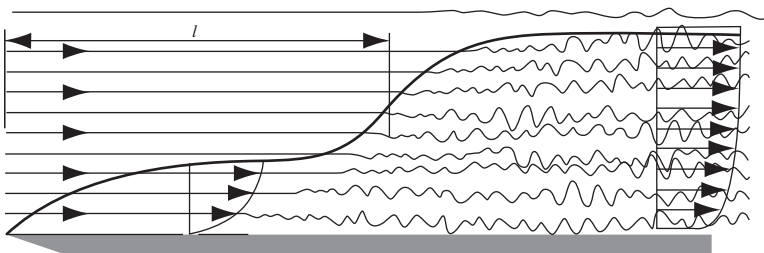


Figure A3.8 Laminar and turbulent boundary layers

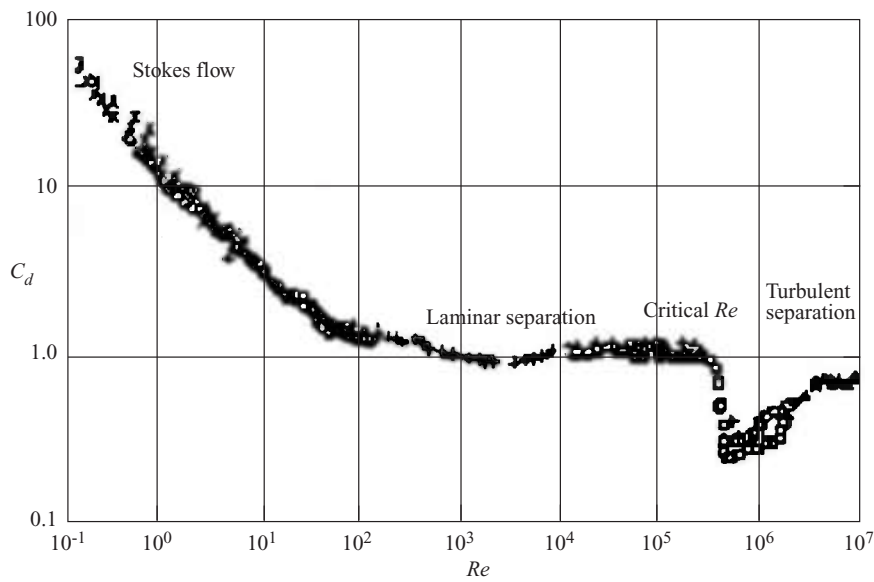


Figure A3.9 Variation of C_D with Re for a long cylinder

Turbulence can be artificially triggered by roughening the body surface or simply by using a ‘trip wire’. General flow turbulence tends to produce turbulent boundary layers at Reynolds numbers ostensibly below the critical value and this certainly seems to happen in the case of wind turbine blades. A sharp edge on a body will *always* cause separation. For a flat plate broad side on to the flow, Figure A3.10, the boundary layer separates at the sharp edges and C_D is almost independent of Re , but *is* dependent upon the plate’s aspect ratio.

So-called streamlined bodies such as an aerofoil taper gently in the aft region so that the adverse pressure gradient is small and separation is delayed until very close to the trailing

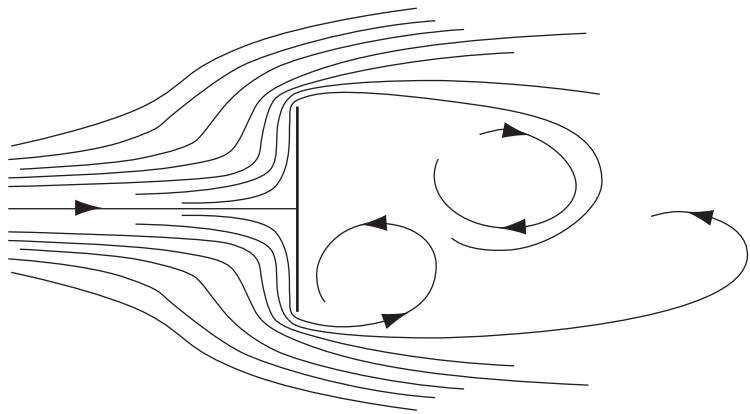


Figure A3.10 Separated flow past a flat plate

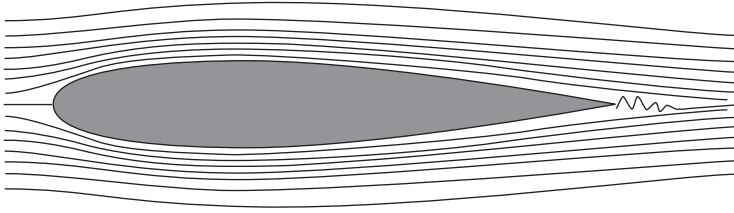


Figure A3.11 Flow past a streamlined body

edge, (Figure A3.11). This produces a very much narrower wake and a very low drag because it is largely caused by skin friction rather than pressure.

A3.6 Definition of lift and its relationship to circulation

The lift on a body immersed is defined as the force on the body in a direction normal to the flow direction.

Lift will only be present if the flow incorporates a circulatory flow about the body such as that which exists about a spinning circular cylinder. If the fluid also has a uniform velocity U past the cylinder, the resulting flow field is as shown in Figure A3.12. The velocity above the cylinder is increased, and so the static pressure there is reduced. Conversely, the velocity beneath is slowed down, giving an increase in static pressure. There is clearly a normal force upwards on the cylinder, a lift force.

The phenomenon is known as the Magnus effect after its original discoverer and explains, for example, why spinning tennis balls veer in flight. The circulatory flow, shown in Figure A3.13, is generated by skin friction and has the same structure as that of a vortex.

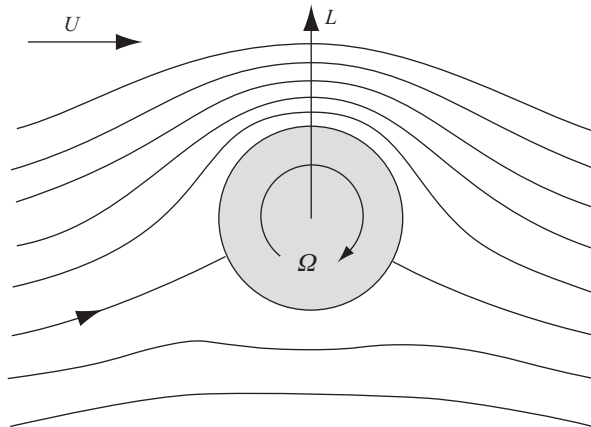


Figure A3.12 Flow past a rotating cylinder

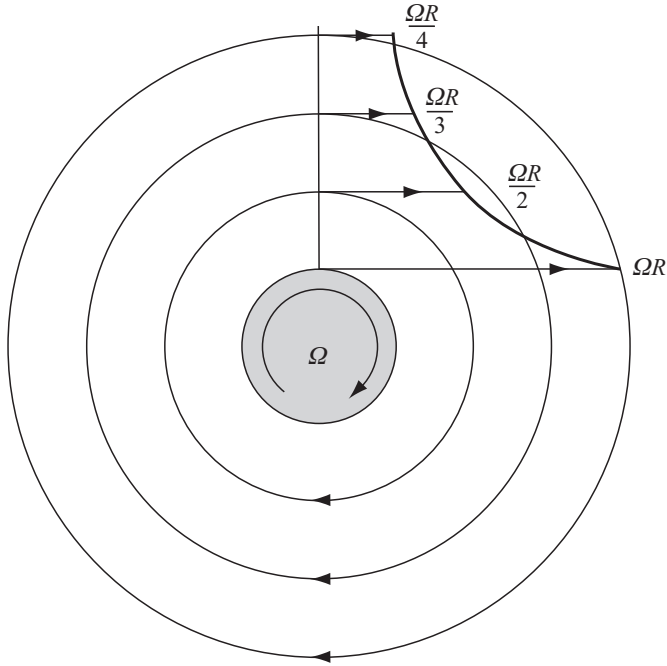


Figure A3.13 Circulatory flow round a rotating cylinder

The lift force is given by the Kutta-Joukowski theorem named after the two pioneering aerodynamicists who, independently, realised that this was the key to the understanding of the phenomenon of lift.

$$L = \rho(\Gamma \times U) \quad (\text{A3.5})$$

where Γ is the circulation, or vortex strength, around the cylinder, defined as the integral

$$\Gamma = \oint v ds \quad (\text{A3.6})$$

around any path enclosing the cylinder and v is the velocity tangential to the path s . For convenience, choosing a circular path of radius r around the cylinder, and ignoring the general flow velocity U , then it can be shown that $v = k/r$, where k is a constant.

At the cylinder wall $r = R$, so $v_R = \Omega R = k/R$. Therefore, $k = \Omega R^2$.

The circulation Γ , which is the same for every path enclosing the cylinder, is given by

$$\Gamma = \oint \frac{\Omega R^2}{r} ds = \int_0^{2\pi} \Omega R^2 d\theta = 2\pi \Omega R^2 \quad (\text{A3.7})$$

To achieve a circulatory flow about a non-rotating body it must have a sharp trailing edge like an aerofoil cross-sectional shape or a thin plate. An aerofoil works in a similar manner to

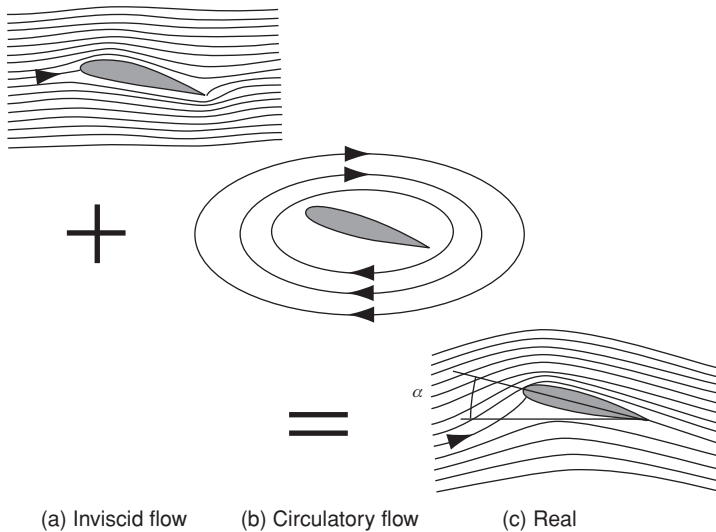


Figure A3.14 Flow past an aerofoil at a small angle of attack

the spinning cylinder and does so because of its sharp trailing edge. Consider an aerofoil at a small angle of attack α to the oncoming flow. The inviscid flow pattern around the aerofoil, in which no boundary layer forms, is as shown in Figure A3.14a. The theoretical inviscid flow condition is such that no force on the aerofoil exists at all.

In real flow, Figure A3.14c, boundary layer separation occurs at the sharp trailing edge, causing the flow to leave the edge smoothly. The separation leaves no low pressure wake so the flow remains attached everywhere else and the flow pattern is now altered such that there is a net circulation around the aerofoil (Figure A3.14b) increasing the velocity over the top and reducing it below, resulting in a lift force. There can be no flow around the sharp trailing edge because this would require very high local velocities that are precluded by the boundary layer. The drag is very low because, in the absence of a wake, it is attributable largely to skin friction caused by the shearing stresses in the boundary layer. The situation, which is imposed by the sharp trailing edge, is known as the Kutta condition.

In a manner similar to the Magnus effect, a pressure difference occurs across the aerofoil and the overall circulation Γ can be shown to be $\pi U c \sin \alpha$, where c is the chord length of the aerofoil and α is termed the *angle of attack*. Although the velocities and pressures above and below the aerofoil at the trailing edge must be the same, the particles which meet there are not the same ones that parted company at the leading edge; the particle which travelled above the aerofoil reaches the trailing edge first because it is speeded up by the circulatory flow.

The pressure variation (minus the ambient static pressure of the undisturbed flow) around an aerofoil is shown in Figure A3.15. The upper surface is subject to suction (with the ambient pressure subtracted) and is responsible for most of the lift force. The pressure distribution is calculated without the presence of the boundary layer.

Figure A3.16 shows the same distribution with the pressure coefficient ($C_p = p - p_\infty / (1/2)\rho U^2$) plotted against the chord-wise co-ordinate of the aerofoil profile: the full line

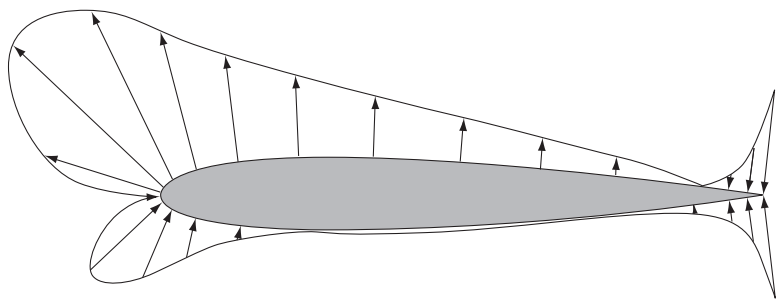


Figure A3.15 The pressure distribution around the NACA0012 aerofoil at $\alpha = 5^\circ$

shows the pressure distribution if the effects of the boundary layer are ignored and the dashed line shows the actual distribution.

The effect of the boundary layer is to modify the pressure distribution at the rear of the aerofoil such that lower pressure occurs there rather than if the boundary is ignored. The modified pressure distribution gives rise to pressure drag that is added to the skin friction drag, also caused by the boundary layer.

A3.7 The stalled aerofoil

If the angle of attack exceeds a certain critical value (10° – 16° , depending on the Re), separation of the boundary layer on the upper surface takes place, (Figure A3.17). This causes a wake to

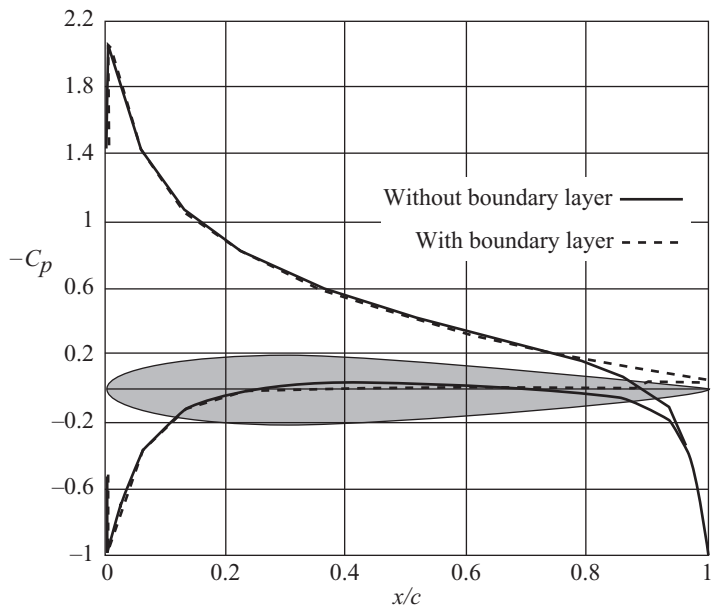


Figure A3.16 The pressure distribution around the NACA0012 aerofoil at $\alpha = 5^\circ$

form from above the aerofoil, reduces the circulation, reduces the lift and increases the drag. The flow past the aerofoil has then stalled. A flat plate will also develop circulation and lift but will stall at a very low angle of attack because of the sharp leading edge. Arching the plate will improve the stalling behaviour but a much greater improvement can be obtained by giving thickness to the aerofoil together with a well rounded leading edge.

A3.8 The lift coefficient

The lift coefficient is defined as

$$C_L = \frac{\text{Lift}}{\frac{1}{2}\rho U^2 A} \quad (\text{A3.8})$$

U is the flow speed and A is the plan area of the body. For a long body, such as an aircraft wing or a wind turbine blade, the lift per unit span is used in the definition and the plan area is replaced by the chord length.

$$C_l = \frac{\text{Lift/unit span}}{\frac{1}{2}\rho U^2 c} = \frac{\rho(\Gamma \times U)}{\frac{1}{2}\rho U^2 c} = \frac{\rho\pi U c \sin \alpha U}{\frac{1}{2}\rho U^2 c} = 2\pi \sin \alpha \quad (\text{A3.9})$$

In practice,

$$C_l = a_0 \sin \alpha \quad (\text{A3.10})$$

where a_0 , called the lift-curve slope $dC_l/d\alpha$, is about 5.73 (0.1/deg.), rather than 2π . Note that a_0 should not be confused with the flow induction factor.

Lift, therefore, depends on two parameters, the angle of attack α and the flow speed U . The same lift force can be generated by different combinations of α and U .

The variation of C_l with the angle of attack α is shown in Figure A3.18 for a typical *symmetrical* aerofoil (NACA0012). Notice that the simple relationship of Equation A3.10 is only valid for the pre-stall region, where the flow is attached, and because the angle of attack is small ($<16^\circ$) the equation is often written as

$$C_l = a_0 \alpha \quad (\text{A3.10a})$$

A3.9 Aerofoil drag characteristics

The definition of the drag coefficient for an aircraft wing or a wind turbine blade is based not on the frontal area but on the plan area, for reasons that will become clear later. The flow past a body which has a large span normal to the flow direction is basically two-dimensional and

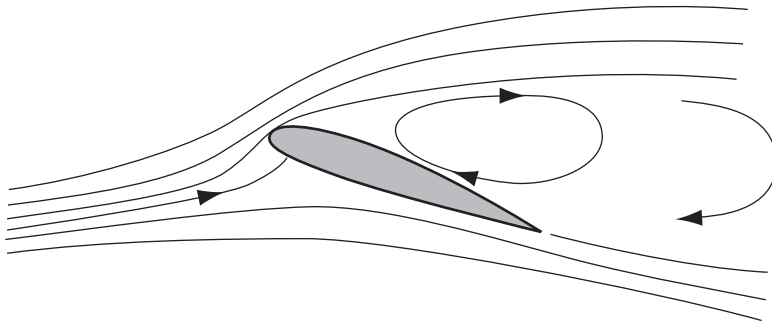


Figure A3.17 Stalled flow around an aerofoil

in such cases the drag coefficient can be based upon the drag force per unit span using the stream-wise chord length for the definition.

$$C_d = \frac{\text{Drag/unit span}}{\frac{1}{2}\rho U^2 c} \tag{A3.11}$$

For a wing of large span the value of C_d is roughly 0.01, at moderate Reynolds numbers. The drag coefficient of an aerofoil also varies with angle of attack. Figure A3.16 shows that on the upper surface pressure is rising as the flow moves towards the trailing edge, this is called an adverse pressure gradient and seeks to slow the air down. If the air is slowed to a standstill stall will occur and the pressure drag will rise sharply. The strength of the adverse pressure gradient increases with angle of attack and so it can be expected that the drag will rise with angle of attack. Figure A3.19 shows the variation of C_d with α also for the symmetrical NACA0012 aerofoil.

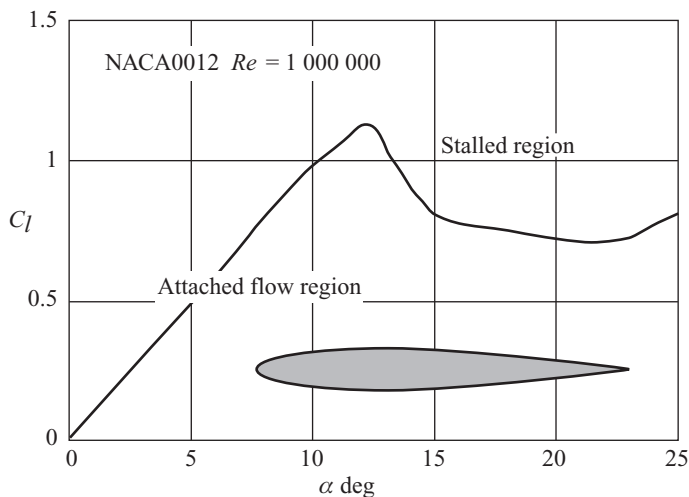


Figure A3.18 $C_l - \alpha$ curve for a symmetrical aerofoil

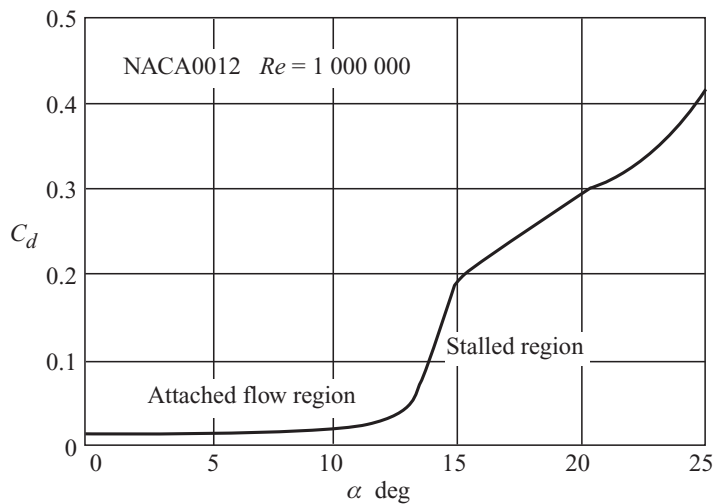


Figure A3.19 Variation of C_d with α for the NACA0012 aerofoil

The lift/drag ratio (shown in Figure A3.20) has a significant effect upon the efficiency of a wind turbine and it is desirable that a turbine blade operates at the maximum ratio.

The nature of the flow pattern around an aerofoil is determined by the Reynolds number and this significantly affects the values of the lift and drag coefficients. The general level of the drag coefficient increases with decreasing Reynolds number and below a critical Reynolds number of about 200 000 the boundary layer remains laminar causing a sharp rise in the coefficient. The effect on the lift coefficient is largely concerned with the angle of attack at which stall occurs. As the Reynolds number rises so does the stall angle and, because the lift

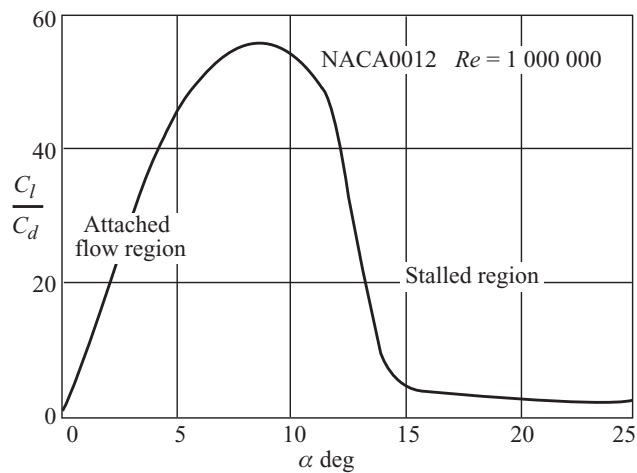


Figure A3.20 Lift/drag ratio variation for the NACA0012 aerofoil

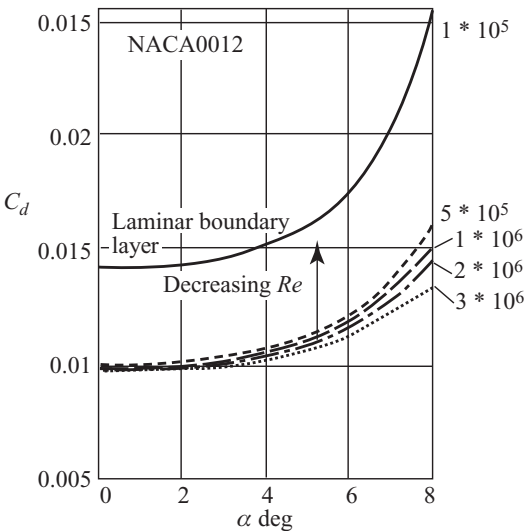


Figure A3.21 Variation of the drag coefficient with Reynolds number at low angles of attack

coefficient increases linearly with angle of attack below the stall, the maximum value of the lift coefficient also rises.

Characteristics for the NACA0012 aerofoil are shown in Figures A3.21 and A3.22.

A3.10 Cambered aerofoils

Cambered aerofoils, such as the NACA4412, shown in Figure A3.23, have curved chord lines and this allows them to produce lift at zero angle of attack.

Generally, cambered aerofoils have higher maximum lift/drag ratios than symmetrical aerofoils for positive angles of attack and this is the reason for their use.

The classification of the NACA 4-digit range of aerofoils, which were commonly used on wind turbines, is very simple and is illustrated in Figure A3.24: from left to right, the first digit represents the amount of camber as a percentage of the chord length, the second digit represents the percentage chord position, in units of 10%, at which the maximum camber occurs and the last two digits are the maximum thickness to chord ratio, as a percentage of the chord length, which, in this family of aerofoils, is at the 30% chord position. The cambered chord line, now called the camber line, comprises two parabolic arcs that join smoothly at the point of maximum camber. For other ranges of aerofoils the reader should refer to *Theory of Wing Sections* by Abbott and von Doenhoff (1959).

The angle of attack α is measured from the chord line which is now defined as the straight line joining the ends of the camber line.

Note that the lift at zero angle of attack is no longer zero; zero lift occurs at a small negative angle of attack. With most cambered aerofoils the zero lift line is approximately at $-A^\circ$ where A is the percentage camber.

The behaviour of the NACA4412 aerofoil is shown in Figure A3.25 for angles of attack below and just above the stall. Note that the lift at zero angle of attack is no longer zero; zero lift occurs at a small negative angle of attack of approximately 4° .

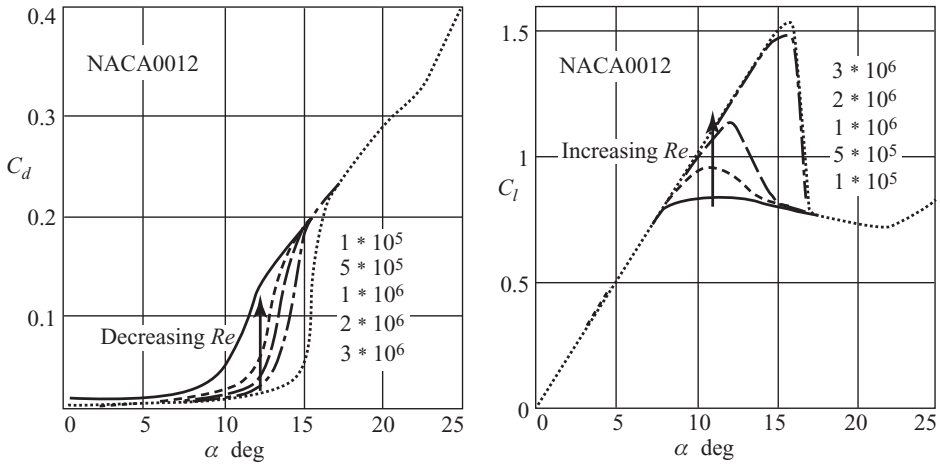


Figure A3.22 Variation of the drag and lift coefficients with Reynolds number in the stall region

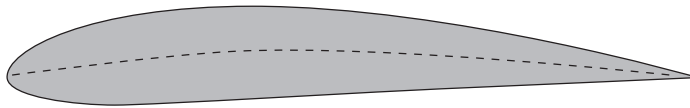


Figure A3.23 The profile of the NACA4412 aerofoil

The centre of pressure, which is at the $\frac{1}{4}$ chord position on symmetrical aerofoils lies aft of the $\frac{1}{4}$ chord position on cambered aerofoils and moves towards the trailing edge with increasing angle of attack. However, if a fixed chordwise position is chosen then the resultant force through that point is accompanied by a pitching moment (nose-up positive, by convention). If a pitching moment coefficient is defined as

$$C_m = \frac{\text{Pitching/unit span}}{\frac{1}{2}\rho U^2 c} \quad (\text{A3.12})$$

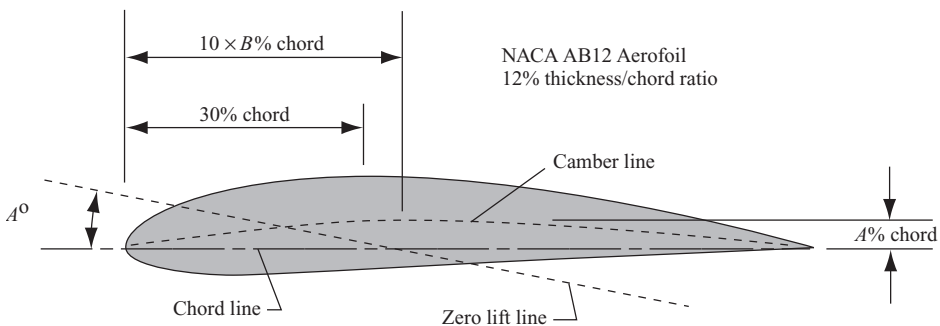


Figure A3.24 Classification of the NACAXXXX aerofoil range

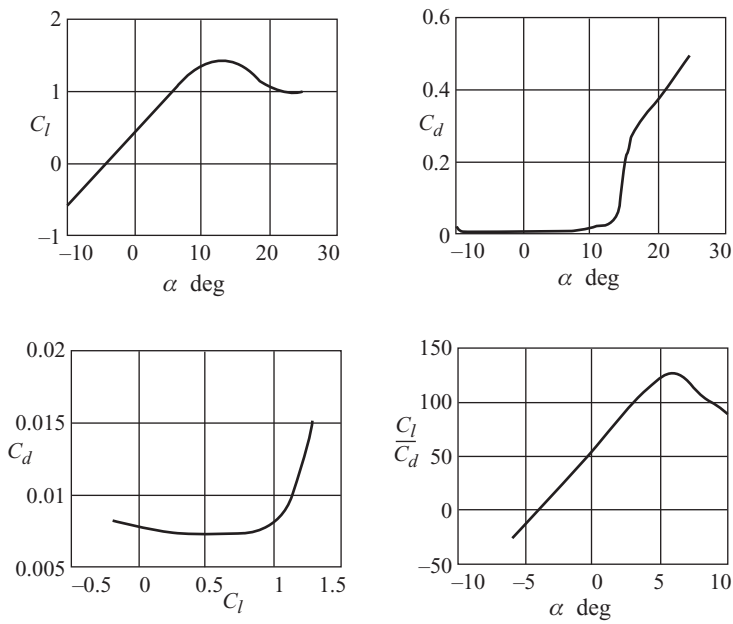


Figure A3.25 The Characteristics of the NACA4412 Aerofoil for $Re = 1.5 \cdot 10^6$

then there will be a position, called the ‘aerodynamic centre’, for which $dC_m/dC_l = 0$. Theoretically, the aerodynamic centre lies at the $\frac{1}{4}$ chord position and is close to this point for most practical aerofoils.

The value of C_m depends upon the degree of camber but for the NACA4412 the value is -0.1 , note that pitching moments are always negative in practice (nose down) despite the sign convention.

Above the stall there is no aerodynamic centre, as defined, and so the pre-stall position continues to be used to determine the pitching moment coefficient, which then becomes dependent upon α .

Further aerodynamic topics for wind turbines

4.1 Introduction

Chapter 3 deals with the aerodynamic behaviour of wind turbines in steady wind conditions and with the rotor aligned with the wind direction but, of course, real wind conditions are never steady and a turbine will always operate with some degree of yaw.

4.2 The aerodynamics of turbines in steady yaw

The rotor axis of a wind turbine rotor is usually not aligned with the wind because the wind is continuously changing direction: the rotor is not capable of following this variability and so spends most of its time in a yawed condition (Figure 4.1). The yawed rotor is less efficient than the non-yawed rotor and so it is vital to assess the efficiency for purposes of energy production estimation.

In the yawed condition, even in a steady wind, the angle of attack on each blade is continuously changing as it rotates and so the loads on the rotor blades are fluctuating, causing fatigue damage.

The changes in angle of attack mean that the blade forces cause not only a thrust in the axial direction but also moments about the yaw (z) axis and the tilt axis.

Even if the rotor is operating with a uniform induced velocity over the rotor disc when aligned with a steady wind, once the rotor is mis-aligned the induced velocity varies both azimuthally and radially which makes its determination much more difficult.

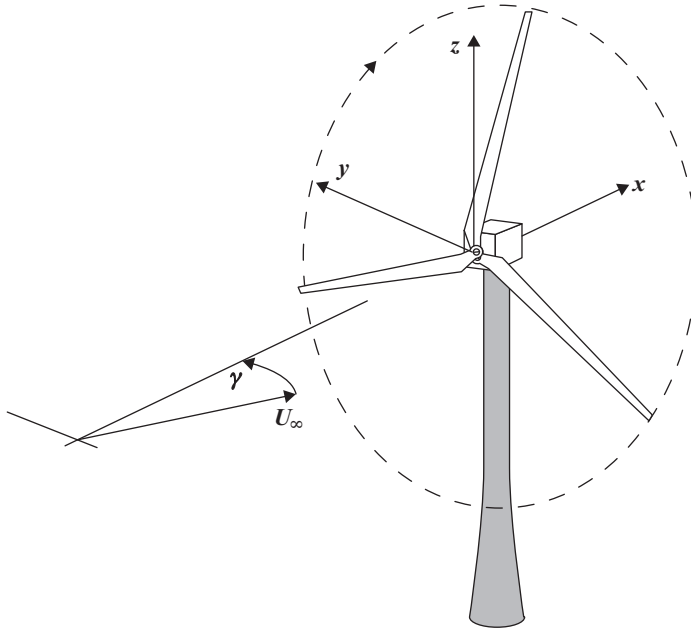


Figure 4.1 A wind turbine yawed to the wind direction

4.2.1 Momentum theory for a turbine rotor in steady yaw

The application of the momentum theory to an actuator disc representing a yawed rotor is somewhat problematical. The momentum theory is only capable of determining an average induced velocity for the whole rotor disc but, although in the non-yawed case the restriction was relaxed to allow some radial variation, it would not be appropriate to do this in the yawed case because the blade circulation is also changing with azimuth position. If it is assumed that the force on the rotor disc, which is a pressure force and so normal to the disc, is responsible for the rate of change of momentum of the flow then the average induced velocity must also be in a direction at right angles to the disc plane, that is, in the *axial* direction. The wake is therefore deflected to one side because a component of the induced velocity is at right angles to the wind direction. As in the non-yawed case the average induced velocity at the disc is half that in the wake.

Let the rotor axis be held at an angle of yaw γ to the steady wind direction (Figure 4.2) then, assuming that the rate of change of momentum in the axial direction is equal to the mass flow rate through the rotor disc times the change in velocity normal to the plane of the rotor

$$T = \rho A_D U_\infty (\cos \gamma - a) 2a U_\infty \quad (4.1)$$

Therefore, the thrust coefficient is

$$C_T = 4a(\cos \gamma - a) \quad (4.2)$$

and the power developed is

$$T U_\infty (\cos \gamma - a), \text{ giving } C_P = 4a(\cos \gamma - a)^2 \quad (4.3)$$

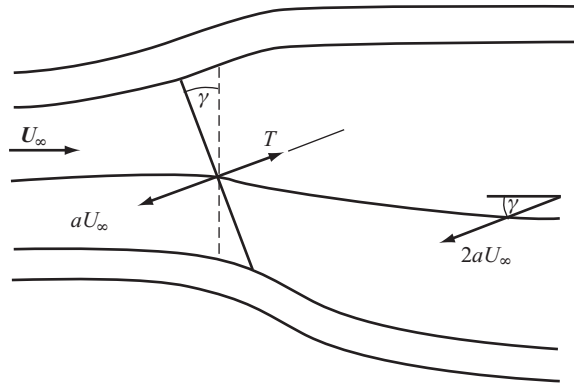


Figure 4.2 Deflected wake of a yawed turbine and induced velocities

To find the maximum value of C_P differentiate Equation 4.3 with respect a and set equal to zero, whence

$$a = \frac{\cos \gamma}{3} \quad \text{and} \quad C_{P \max} = \frac{16}{27} \cos^3 \gamma \quad (4.4)$$

This $\cos^3 \gamma$ rule is commonly adopted for power assessment in yawed flow.

Figure 4.3 shows decrease in power as the yaw angle increases.

A question remains: is it legitimate to apply the momentum theory in the above manner to the yawed rotor? Transverse pressure gradients which cause the wake to skew sideways may well also contribute to the net force on the flow in the axial direction, influencing the axial induced velocity. The above analysis might be satisfactory for determining the average axial induced velocity but there is even less justification to apply the momentum theory to each blade element position than there is in the non-yawed case. If a theory is going to be of any use in design it must be capable of determining the induced velocity at each blade element

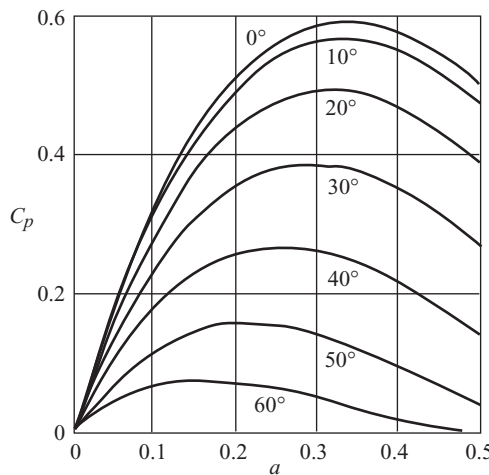


Figure 4.3 Power coefficient variation with yaw angle and axial flow factor

position to a satisfactory accuracy. The satisfactory calculation of blade forces is as important as the estimation of power.

4.2.2 Glauert’s momentum theory for the yawed rotor

Glauert (1926) was primarily interested in the autogyro, which is an aircraft with a freely windmilling rotor to provide lift and a conventional propeller to provide forward thrust. The lifting rotor has a rotational axis which inclines backwards from the vertical and by the virtue of the forward speed of the aircraft air flows through the rotor disc causing it to rotate and to provide an upward thrust. Thus, the autogyro rotor is just like a wind turbine rotor in yaw, when in forward flight. At high forward speeds the yaw angle is large but in a power-off vertical descent the yaw angle is zero.

Glauert maintained that at high forward speed the rotor disc, which is operating at a high tip speed ratio, is like a wing of circular plan-form at a small angle of attack (large yaw angle) and so the thrust on the disc is the lift on the circular wing. Simple lifting line wing theory (see Prandtl and Tiejens, 1957) states that the down-wash (induced velocity) at the wing, caused by the trailing vortex system, is uniform over the wing span (transverse diameter of the disc) for a wing with an elliptical plan-form and this would include the circular plan-form of the autogyro rotor.

The theory gives the uniform (average) induced velocity as

$$u = \frac{2L}{\pi(2R)^2\rho V} \tag{4.5}$$

where L is the lift and V is the forward speed of the aircraft.

The lift is in a direction normal to the effective incident velocity W , see Figure 4.4, and so is not vertical but leans backwards. The vertical component of the lift supports the weight of the aircraft and the horizontal component constitutes drag. In horizontal flight the vertical component of the lift doesn’t do any work but the drag does.

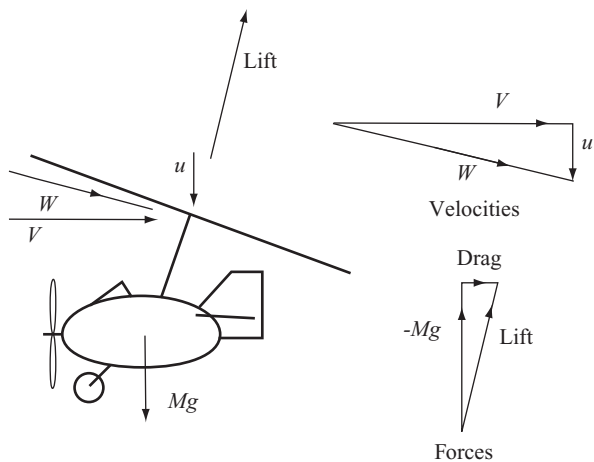


Figure 4.4 Velocities and forces on an autogyro in fast forward flight

The vector triangles of Figure 4.4 show that

$$\frac{D}{L} = \frac{u}{V} \quad (4.6)$$

In the wake of the aircraft the induced velocity u_w caused by the trailing vortices is greater than that at the rotor. A certain mass flow rate of air, $\rho V S$, where S is an area, normal to velocity V , yet to be determined, undergoes a downward change in velocity u_w in the far wake. By the momentum theory the rate of change of downward momentum is equal to the lift, therefore,

$$L = \rho V S u_w \quad (4.7)$$

The rate of work done by the drag DV must be equal to the rate at which kinetic energy is created in the wake $\frac{1}{2}\rho u_w^2 V S$, because the ambient static pressure in the wake of the aircraft is the same as the pressure ahead of the aircraft.

$$DV = \frac{1}{2}\rho u_w^2 V S \quad (4.8)$$

Combining Equations 4.6, 4.7 and 4.8 gives

$$D = \frac{L^2}{2\rho V^2 S} \quad (4.9)$$

and

$$u_w = 2u \quad (4.10)$$

Equation 4.10 should look familiar. Combining Equation 4.5, the lifting line theory's assessment of the induced velocity at the rotor, with Equation 4.6 gives

$$D = \frac{2L^2}{\rho V^2 \pi (2R)^2} \quad (4.11)$$

Comparing Equations 4.9 and 4.11 leads to an estimate of the required area S

$$S = \pi R^2 \quad (4.12)$$

S has the same area as the rotor disc but is normal to the flight direction.

Note that the above analysis has been simplified by assuming that the angle of attack is small. Actually, the trailing vortices from the rotor are influenced by their own induced velocity and so trail downwards behind the rotor. The induced velocity must, therefore, have a forward component, which means that the air undergoes a rate of change of momentum in the forward direction as well, thus balancing the drag. The drag is termed induced drag as it comes about by the backward tilting of the lift force caused by the induced velocity and has nothing to do with viscosity, it is entirely a pressure drag. Equation 4.5 should also be modified to replace V by W , the resultant velocity at the disc, and the area S will be in a plane normal to W . Also W has a direction which lies close to the plane of the rotor and so the lift force L will be almost the same as the thrust force T , which is normal to the plane

of the rotor, and by the same argument the induced velocity is almost normal to the plane of the rotor

$$u = \frac{2T}{\pi(2R)^2\rho W} \quad (4.13)$$

It can be assumed that a wind turbine rotor at high angles of yaw behaves just like the autogyro rotor.

At zero yaw the thrust force on the wind turbine rotor disc, given by the momentum theory, is

$$T = \pi R^2 \frac{1}{2} \rho 4u(U_\infty - u) \quad (4.14)$$

where U_∞ now replaces V .

So the induced velocity is

$$u = \frac{2T}{\pi(2R)^2\rho(U_\infty - u)} \quad (4.15)$$

The area S now coincides in position with the rotor disc.

Putting $W = U_\infty - u$ to represent the resultant velocity of the flow at the disc in Equation 4.15 then gives exactly the same Equation as 4.13 which is for a large angle of yaw. On the basis of this argument Glauert assumed that Equation 4.13, which is the simple momentum theory, could be applied at all angles of yaw, area $S = \pi R^2$, through which the mass flow rate is determined, always lying in a plane normal to the resultant velocity. The rotation of the area S is a crucially different assumption to that of the theory of Section 4.1.1 (which will now be referred as the axial momentum theory) and allows for part of the thrust force to be attributable to an overall lift on the rotor disc.

Thus,

$$T = \rho \pi R^2 W 2u \quad (4.16)$$

where

$$W = \sqrt{U_\infty^2 \sin^2 \gamma + (U_\infty \cos \gamma - u)^2} \quad (4.17)$$

Thrust is equal to the mass flow rate times the change in velocity in the direction of the thrust. Both T and u are assumed to be normal to the plane of the disc.

Noting that $u = aU_\infty$, the thrust coefficient is then

$$C_T = 4a\sqrt{1 - a(2\cos\gamma - a)} \quad (4.18)$$

The power developed is a scalar quantity and so is the scalar product of the thrust force and the resultant velocity at the disc W . Hence, the power coefficient is

$$C_P = 4a(\cos\gamma - a)\sqrt{1 - a(2\cos\gamma - a)} \quad (4.19)$$

However, as some of the thrust is attributable to lift on the rotor disc acting as a circular wing that lift will not extract power from the wind because the net velocity field associated with the lift does not give rise to a flow through the rotor disc. Only that proportion of the thrust which arises from net flow through the disc will extract energy from the flow. Consequently,

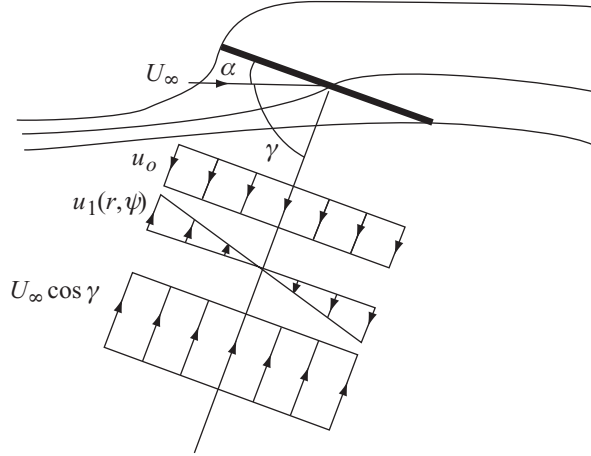


Figure 4.5 Velocities normal to the yawed rotor

the axial momentum theory is more likely to estimate the power extraction correctly, whereas the Glauert theory is more likely to estimate the thrust correctly.

One very useful concept that emerges from Glauert's autogyro theory is his prediction that the induced velocity through the rotor would not be uniform. The flow through the yawed rotor is depicted in Figure 4.5 and a simplification of the contributions to the velocity normal to the plane of the rotor along the rotor diameter parallel to the flight direction are shown. The mean induced velocity through the rotor, as determined by Equation 4.13, is shown as u_0 , the normal component of the forward velocity of the aircraft is $U_\infty \cos \gamma$, also uniform over the disc, but, to account for the flow pattern shown, there needs to be a non-uniform component which decreases the normal induced velocity at the leading edge of the rotor disc and increases it at the rear. From symmetry, the induced velocity along the disc diameter normal to the flight direction (normal to the plane of the diagram) is uniform. The simplest form of the non-uniform component of induced velocity would be

$$u_1(r, \psi) = u_1 \frac{r}{R} \sin \psi \quad (4.20)$$

where ψ is the blade azimuth angle measured in the direction of rotation, 0° being when the blade is normal to the flight direction (or when the wind turbine blade is vertically upwards), and u_1 is the amplitude of the non-uniform component which is dependent on the yaw angle. There would, of course need to be induced velocities parallel to the plane of the rotor disc but these are of secondary importance; the normal induced velocity has a much greater influence on the blade angle of attack than the in-plane component and, therefore, a much greater influence on blade element forces.

The value of u_1 in Equation 4.20 cannot be determined from momentum theory but Glauert suggested that it would be of the same order of magnitude as u_0 . The total induced velocity, normal to the rotor plane, may then be written as

$$u = u_0 \left(1 + K \frac{r}{R} \sin \psi \right) \quad (4.21)$$

The value of K must depend upon the yaw angle.

4.2.3 Vortex cylinder model of the yawed actuator disc

The vortex theory for the non-yawed rotor given in Section 3.4 is demonstrated to be equivalent to the momentum theory in its main results but, in addition, was shown to give much more detail about the flow-field. As the momentum theories of Sections 4.2.1 and 4.2.2 yield very limited results, using the vortex approach for the yawed rotor may also prove to be useful, giving more flow structure detail than the momentum theory and, perhaps, a means of allying it with the blade element theory.

The wake of a yawed rotor is skewed to one side because the thrust T on the disc is normal to the disc plane and so has a component normal to the flow direction. The force on the flow, therefore, is in the opposite sense to T causing the flow to accelerate both upwind and sideways. The centre line of the wake will be at an angle χ to the axis of rotation (axis normal to the disc plane) known as the wake skew angle. The skew angle will be greater than the yaw angle. The same basic theory as in Section 3.4 can be carried out for an actuator disc with a wake skewed to the rotor axis by an angle χ . There is an important proviso however; it must be assumed that the bound circulation on the rotor disc is radially and azimuthally uniform. As will be demonstrated, the angle of attack of the blades is changing cyclically and so it would be impossible for the uniform circulation condition ever to be valid. What must be assumed is that the variation of circulation around a mean value has but a small effect on the induced velocity and the wake is therefore dominated by the vorticity shed from the blade tips by the mean value of circulation (see Figure 4.6).

The expansion of the wake again imposes a difficulty for analysis and so, as before, it will be ignored (Figure 4.7).

The analysis of the yawed rotor was first carried out for purposes of understanding a helicopter rotor in forward flight by Coleman *et al.* (1945) but it can readily be applied to a wind turbine rotor by reversing the signs of the circulation and the induced velocities. An infinite number of blades is assumed as in the analysis of Section 3.4. The vorticity g_ψ has a direction which remains parallel to the yawed disc and assuming it to be uniform (not varying with the azimuth angle), using the Biot-Savart law, induces a *time average* velocity at the disc of $aU_\infty \sec(\chi/2)$ in a direction which bisects the skew angle, as shown in Figure 4.8. The average axial induced velocity, normal to the rotor plane, is aU_∞ , as in the non-yawed case. In the fully developed wake the induced velocity is twice that at the rotor disc.

Because the average induced velocity at the disc is not in the rotor's axial direction, as is assumed for the momentum theory of Sections 4.1.1 and 4.1.2, the force T on the disc, which

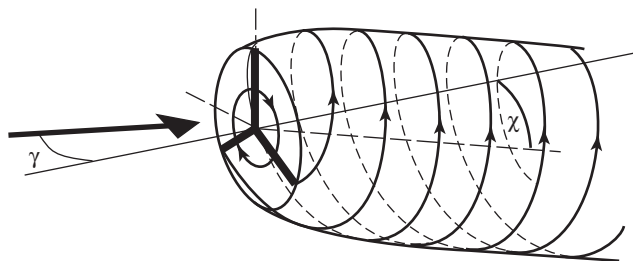


Figure 4.6 The deflected vortex wake of a yawed rotor showing the shed vortices of three blades

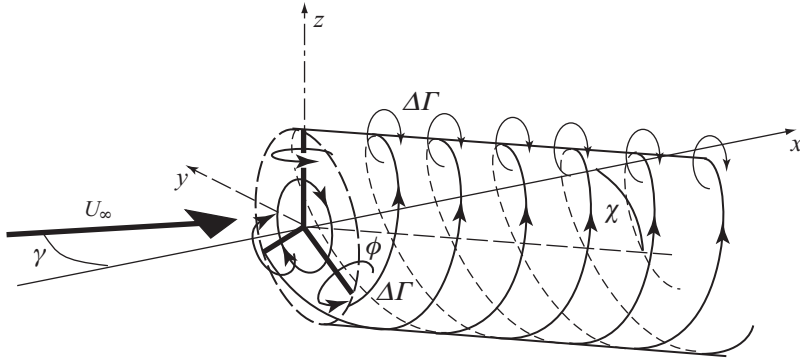


Figure 4.7 A yawed rotor wake without wake expansion

must be in the axial direction, cannot be solely responsible for the overall rate of change of momentum of the flow; there is a change of momentum in a direction normal to the rotor axis.

The velocity components at the rotor disc define the skew angle:

$$\tan \chi = \frac{U_\infty \left(\sin \gamma - a \cdot \tan \frac{\chi}{2} \right)}{U_\infty (\cos \gamma - a)} = \frac{2 \tan \frac{\chi}{2}}{1 - \tan^2 \frac{\chi}{2}} \quad (4.22)$$

From which it can be shown that a close, approximate relationship between χ , γ and a is

$$\chi = (0.6a + 1)\gamma \quad (4.23)$$

Using the velocities shown in Figure 4.9 a fresh analysis can be made of the flow. The average force on the disc can be determined by applying Bernoulli's Equation to both the upwind and downwind regions of the flow:

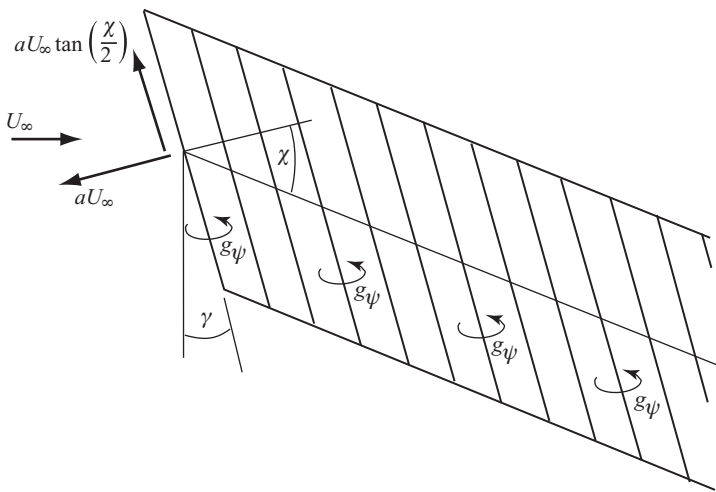


Figure 4.8 A yawed actuator disc and the skewed vortex cylinder wake

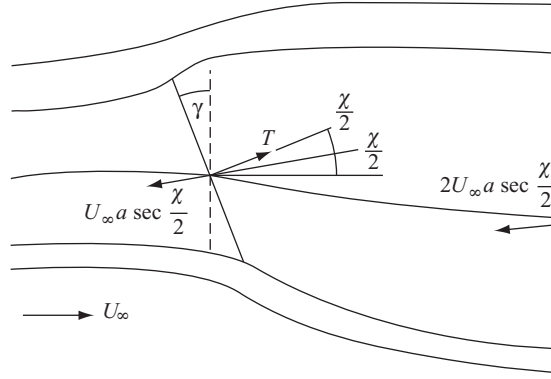


Figure 4.9 Average induced velocities caused by a yawed actuator disc

$$\text{Upwind } p_{\infty} + \frac{1}{2}\rho U_{\infty}^2 = p_D^+ + \frac{1}{2}\rho U_D^2$$

$$\text{Downwind } p_D^- + \frac{1}{2}\rho U_D^2 = p_{\infty} + \frac{1}{2}\rho U_{\infty}^2 \left[(\cos \gamma - 2a)^2 + \left(\sin \gamma - 2a \tan \frac{\chi}{2} \right)^2 \right]$$

where U_D is the resultant velocity at the disc.

Subtracting the two Equations to obtain the pressure drop across the disc

$$p_D^+ - p_D^- = \frac{1}{2}\rho U_{\infty}^2 4a \left(\cos \gamma + \tan \frac{\chi}{2} \sin \gamma - a \sec^2 \frac{\chi}{2} \right)$$

The coefficient of thrust on the disc is therefore,

$$C_T = 4a \left(\cos \gamma + \tan \frac{\chi}{2} \sin \gamma - a \sec^2 \frac{\chi}{2} \right) \quad (4.24)$$

and the power coefficient is

$$C_P = 4a(\cos \gamma - a) \left(\cos \gamma + \tan \frac{\chi}{2} \sin \gamma - a \sec^2 \frac{\chi}{2} \right) \quad (4.25)$$

In a similar manner to the Glauert theory, it is not clear how much of the thrust in Equation 4.24 is capable of extracting energy from the flow and so the expression for power in Equation 4.25 will probably be an over estimate. A comparison of the maximum C_P values derived from the three theories, as a function of the yaw angle, is shown in Figure 4.10.

4.2.4 Flow expansion

The induced velocity component parallel to the skewed axis of the wake is uniform over the disc with a value aU_{∞} , as can be deduced from Figure 4.9. The induced horizontal velocity, normal to the skewed axis of the wake is also uniform over the area of the disc with a value $aU_{\infty} \tan(\chi/2)$.

In addition to the uniform induced velocities of Figure 4.9 the expansion of the flow gives rise to velocities in the y and z directions, respectively (i.e. directions in a vertical plane at

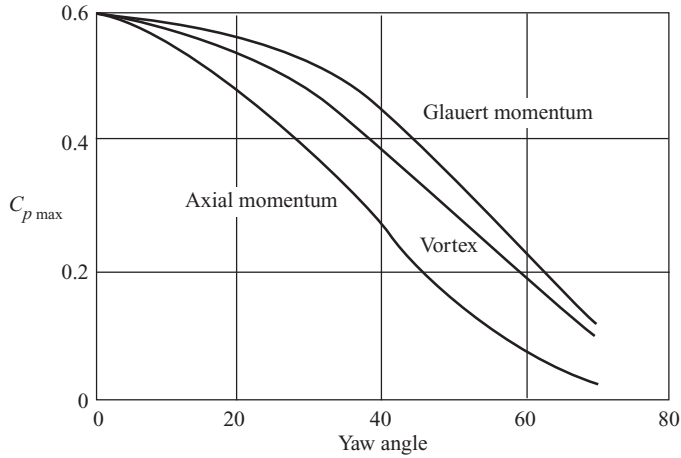


Figure 4.10 Maximum power coefficient variation with yaw angle, comparison of momentum and vortex theories

the skew angle χ to the rotor plane, see Figure 4.11). When resolved into the rotor plane the flow expansion velocities will give rise to a normal induced velocity of the type predicted by Glauert in Equation 4.21. Note that the positive direction for u (and u'') is now defined as *downwind* (see Figure 4.11), whereas it was upwind in Section 4.2.2.

At a point on the disc at radius r and azimuth angle ψ , defined in Figure 4.11, the induced flow expansion velocities are non-simple functions of r and ψ . Across the horizontal

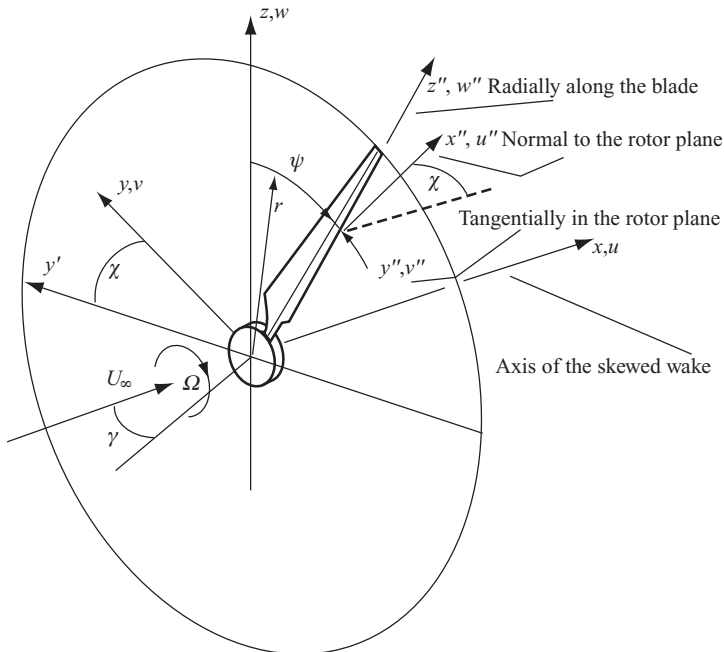


Figure 4.11 Axis system for a yawed rotor

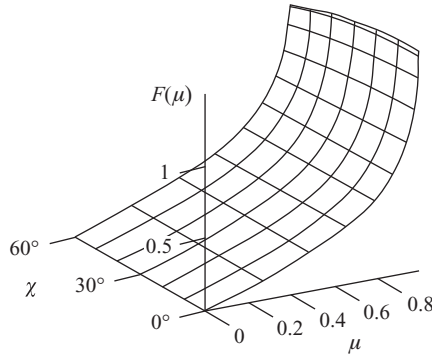


Figure 4.12 Flow expansion function variation with radial position and skew angle

diameter, where $\psi = \pm 90^\circ$, Coleman *et al.* obtained an analytical solution for the flow expansion velocity in the y direction that involves complete elliptic integrals: the solution is not very practicable because numerical evaluation requires calculating the difference between two large numbers. Simplification of the analytical solution leads to the following expression for the horizontal flow expansion velocity which removes the evaluation difficulty but it is not in closed form:

$$v(\chi, \psi, \mu) = \frac{-2aU_\infty\mu \sin \psi}{\pi} \int_0^{\frac{\pi}{2}} \frac{\sin^2 2\varepsilon}{\sqrt{(1+\mu)^2 - 4\mu \sin^2 \varepsilon}} \frac{1}{(\mu + \cos 2\varepsilon)^2 \cos^2 \chi + \sin^2 2\varepsilon} d\varepsilon \quad (4.26)$$

where $\mu = r/R$, ε is a parameter arising from the elliptic integrals, which is eliminated from the function by the definite integral, and aU_∞ is the average induced velocity as previously defined. An important feature of Equation 4.26 is that the flow expansion velocity is proportional to the average axial flow induction factor. Furthermore, if Equation 4.26 is divided by $\sec^2(\chi/2)$ the result is almost independent of the skew angle χ and is as shown in Figure 4.12, which clearly demonstrates how little it changes over a range of skew angle of 0° – 60° .

At all skew angles the value of the flow expansion function is infinite at the edge of the rotor disc, indicating a singularity in the flow which, of course, does not occur in practice but is a result of assuming uniform blade circulation. Circulation must fall to zero at the disc edge in a smooth fashion.

No analytical expressions for the flow expansion velocity components for values of ψ other than $\pm 90^\circ$ were developed by Coleman *et al.* but numerical evaluations of the flow expansion velocities can be made using the Biot-Savart law.

The radial variation of the vertical flow expansion velocity across the vertical diameter of the rotor disc is much the same as $F(\mu)$ for skew angles between $\pm 45^\circ$ but outside of this range the vertical velocity increases more sharply than the horizontal velocity at the disc edge. As will be shown, the vertical expansion velocity is of less importance than the horizontal velocity in determining the aerodynamic behaviour of the yawed rotor.

The variation of the horizontal and vertical flow expansion velocities along radial lines on the rotor disc surface at varying azimuth angles (a radius sweeping out the disc surface as it

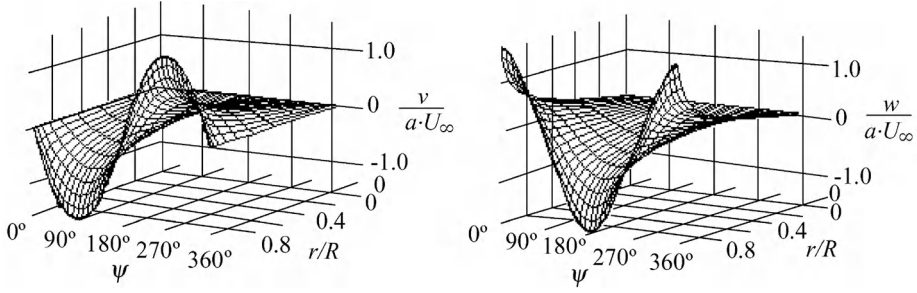


Figure 4.13 Azimuthal and radial variation of horizontal (v) and vertical (w) velocities on the rotor plane for a skew angle of 30°

rotates about the yawed rotor axis) shows that some further simplifications can be made for small skew angles.

Figure 4.13 shows the variation of the flow expansion velocities across the rotor disc for a skew angle of 30° . It should be emphasised that the velocity components lie in planes that are normal to the skewed axis of the wake. Inspection of the variations leads to simple approximations for the two velocity components.

$$v(\chi, \psi, \mu) = -aU_\infty F(\mu) \sin \psi \quad (4.27)$$

$$w(\chi, \psi, \mu) = aU_\infty F(\mu) \cos \psi \quad (4.28)$$

where

$$F(\mu) = \frac{2\mu}{\pi} \int_0^{\pi/2} \frac{\sin^2 2\varepsilon}{\sqrt{(1+\mu)^2 - 4\mu \sin^2 \varepsilon}} \frac{1}{(\mu + \cos 2\varepsilon)^2 \cos^2 \chi + \sin^2 2\varepsilon} d\varepsilon \quad (4.29)$$

The drawback of Equations 4.27 and 4.28 is the singularity in the flow expansion function Equation 4.29 at the outer edge of the disc. If the actuator disc is replaced with rotor which has a small number of blades then the flow expansion function changes very significantly. Conducting a calculation using the Biot-Savart law for a non-yawed, single bladed rotor represented by a lifting line vortex of radially uniform strength the flow expansion function can be determined numerically. It is found that the flow expansion velocity along the radial lifting line is a function of the helix (flow) angle of the discrete line vortex shed from the tip of the lifting line (blade). The vortex wake is assumed to be rigid in that the helix angle and the wake diameter are fixed everywhere at the values that pertain at the rotor. The solutions for a single blade rotor can be used to determine the flow fields for multi-blade rotors by a simple process of superposition. The resulting flow expansion functions $F(\mu)_N$ are depicted in Figure 4.14 for one-, two- and three-blade rotors.

The radial variations in Figure 4.14 have been extended beyond the rotor radius to show the continuity which exists for the discrete blade situation as compared with the singularity that occurs for the actuator disc. There are two striking features of the flow expansion functions of Figure 4.14: the function is heavily modified by the value of the helix angle ϕ_t , at which

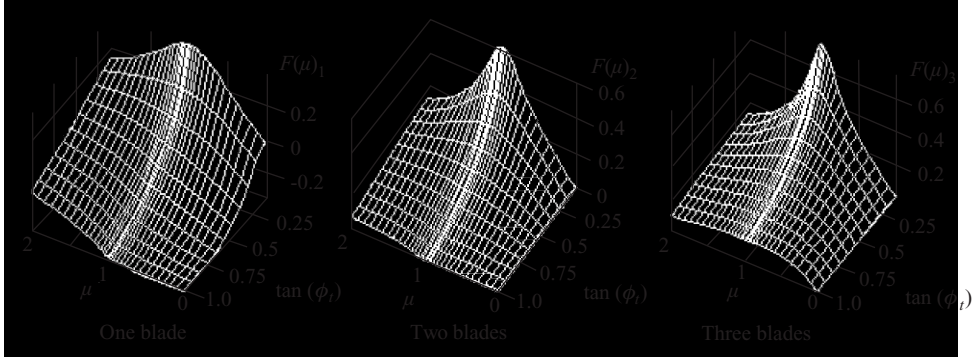


Figure 4.14 Flow expansion functions for one-, two- and three-blade rotors by lifting line theory

the tip vortex is shed from the blade tips, and the negative values (flow contraction) that can occur for the single-blade rotor.

An analytical expression that approximates the form of the diagrams shown in Figure 4.14 for two- and three-bladed rotors is

$$F_a(\mu, \phi_t, B) = \frac{F(\mu)}{\sqrt{1 + 50 \frac{\tan^2 \phi_t}{B^2} \left(\frac{1}{\tan \phi_t} + 8 \right) F(\mu)^2 [\mu(2 - \mu)F(\mu)]^{\tan \phi_t} \frac{0.05}{\tan \phi_t}}} \quad (4.30)$$

where $\tan \phi_t = (1 - a)/(\lambda(1 + a'))$ is the tangent of the flow angle. The results of Equation 4.30 are shown in Figure 4.15 for two and three blade rotors.

When transformed as components of velocity with respect to axes rotating about the rotor axis (x'' , y'' and z'' axes as shown in Figure 4.11), the flow expansion velocities of Equations 4.27 and 4.28 are resolved into the components that are normal and tangential to the blade element (see Figure 4.18).

The normal component is

$$u'' = -aU_\infty \left(1 + 2 \sin \psi \tan \frac{\chi}{2} F(\mu) \right) \quad (4.31)$$

and the tangential component is

$$v'' = aU_\infty \cos \psi \tan \frac{\chi}{2} \left(1 + 2 \sin \psi \tan \frac{\chi}{2} F(\mu) \right) \quad (4.32)$$

to which must be added the components of the wind velocity U_∞ the normal component

$$U'' = U_\infty \cos \gamma \quad (4.33)$$

and the tangential component

$$V'' = U_\infty \cos \psi \sin \gamma \quad (4.34)$$

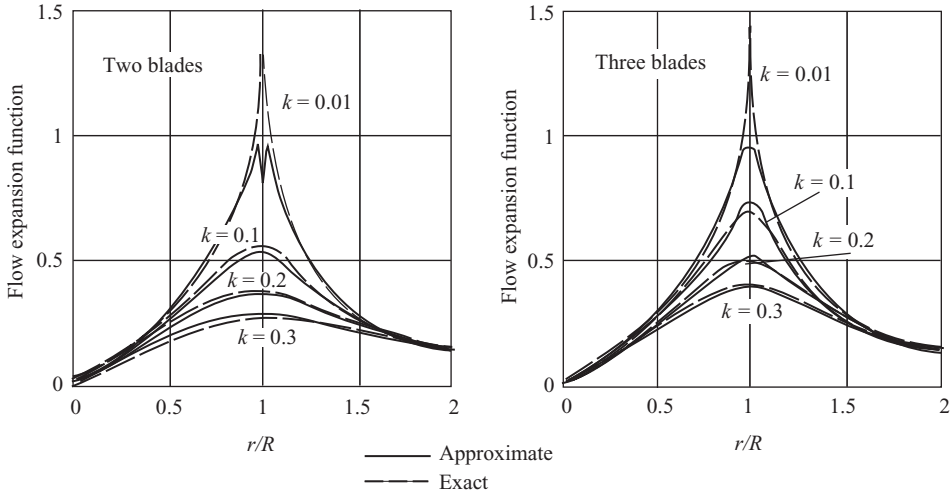


Figure 4.15 Approximate flow expansion functions for two- and three-blade rotors ($k = \tan \phi_t$)

There is a radial (span-wise) velocity component but this will not influence the angle of attack so can be ignored.

Clearly, from Equation 4.31, the Coleman theory determines the influence of skew angle (the factor K in Equation 4.21) as being

$$K(\chi) = 2 \tan \frac{\chi}{2} \quad (4.35)$$

In addition there is the tangential velocity Ωr caused by blade rotation and also the induced wake rotation, but the latter will be ignored initially.

The velocities of Equations 4.31–4.34 will produce a lower angle of attack when the azimuth angle ψ is positive, see Figure 4.16, than when it is negative and so the angle of attack will vary cyclically. When ψ is positive the incident flow velocity lies closer to the radial axis of the blade than when ψ is negative. The difference in angle of attack can be attributed to flow expansion as depicted in Figure 4.16.

The variation of the angle of attack makes the flow about a blade aerofoil unsteady and so the lift will have a response of the kind discussed in Section 4.4. The blade circulation will, therefore, vary during the course of a revolution, which means that the vortex model is incomplete because it is derived from the assumption that the circulation is constant.

There is clearly additional vorticity in the wake being shed from the blades' trailing edges influencing the induced velocity, which is not accounted for in the theory. The additional induced velocity would be cyclic so would probably not affect the average induced velocity normal to the rotor disc but would affect the amplitude and phasing of the angle of attack.

Further numerical analysis of the Coleman vortex theory reveals that at skew angles greater than $\pm 45^\circ$ higher harmonics than just the one per revolution term in Equation 4.21 become significant in the flow expansion induced velocities. Only odd harmonics are present, reflecting the anti-symmetry about the yaw axis.

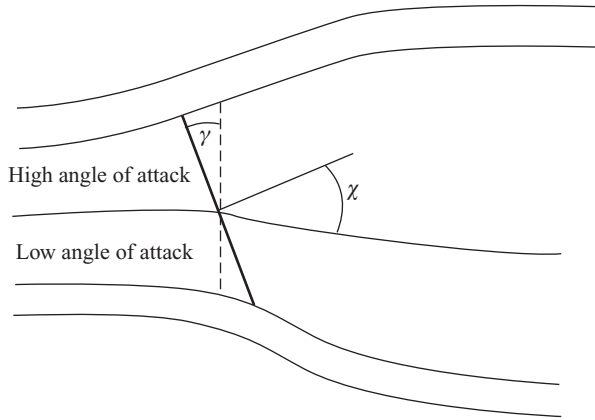


Figure 4.16 Flow expansion causes a differential angle of attack

4.2.5 Related theories

A number of refinements to the Glauert and Coleman theories have been proposed by other researchers: mostly addressing helicopter aerodynamics but some have been directed specifically at wind turbines. In particular, Øye (1992) undertook the same analysis as Coleman and proposed a simple curve-fit to Equation 4.29

$$F_{\phi}(\mu) = \frac{1}{2}(\mu + 0.4\mu^3 + 0.4\mu^5) \quad (4.36)$$

See Figure 4.17. Øye has clearly avoided the very large values that Equation 4.29 produces close to the outer edge of the disc and Equation 4.36 is in general accordance with the flow expansion functions shown in Figure 4.14.

Meijer Drees (1949) has extended the Coleman *et al.* vortex model to include a cosinusoidal variation of blade circulation. The main result is a modification to the function $K(\chi)$ but Meijer Drees retained Glauert's assumption of linear variation of normal induced velocity with radius

$$u'' = -aU_{\infty} \left[1 + \frac{4}{3}\mu \left(1 - 1.8 \left(\frac{\sin \gamma}{\lambda} \right)^2 \right) \sin \psi \tan \frac{\chi}{2} \right] \quad (4.37)$$

(see Schepers and Snel, 1995).

4.2.6 Wake rotation for a turbine rotor in steady yaw

Wake rotation is, of course, present in the wake flow but cannot be related only to the torque. The vortex theory needs also to include a root vortex, which will lie in the wake along the skewed wake axis. The rotation in the wake will, therefore, be about the skewed wake axis and not about the axis of rotation, and the wake rotation velocity will lie in a plane normal to the skewed wake axis.

To determine the wake rotation velocity the rate of change of angular momentum about skewed wake axis will be equated to the moment about the axis produced by blade forces.

If the wake rotation velocity is described, as before, in terms of the angular velocity of the rotor then

$$v''' = \Omega r''' a' h(\psi) \quad (4.38)$$

where the triple prime denotes an axis system rotating about the wake axis and $h(\psi)$ is a function which determines the intensity of the root vortex's influence. In the non-yawed case the root vortex induces a velocity at the rotor which is half of that it induces in the far wake at the same radial distance and the same would apply to a disc normal to the skewed axis with a centre located at the same position as the actual rotor disc. The distance upstream or downstream of a point on actual rotor disc from the plane of the disc normal to the wake axis determines the value of root vortex influence function $h(\psi)$. The value of $h(\psi)$ will lie between 0.0 and 2.0, being equal to 1.0 at points on the vertical diameter.

The velocity induced by a semi-infinite line vortex of strength Γ lying along the x-axis from zero to infinity at a point with cylindrical co-ordinates (x''', ψ''', r''') is, using the Biot-Savart law,

$$\vec{v}''' = \frac{\Gamma}{4\pi r'''} \begin{bmatrix} 0 \\ 1 + \frac{x'''}{\sqrt{x'''^2 + r'''^2}} \\ 0 \end{bmatrix} = \begin{bmatrix} 0 \\ v''' \\ 0 \end{bmatrix} \quad (4.39)$$

The induced velocity when $x''' = \infty$ is twice that when $x''' = 0$ and is zero when $x''' = -\infty$.

For a point on the rotor disc $(0, \psi, r)$ the corresponding co-ordinates (x''', ψ''', r''') in the wake cylindrical co-ordinate system are

$$\begin{aligned} x''' &= -y' \sin \chi = r \sin \psi''' \sin \chi, \quad r''' = r \sqrt{\cos^2 \psi + \cos^2 \chi \sin^2 \psi} \\ \text{and } \cos \psi''' &= \frac{r}{r'''} \cos \psi, \quad \sin \psi''' = \frac{r}{r'''} \sin \psi \cos \chi \end{aligned} \quad (4.40)$$

Hence, substituting $4\pi r'''^2 \Omega a'$ for the circulation (Equation 3.33) the induced velocity at the same point is

$$v''' = \Omega r''' a' \left(1 + \frac{x'''}{\sqrt{x'''^2 + r'''^2}} \right) \quad (4.41)$$

So, transforming the velocity of Equation 4.39 to the rotating axes in the plane of the rotor disc

$$\vec{v}'' = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos \psi & \sin \psi \\ 0 & -\sin \psi & \cos \psi \end{bmatrix} \begin{bmatrix} \cos \chi & \sin \chi & 0 \\ -\sin \chi & \cos \chi & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos \psi''' & -\sin \psi''' \\ 0 & \sin \psi''' & \cos \psi''' \end{bmatrix} \begin{bmatrix} 0 \\ v''' \\ 0 \end{bmatrix} \quad (4.42)$$

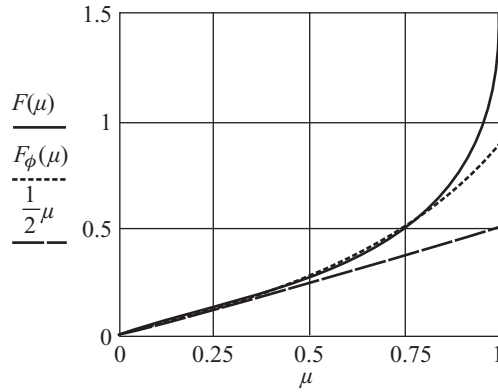


Figure 4.17 Øye's curve-fit to Coleman's flow expansion function

Substituting Equations 4.40 and 4.41 into Equation 4.42 gives

$$\vec{V}'' = \begin{bmatrix} \cos \psi \sin \chi \\ \cos \chi \\ 0 \end{bmatrix} \Omega r a' (1 + \sin \psi \sin \chi) \quad (4.43)$$

Thus, the wake rotation produces two velocity components, one in the rotor plane and one normal to the rotor plane; there is no radial component.

4.2.7 The blade element theory for a turbine rotor in steady yaw

There is doubt about the applicability of the blade element theory in the case of a yawed turbine because the flow, local to a blade element, is unsteady and because the theory representing the vortex half of the equation, which replaces the momentum theory, is incomplete in this respect. However, it is not clear how large or significant are the unsteady forces. In a steady yawed condition the flow velocities at a point on the rotor disc do not change with time, if an infinity of blades is assumed, and so there is no effect to consider. However, the change of angle of attack with time at a point on the blade does mean that the two dimensional lift force should really be modified by a lift deficiency function similar to that determined by Theodorsen (14) for the rectilinear wake of a sinusoidally pitching aerofoil.

Neglecting the effects of shed vorticity the net velocities in the plane of a local blade element are shown in Figure 4.18. The radial (span-wise) velocity component is not shown in Figure 4.18 but it is neglected as it is not considered to have any influence on the angle of attack and, therefore, on the lift force.

The flow angle ϕ is then determined by the components of velocity shown in Figure 4.18.

$\tan \phi$

$$= \frac{U_{\infty}(\cos \gamma - a(1 + F(\mu)K(\chi) \sin \psi)) + \Omega r a' \cos \psi \sin \chi (1 + \sin \psi \sin \chi)}{\Omega r (1 + a' \cos \chi (1 + \sin \psi \sin \chi)) + U_{\infty} \cos \psi \left(a \tan \frac{\chi}{2} (1 + F(\mu)K(\chi) \sin \psi) - \sin \gamma \right)} \quad (4.44)$$

where $\mu = r/R$ is measured radially from the axis of rotor rotation.

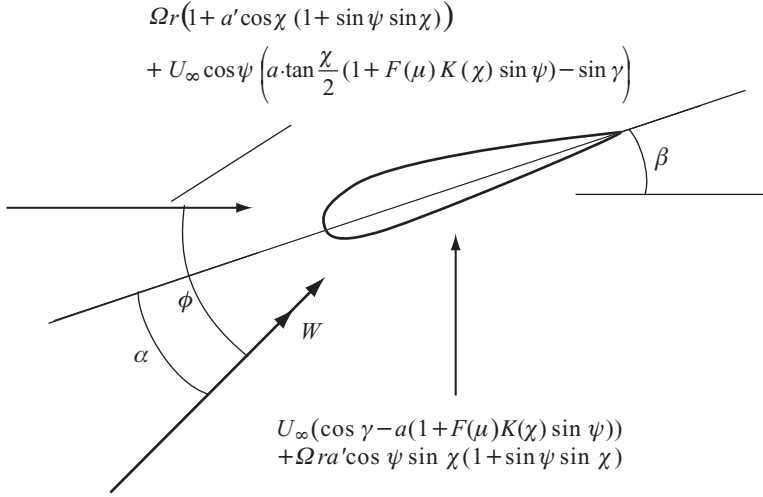


Figure 4.18 The velocity components in the plane of a blade cross-section

The angle of attack α is found from

$$\alpha = \phi - \beta \quad (4.45)$$

Lift and drag coefficients taken from two dimensional experimental data, just as for the non-yawed case, are determined from the angle of attack calculation for each blade element (each combination of μ and ψ).

4.2.8 The blade element – momentum theory for a rotor in steady yaw

The forces on a blade element can be determined via Equations 4.44 and 4.45 for given values of the flow induction factors.

The thrust force will be calculated using Equation 3.46 in Section 3.5.3, which is for a complete annular ring of radius r and radial thickness δr .

$$\delta T = \delta L \cos \phi + \delta D \sin \phi = \frac{1}{2} \rho W^2 B c (C_l \cos \phi + C_d \sin \phi) \delta r \quad (3.46)$$

For an elemental area of the annular ring swept out as the rotor turns through an angle $\delta \psi$ the proportion of the force is

$$\delta T_b = \frac{1}{2} \rho W^2 B c (C_l \cos \phi + C_d \sin \phi) \delta r \frac{\delta \psi}{2\pi}$$

then, putting $C_x = C_l \cos \phi + C_d \sin \phi$ and $\sigma_r = Bc/2\pi r$

$$\delta T_b = \frac{1}{2} \rho W^2 \sigma_r C_x r \delta r \delta \psi \quad (4.46)$$

The values of C_l and C_d should really include unsteady effects because of the ever changing blade circulation with azimuth angle which will depend upon the level of the reduced frequency of the circulation fluctuation.

If it is chosen to ignore drag, or use only that part of the drag attributable to pressure, then Equation 4.46 should be modified accordingly.

The rate of change of momentum will use either Equation 4.18, Glauert's theory, or Equation 4.24, the vortex cylinder theory: in both Equations the flow induction factor a should be replaced by af to account for Prandtl tip-loss.

For Glauert's theory,

$$\delta T_m = \frac{1}{2} \rho U_\infty^2 4af \sqrt{1 - af(2 \cos \gamma - af)} r \delta \psi \delta r \quad (4.47a)$$

Or, for the vortex theory,

$$\delta T_m = \frac{1}{2} \rho U_\infty^2 4af \left(\cos \gamma + \tan \frac{\chi}{2} \sin \gamma - af \sec^2 \frac{\chi}{2} \right) r \delta \psi \delta r \quad (4.47b)$$

The algebraic complexity of estimating the wake rotation velocities is great and even then fluctuation of bound circulation is ignored. The drop in pressure caused by wake rotation, however, is shown to be small in the non-yawed case and so it is assumed that it can safely be ignored in the yawed case.

The moment of the blade element force about the wake axis is

$$\delta M_b = \frac{1}{2} \rho W^2 Bc (C_y \cos \chi - C_x \cos \psi \sin \chi) r \delta r \frac{\delta \psi}{2\pi}$$

where

$$C_y = C_l \sin \phi - C_d \cos \phi$$

therefore

$$\delta M_b = \frac{1}{2} \rho W^2 \sigma_r (C_y \cos \chi - C_x \cos \psi \sin \chi) r^2 \delta r \delta \psi \quad (4.48)$$

The rate of change of angular momentum is the mass flow rate through an elemental area of the disc times the tangential velocity times radius.

$$\delta M_m = \rho U_\infty (\cos \gamma - af) r \delta \psi \delta r 2a' f \Omega r'^{''2}$$

where

$$r'^{''2} = r^2 (\cos^2 \psi + \cos^2 \chi \sin^2 \psi)$$

therefore,

$$\delta M_m = \frac{1}{2} \rho U_\infty^2 \lambda \mu 4a' f(\cos \gamma - af)(\cos^2 \psi + \cos^2 \chi \sin^2 \psi) r^2 \delta r \delta \psi \quad (4.49)$$

The momentum theory, as developed, applies only to the whole rotor disc where the flow induction factor a is the average value for the disc. However, it may be argued that it is better to apply the momentum equations to an annular ring, as in the non-yawed case, to determine a distribution of the flow induction factors varying with radius, reflecting radial variation of circulation. Certainly, for the angular momentum case the tangential flow factor a' will not vary with azimuth position because it is generated by the root vortex and although, in fact, the axial flow factor a does vary with azimuth angle it is consistent to use an annular average for this factor as well.

To find an average for an annular ring the elemental values of force and moment must be integrated around the ring.

For the axial momentum case, taking the vortex method as an example,

$$\int_0^{2\pi} \frac{1}{2} \rho U_\infty^2 4af \left(\cos \gamma + \tan \frac{\chi}{2} \sin \gamma - af \sec^2 \frac{\chi}{2} \right) \delta \psi r \delta r = \sigma_r \int_0^{2\pi} \frac{1}{2} \rho W^2 C_x \delta \psi r \delta r$$

Therefore,

$$8\pi af \left(\cos \gamma + \tan \frac{\chi}{2} \sin \gamma - af \sec^2 \frac{\chi}{2} \right) = \sigma_r \int_0^{2\pi} \frac{W^2}{U_\infty^2} C_x d\psi \quad (4.50)$$

The resultant velocity W and the normal force coefficient C_x are functions of ψ .

And for the angular momentum case,

$$\begin{aligned} & \int_0^{2\pi} \frac{1}{2} \rho U_\infty^2 \lambda \mu 4a' f(\cos \gamma - af)(\cos^2 \psi + \cos^2 \chi \sin^2 \psi) d\psi r^2 \delta r \\ &= \sigma_r \int_0^{2\pi} \frac{1}{2} \rho W^2 (C_y \cos \chi - C_x \sin \chi \cos \psi) d\psi r^2 \delta r \end{aligned}$$

which reduces to

$$\begin{aligned} & 4a' f(\cos \gamma - af) \lambda \mu \pi (1 + \cos^2 \chi) \\ &= \sigma_r \int_0^{2\pi} \frac{W^2}{U_\infty^2} (C_y \cos \chi - C_x \sin \chi \cos \psi) d\psi \end{aligned} \quad (4.51)$$

The non-dimensionalised resultant velocity relative to a blade element is given by

$$\begin{aligned} \frac{W^2}{U_\infty^2} &= [\cos \gamma - a + \lambda \mu a' \sin \chi \cos \psi (1 + \sin \chi \sin \psi)]^2 \\ &+ \left[\lambda \mu (1 + a' \cos \chi (1 + \sin \chi \sin \psi)) + \cos \psi \left(a \cdot \tan \frac{\chi}{2} - \sin \gamma \right) \right]^2 \end{aligned} \quad (4.52)$$

Note that the flow expansion terms, those terms that involve $F(\mu)K(\chi)$, have been excluded from the velocity components in Equation 4.52 because flow expansion is not present in the wake and so there is no associated momentum change. The blade force, which arises from the flow expansion velocity, is balanced in the wake by pressure forces acting on the sides of the stream-tubes, which have a stream-wise component because the stream-tubes are expanding.

Equations 4.50 and 4.51 can be solved by iteration, the integrals being determined numerically. Initial values are chosen for a and a' , usually zero. For a given blade geometry, at each blade element position μ and at each blade azimuth position ψ , the flow angle ϕ is calculated from Equation 4.44, suitably modified to remove the flow expansion velocity, in accordance with Equation 4.52. Then, knowing the blade pitch angle β at the blade element, the local angle of attack can be found. Lift and drag coefficients are obtained from tabulated aerofoil data. Once an annular ring (constant μ) has been completed the integrals are calculated. The new value of axial flow factor a is determined from Equation 4.50 and then the tangential flow factor a' is found from Equation 4.51. Iteration proceeds for the same annular ring until a satisfactory convergence is achieved before moving to the next annular ring (value of μ).

Although the theory supports only the determination of azimuthally averaged values of the axial flow induced velocity, once the averaged tangential flow induction factors have been calculated the elemental form of the momentum equations (Equation 4.47) and the blade element forces (Equation 4.46) can be employed to yield values of a which vary with azimuth.

For the determination of blade forces the flow expansion velocities must be included. The total velocity components, normal and tangential to a blade element, are then as shown in Figure 4.18 and the resultant velocity is

$$\begin{aligned} \frac{W^2}{U_\infty^2} = & \left[\cos \gamma - a(1 + F(\mu)K(\chi) \sin \psi) + \lambda \mu a' \sin \chi \cos \psi (1 + \sin \chi \sin \psi) \right]^2 \\ & + \left[\lambda \mu (1 + a' \cos \chi (1 + \sin \chi \sin \psi)) \right. \\ & \left. + \cos \psi \left(a \tan \frac{\chi}{2} (1 + F(\mu)K(\chi) \sin \psi) - \sin \gamma \right) \right]^2 \end{aligned} \quad (4.52a)$$

4.2.9 Calculated values of induced velocity

The measurement of the induced velocities of a wind turbine rotor in yaw has been undertaken at Delft University of Technology, (Schepers and Snel, 1995). The tests were carried out using a small wind tunnel model so that a steady yaw could be maintained in a steady wind with no tower shadow and no wind shear. The rotor had two blades of 1.2 m diameter which were twisted but had a uniform chord length of 80 mm. The blade root was at a radius of 180 mm and the blade twist was 9° at the root varying linearly with radius to 4° at 540 mm radius and remaining at 4° from there to the tip. The blade aerofoil profile was NACA 0012. The rotor speed was kept constant at 720 rev/minute and the wind speed was held constant at 6.0 m/s. Tests were carried out at 10° , 20° and 30° of yaw angle.

Calculated induced velocities using the momentum equations for the Delft turbine are shown in Figure 4.19: these are the average values for each annulus obtained using Equations 4.50 and 4.51.

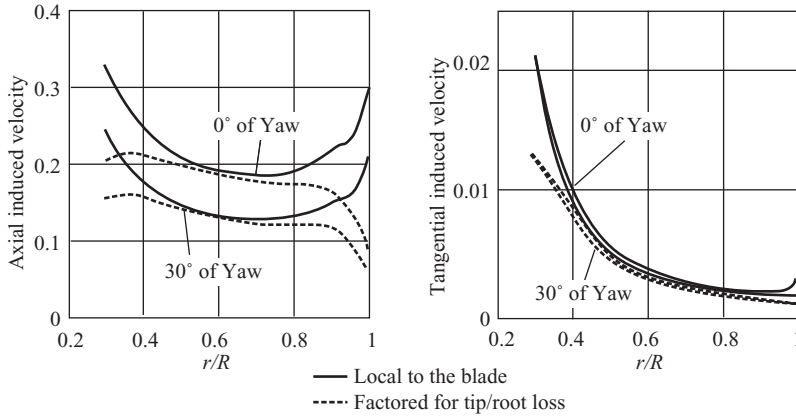


Figure 4.19 Azimuthally averaged induced velocity factors for the Delft turbine

The component velocities at each blade element, as defined in Figure 4.18, are shown in Figure 4.20. Because of the rotational speed of the blades the tangential velocity is much greater than the normal velocity but it is the latter which most influences the variation in angle of attack at the important, outboard sections of the blades, shown in Figure 4.21.

At the inboard sections of the blades it is the variation in tangential velocity which mostly influences the angle of attack variation and this is largely as a result of the changing geometry with azimuth angle rather than the effect of induced velocity.

4.2.10 Blade forces for a rotor in steady yaw

Once the flow induction factors have been determined blade forces can then be calculated. Although the flow expansion velocity is excluded from the determination of the flow induction factors, on the grounds that the consequent blade forces do not cause any change in the momentum of the flow because flow expansion is not present in the developed wake, it must be included when the blade forces are calculated. The flow expansion velocity should

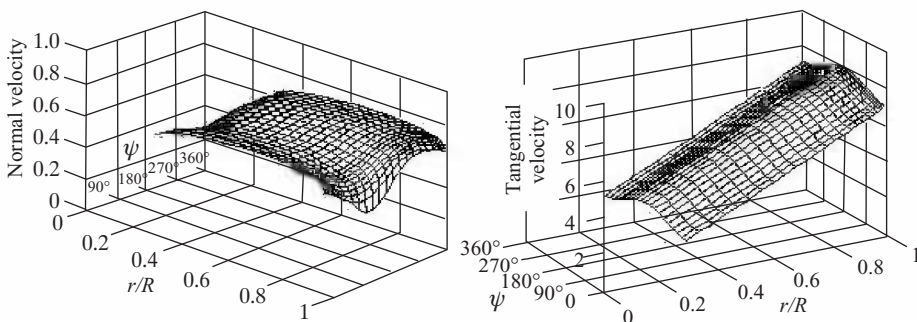


Figure 4.20 Component velocities, normalised with wind speed, at 30° of yaw

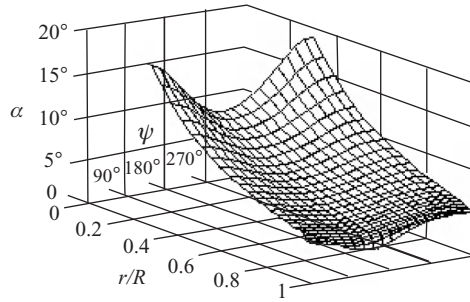


Figure 4.21 Angle of attack variation at 30° of yaw

be dependent on an overall average value of the axial flow induction factor but it is more convenient to use the annular average value as determined by Equation 4.50.

The flow angle and the angle of attack need to be determined anew at each blade element position μ and at each blade azimuth position θ because the flow expansion velocity must now be included, so Equation 4.44 is used in its unmodified form. Drag must also be included in the determination of forces even if it was not in the calculation of the induced velocities.

The blade force per unit span normal to the plane of rotation is

$$\frac{dF_x}{dr} = \frac{1}{2} \rho W^2 c C_x \quad (4.53)$$

which will vary with the azimuth position of the blade. The total force normal to the rotor plane can be obtained by integrating Equation 4.53 along the blade length for each of the blades, taking account of their azimuthal separation, and summing the results. The total normal force will also vary with rotor azimuth.

Similarly, the tangential blade force per unit span is

$$\frac{dF_y}{dr} = \frac{1}{2} \rho W^2 c C_y$$

and the blade torque contribution about the axis of rotation is

$$\frac{dQ}{dr} = \frac{1}{2} \rho W^2 c r C_y \quad (4.54)$$

The total torque is found by integrating along each blade and summing over all the blades, just as for the normal force. Again, the torque on the rotor will vary with azimuth position so to find the average torque will require a further integration with respect to azimuth.

4.2.11 Yawing and tilting moments in steady yaw

The asymmetry of the flow through a yawed rotor, caused by the flow expansion, means that a blade sweeping upwind has a higher angle of attack than when it is sweeping downwind, as

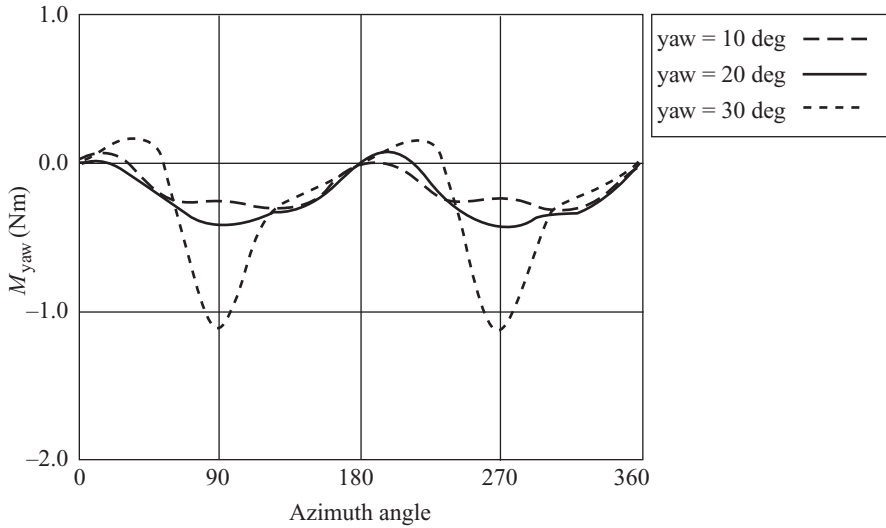


Figure 4.22 Measured yaw moments on the Delft turbine

shown in Figure 4.16. The blade lift upwind will, therefore, be greater than the lift downwind and a similar differential applies to the forces normal to the rotor plane. It can be seen, therefore, that there is a net moment about the yaw (vertical axis) in a direction which will tend to restore the rotor axis to a position aligned with the wind direction. The yawing moment is obtained from the normal force of Equation 4.53

$$\frac{dM_z}{dr} = \frac{1}{2} \rho W^2 c r C_x \sin \psi \quad (4.55)$$

which will also vary with the azimuth position of the blade. The total single-blade yawing moment at each azimuth position is obtained by integrating Equation 4.55 along the length of the blade. Summing the moments for all blades, suitably separated in phase, will result in the yawing moment on the rotor.

A similar calculation can be made for the tilting moment, the moment about the horizontal diametral axis (y axis) of the rotor.

$$\frac{dM_y}{dr} = \frac{1}{2} \rho W^2 c r C_x \cos \psi \quad (4.56)$$

The existence of yawing and tilting moments predicted by the blade element theory is inconsistent with the momentum and vortex theories because they predict only a uniform pressure distribution on the rotor disc. Actually, the momentum and vortex theories predict velocities from which it is only possible to deduce a uniform pressure distribution.

Measured results of rotor yaw moment for the Delft turbine are shown in Figure 4.22 and the corresponding calculated yawing moments are shown in Figure 4.23.

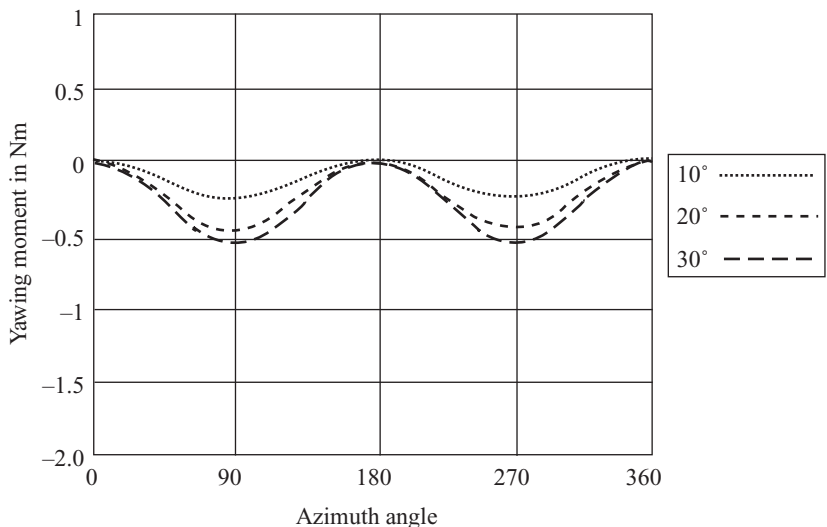


Figure 4.23 Calculated yaw moments on the Delft turbine

The measured yawing moments were derived from strain gauge readings of the flat-wise bending strain close to the root of the blade at 129 mm radius. Flat-wise, or flap-wise bending causes only displacements normal to the rotor plane. The calculated yawing moments are determined at the same radial position on the blade and are, therefore, not the true yawing moments about the actual yaw axis.

The comparison between the measured and calculated yaw moments is quite good taking into account the limitations of the theory. At 30° of yaw the calculated values underestimate the measurements significantly whereas at the two lower angles the correspondence is much closer.

It should be noted that the mean yawing moment is not zero and that the sign of the moment, being negative, means that it endeavours to restore the rotor axis to alignment with the wind direction.

The yawing moment comparison is a test of the usefulness of the theory developed in this section and it would seem that for general engineering purposes it passes the test.

The measured tilting moments (Figure 4.24) appear to be of about the amplitude for all three yaw angles whereas the calculated moments (Figure 4.25) increase with yaw angle.

For 30° of yaw the magnitudes of the measured and calculated tilting moments are comparable. The measured mean tilting moment is quite definitely non-zero and positive but the calculated mean moment is much smaller although still positive. A positive tilt rotation would displace the upper part of the rotor disc in the downwind direction. In the theory the small mean tilting moment is caused by the wake rotation velocities.

A theory based upon computational fluid mechanics should provide a much more accurate prediction of the aerodynamics of a wind turbine in yaw. However, the severe computational time limitations associated with CFD solutions precludes their use in favour of the simple theory outlined in these pages.

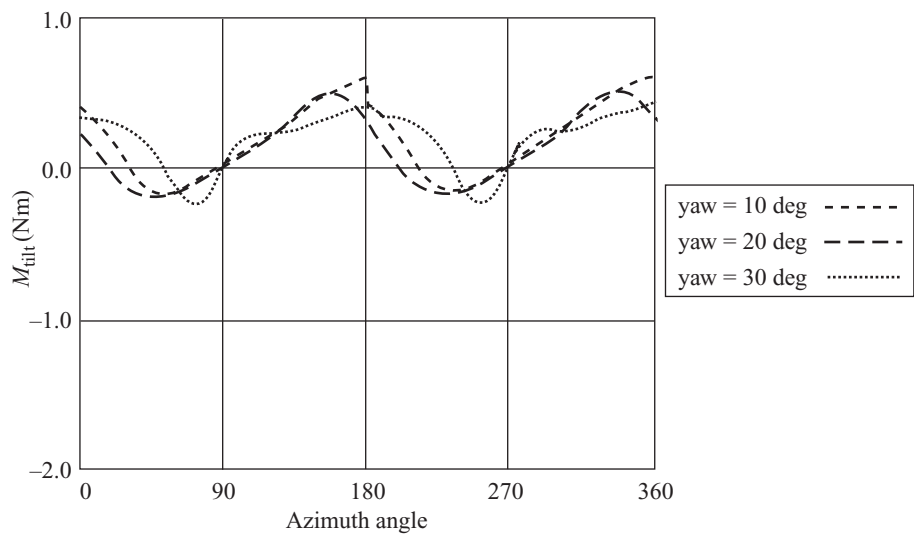


Figure 4.24 Measured tilt moments on the Delft turbine

4.3 The method of acceleration potential

4.3.1 Introduction

An aerodynamic model that is applied to the flight performance of helicopter rotors, and which can also be applied to wind turbine rotors that are lightly loaded, is that based upon the idea of acceleration potential. The method allows distributions of the pressure drop across

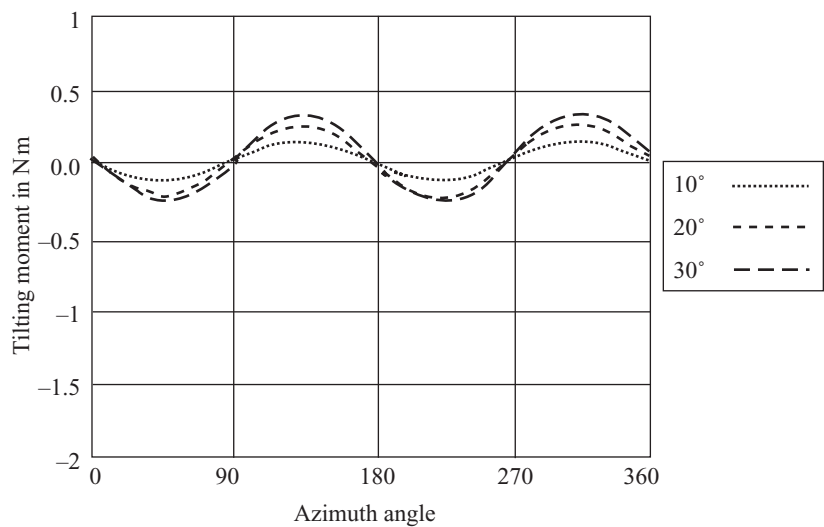


Figure 4.25 Calculated tilt moments on the Delft turbine

an actuator disc that are more general than the, strictly, uniform pressure distribution of the momentum theory. The model has been expounded by Kinner (1937), inspired by Prandtl, who has developed expressions for the pressure field in the vicinity of an actuator disc, treating it as a circular wing. To regard a rotor as a circular wing requires an infinity of very slender blades so that the solidity remains small.

Kinner's theory, which is derived from the Euler Equations, assumes that the induced velocities are small compared with the general flow velocity. If u , v and w are the velocities induced by the actuator disc in the x , y and z directions, respectively, and which are very much smaller than the free-stream velocity in the x direction U_∞ , then the rate of change of momentum in the x direction of a unit volume of air will be in response to the pressure gradient in that direction

$$\rho \left((U_\infty + u) \frac{\partial(U_\infty + u)}{\partial x} + v \frac{\partial(U_\infty + u)}{\partial y} + w \frac{\partial(U_\infty + u)}{\partial z} \right) = - \frac{\partial p}{\partial x} \quad (4.57)$$

The free-stream velocity U_∞ does not change with position, therefore, for example, $\partial U_\infty / \partial x = 0$. Also, $U_\infty \gg (u, v, w)$ and so, for example, $v(\partial u / \partial y)$ can be ignored in comparison with $U_\infty(\partial u / \partial x)$. The momentum equation in the x direction then simplifies to

$$\rho U_\infty \frac{\partial u}{\partial x} = - \frac{\partial p}{\partial x} \quad (4.58a)$$

Similarly, in the y and z directions, the momentum equations are also simplified

$$\rho U_\infty \frac{\partial v}{\partial x} = - \frac{\partial p}{\partial y} \quad (4.58b)$$

and

$$\rho U_\infty \frac{\partial w}{\partial x} = - \frac{\partial p}{\partial z} \quad (4.58c)$$

Differentiating each momentum equation with respect to its particular direction and adding together the results gives

$$\rho U_\infty \frac{\partial}{\partial x} \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) = - \left(\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 p}{\partial z^2} \right)$$

but, for continuity of the flow, $(\partial u / \partial x + \partial v / \partial y + \partial w / \partial z) = 0$,

Therefore,

$$\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 p}{\partial z^2} = 0 \quad (4.59)$$

which is the Laplace Equation governing the pressure field on and surrounding the actuator disc. Given the boundary conditions at the actuator disc, Equation 4.59 can be solved for the pressure field and, in particular, the pressure distribution at the disc. The pressure is continuous

everywhere except across the disc surfaces where there is the usual pressure discontinuity, or pressure drop, in the wind turbine case.

In Coleman's analysis the pressure drop distribution across the disc is uniform (it is only as a result of combining the theory with the blade element theory that a non-uniform pressure distribution can be achieved) but falls to zero, abruptly, at the disc edge. Kinner assumes that the pressure drop is zero at the disc edge and changes in a continuous manner as radius decreases.

The simplified Euler Equations 4.58 allow pressure to be regarded as the potential field from which the acceleration field can be obtained, by differentiation, and thence the velocity field, by integration. Commencing upstream where the known free-stream conditions apply the velocity components can be determined by progressive integration towards the disc.

The pressure discontinuity across the rotor disc will be as shown in Figure 3.2, Chapter 3. The magnitude of the step in pressure will be twice the pressure level (above the far field level) that occurs just upstream of the disc. The pressure gradient, however, normal to the rotor disc, will be continuous.

4.3.2 The general pressure distribution theory of Kinner

Kinner's solution (1937) of Equation 4.59 is mathematically complex and is achieved by means of a co-ordinate transformation. The Cartesian co-ordinates centred in the rotor plane (x'', y', z), as defined in Figure 4.11, are transformed to, what is termed, an ellipsoidal co-ordinate system (v, η, ψ), where ψ is the azimuth angle.

$$\frac{x''}{R} = v\eta, \quad \frac{y'}{R} = \sqrt{1-v^2}\sqrt{1+\eta^2}\sin\psi \quad \text{and} \quad \frac{z}{R} = \sqrt{1-v^2}\sqrt{1+\eta^2}\cos\psi \quad (4.60)$$

On the surface of the rotor disc $\eta = 0$ and $r/R = \mu = \sqrt{1-v^2}$ or, conversely, $v = \sqrt{1-\mu^2}$

The transformation separates the variables and allows the pressure field to be expressed as the product of three functions

$$p(v, \eta, \psi) = \Phi_1(v)\Phi_2(\eta)\Phi_3(\psi) \quad (4.61)$$

each separate function being the solution of a separate, ordinary differential equation,

$$\frac{d}{dv} \left((1-v^2) \frac{d\Phi_1(v)}{dv} \right) + \left(n(n-1) - \frac{m^2}{1-v^2} \right) \Phi_1(v) = 0 \quad (4.62a)$$

$$\frac{d}{d\eta} \left((1+\eta^2) \frac{d\Phi_2(\eta)}{d\eta} \right) + \left(\frac{m^2}{1+\eta^2} - n(n+1) \right) \Phi_2(\eta) = 0 \quad (4.62b)$$

$$\frac{d^2\Phi_3(\psi)}{d\psi^2} + m^2\Phi_3(\psi) = 0 \quad (4.62c)$$

where m and n are positive integers.

Equations 4.62a and 4.62b have the form of Legendre's associated differential equation which has solutions called associated Legendre polynomials of the first and second kinds, respectively, see Pipes and Harvill (1970) and van Bussel (1995).

If $m = 0$ then Equation 4.62a is reduced to Legendre's differential equation, the two solutions for which are:

$$\Phi_1(v) = P_n(v) \quad (4.63a)$$

and

$$\Phi_1(v) = Q_n(v) \quad (4.63b)$$

where $P_n(v)$ is a Legendre polynomial of the first kind and $Q_n(v)$ is a Legendre polynomial of the second kind.

$$P_n(v) = \frac{1}{2^n n!} \frac{d^n}{dv^n} (v^2 - 1)^n \quad (4.64)$$

Although the polynomials extend beyond the range $v^2 \leq 1$, over that interval the polynomials are mutually orthogonal

$$\int_{-1}^1 P_n(v) P_k(v) dv = 0 \quad n \neq k \quad (4.65)$$

For $n = 0$ the Legendre polynomial of the second kind is

$$Q_0(v) = \frac{1}{2} \ln \left(\frac{1+v}{1-v} \right) \quad (4.66)$$

For $n > 0$ the Legendre polynomials of the second kind $Q_n(v)$ can be obtained from the polynomials of the first kind. For $n = 1$ to 4

$$\begin{aligned} Q_1(v) &= P_1(v)Q_0(v) - 1 \\ Q_2(v) &= P_2(v)Q_0(v) - \frac{3}{2}v \\ Q_3(v) &= P_3(v)Q_0(v) - \frac{5}{2}v^2 + \frac{2}{3} \\ Q_4(v) &= P_4(v)Q_0(v) - \frac{35}{8}v^3 + \frac{55}{24}v \end{aligned} \quad (4.67)$$

The solutions for Equation 4.62b, with $m = 0$, are the same as for 4.62a but with imaginary arguments, that is,

$$\Phi_2(\eta) = P_n(i\eta) \quad (4.68a)$$

and

$$\Phi_2(\eta) = Q_n(i\eta) \quad (4.68b)$$

where $Q_0(i\eta) = i \tan^{-1} \eta$ for $\eta < 1$ and $Q_0(i\eta) = i(\pi/2 - \tan^{-1} \eta)$ for $\eta > 1$.

For non-zero values of the integer m the solutions of Equation 4.62a become

$$P_n^m(v) = (1 - v^2)^{m/2} \frac{d^m}{dv^m} P_n(v) \quad (4.69)$$

If $m > n$ then $P_n^m(v) = 0$.

$$Q_n^m(v) = (1 - v^2)^{m/2} \frac{d^m}{dv^m} Q_n(v)$$

But, from Equation 4.66, $Q_n^m(v) = \infty$ at $v^2 = 1$, which is considered physically unapplicable, and these functions are excluded from the solution.

For solutions of Equation 4.62b for non-zero values of m ,

$$P_n^m(i\eta) = (1 + \eta^2)^{m/2} \frac{d^m}{d(i\eta)^m} P_n(i\eta)$$

Inspection reveals that $P_n^m(i\eta) \rightarrow \infty$ as $\eta \rightarrow \infty$ which means that pressure would be infinite in the far field which is not physically acceptable, therefore, these terms will also not be included.

$$Q_n^m(i\eta) = (1 + \eta^2)^{m/2} \frac{d^m}{d(i\eta)^m} Q_n(i\eta) \quad (4.70)$$

Equations 4.69 and 4.70 are known as associated Legendre polynomials.

The solution to differential Equation 4.62c is more straight forward than for the other two governing equations and has the two solutions

$$\Phi_3(\psi) = \cos m\psi, \Phi_3(\psi) \sin m\psi \quad (4.71)$$

The complete solution, therefore, for the pressure field surrounding a rotor disc is

$$p(v, \eta, \psi) = \sum_{m=0}^M \sum_{n=m}^N P_n^m(v) Q_n^m(i\eta) (C_n^m \cos m\psi + D_n^m \sin m\psi) \quad (4.72)$$

The upper limits M and N can have any positive integer value.

The polynomial $Q_n^m(i\eta)$ is imaginary for odd values of m and real for even values therefore the arbitrary constants C_n^m and D_n^m must be real or imaginary, accordingly, in order that the pressure field be real.

Any combination of terms in Equation 4.72 can be used, whatever suits the conditions. For there to be a pressure discontinuity across the disc, but continuously varying pressure elsewhere, the solutions must be restricted to those for which $n + m$ is odd. Of course, limiting the number of terms, other than has been described, may result in an approximate solution.

The pressure discontinuity across the rotor disc will be as shown in Figure 3.2. The magnitude of the step in pressure will be twice the pressure level (above the far field level) that occurs just upstream of the disc. The pressure gradient, however, normal to the rotor disc, will be continuous.

4.3.3 The axi-symmetric pressure distributions

For the wind turbine rotor disc the simplest situation is for $m = 0$ which means that the pressure distribution is axi-symmetric. The permitted values of n must be odd.

For $n = 1$ the polynomials are

$$P_1^0(v) = v = \sqrt{1 - \mu^2} \quad (4.73a)$$

and

$$Q_1^0(i\eta) = \eta \cdot \tan^{-1} \frac{1}{\eta} - 1 \quad (4.73b)$$

So, on the disc, where $\eta = 0$,

$$Q_1^0(i0) = -1$$

Therefore, the pressure distribution is

$$p(\mu) = -C_1^0 \sqrt{1 - \mu^2} \quad (4.74)$$

If the pressure in Equation 4.74 is non-dimensionalised using the free-stream dynamic pressure $\frac{1}{2} \rho \cdot U_\infty^2$ the value of C_1^0 can be related to the thrust coefficient by integrating the pressure distribution of Equation 4.74 over the disc area

$$\pi R^2 C_T = -R^2 C_1^0 \int_0^{2\pi} \int_0^1 \sqrt{1 - \mu^2} \mu d\mu d\psi = -\frac{2}{3} \pi R^2 C_1^0$$

Therefore,

$$C_1^0 = -\frac{3}{2} C_T \quad (4.75)$$

and so the pressure step across the disc is

$$p_1(\mu) = \frac{3}{2} C_T \sqrt{1 - \mu^2} \quad (4.76)$$

All the remaining polynomials (for $m = 0$ and odd values of $n > 1$) produce zero thrust.

To modify the pressure distribution to suit the boundary conditions an appropriate linear combination of solutions can be added to that of Equation 4.76.

The application to helicopter rotors leads to a requirement for the pressure and the radial pressure gradient to be zero at the rotor axis as these conditions correspond to the pressure on actual rotors. The above pressure distribution does not have zero pressure at the rotor axis and so needs to be combined with at least one other solution. The second axi-symmetric solution, $n = 3$, is

$$P_3^0(v) = \frac{1}{2} v(5v^2 - 3) = \frac{1}{2} \sqrt{1 - \mu^2} (2 - 5\mu^2) \quad (4.77a)$$

and

$$Q_3^0(i\eta) = -\frac{\eta}{2}(5\eta^2 + 3) \tan^{-1} \frac{1}{\eta} + \frac{5}{2}\eta^2 + \frac{2}{3}, \text{ so } Q_3^0(i0) = \frac{2}{3} \quad (4.77b)$$

The second pressure distribution is, therefore,

$$p_2(\mu) = \frac{1}{3}C_3^0\sqrt{1-\mu^2}(2-5\mu^2) \quad (4.78)$$

The sum of the two pressure distributions must be zero where $\mu = 0$, so

$$C_3^0 = -\frac{9}{4}C_T$$

and the combination of the two distributions is

$$p_{1-2}(\mu) = \frac{15}{4}C_T\mu^2\sqrt{1-\mu^2} \quad (4.79)$$

Note that the pressure distributions given in Equations (4.76), (4.78) and (4.79) are normalised by the free stream dynamic pressure, $0.5\rho U_\infty^2$, in each case. The three normalised distributions are shown in Figure 4.26.

As most modern wind turbines are designed to achieve as uniform a pressure distribution as practicable, to maximise efficiency, the form chosen for the helicopter rotor might need to be modified. A uniform pressure distribution can be formed by combining solutions but, because the pressure discontinuity must itself be discontinuous at the disc edge, it would mean that a great many solutions would be required. Tip-loss effects would require zero pressure at both the blade tips and at the hub, but for most of the blade span the pressure should be

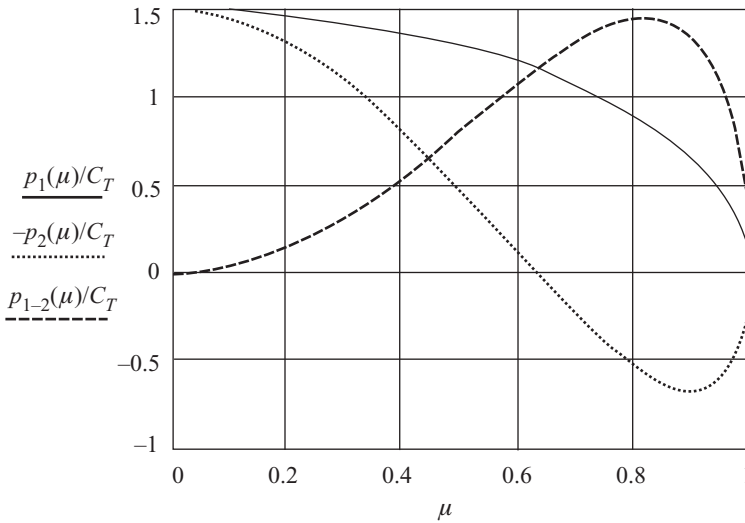


Figure 4.26 Radial pressure distributions of the first two solutions and their combination to satisfy the requirements at the rotor axis

uniform. It should be pointed out that the *blade loading* caused by uniform pressure does increase linearly with radius.

The induced velocity field caused by the axi-symmetric pressure distribution has to be obtained from the pressure field by integrating Equations 4.58 commencing far upstream where free-stream conditions are assumed to apply. The upstream conditions also depend upon the angle of yaw of the disc. The integration continues until a point on the disc is reached where the induced velocity is to be determined.

The particular induced velocity component that is most important for determining the angle of attack on a blade element is normal to the rotor disc, that is, the axial induced velocity. Mangler and Squire (1950) calculated the axial induced velocity distribution as a function of yaw angle by expressing the velocity as a Fourier series of the azimuth angle ψ .

$$\frac{u}{U_\infty} = C_T \left(\frac{A_0(\mu, \gamma)}{2} + \sum_{k=1}^{\infty} A_k(\mu, \gamma) \sin k\psi \right) \quad (4.80)$$

For the pressure distribution of Equation 4.79 the Fourier coefficients are

$$A_0(\mu, \gamma) = -\frac{15}{8}\mu^2\sqrt{1-\mu^2} \quad (4.81a)$$

$$A_1(\mu, \gamma) = -\frac{15}{256}\mu(9\mu^2 - 4)\tan\frac{\gamma}{2} \quad (4.81b)$$

$$A_3(\mu, \gamma) = -\frac{45\pi}{256}\mu^3\tan^3\frac{\gamma}{2} \quad (4.81c)$$

Higher order odd terms are zero. There are also even terms which have the general form

$$A_k = -(-1)^{\frac{k-2}{2}} \frac{3}{4} \left[\frac{k+v}{k^2-1} \left(\frac{9v^2+k^2-6}{k^2-9} \right) + \frac{3v}{k^2-9} \right] \left(\frac{1-v}{1+v} \right)^{\frac{k}{2}} \tan^{\frac{k}{2}} \frac{\gamma}{2}$$

where $v^2 = 1 - \mu^2$ and k is an even integer greater than zero.

The average value of the axial induced velocity is independent of yaw angle and is given by

$$a_0 = \frac{u_0}{U_\infty} = \frac{1}{4}C_T$$

where u_0 is the average axial induced velocity.

Thus, the *average* value of the axial flow induced velocity is related to the thrust coefficient by

$$C_T = 4a_0 \quad (4.82)$$

compared with the momentum theory $C_T = 4a_0(1 - a_0)$ or compared with any of the expressions developed for yawed conditions, Equations 4.2, 4.18 and 4.24.

Because of the assumption that the induced velocity is small compared with the flow velocity, a_0 is small compared with 1.

Clearly, the acceleration potential method only applies if the value of C_T is much less than 1.

The once per revolution term in Equation 4.80 will cause an angle of attack variation and, hence, a lift variation that will cause a yawing moment on the disc. However, the pressure distribution, being axi-symmetric, cannot cause a yawing moment. The situation is much the same as for the vortex theory of Coleman *et al.* (1945).

Pitt and Peters (1981) use, or rather impose, Glauert's assumption (Equation 4.21) for the variation of the axial induced flow factor.

$$a = a_0 + a_s \mu \sin \psi \quad (4.83)$$

The value of a_s is obtained by equating the first moment about the yaw axis of Equation 4.83 with the first moment of Equation 4.80 using the Mangler and Squire velocity distributions of Equations 4.81.

$$\begin{aligned} & \int_0^{2\pi} \int_0^1 \mu \sin \psi (a_0 + a_s \mu \sin \psi) \mu d\mu d\psi \\ &= \int_0^{2\pi} \int_0^1 C_T \mu \sin \psi \left(\frac{A_0(\mu, \gamma)}{2} + \sum_{k=1}^{\infty} A_n(\mu, \gamma) \sin k\psi \right) \mu d\mu d\psi \end{aligned} \quad (4.84)$$

All terms, apart from that containing A_1 , vanish on integration, giving

$$a_s = \frac{15\pi}{128} C_T \tan \frac{\gamma}{2} \quad (4.85)$$

Hence, using Equation 4.82, the axial induced velocity becomes

$$a = a_0 \left(1 + \frac{15\pi}{32} \mu \tan \frac{\gamma}{2} \sin \psi \right) \quad (4.86)$$

which, apart from the use of the yaw angle instead of the wake skew angle, has the same form as Equations 4.21 and 4.31 and so there is some consistency in the various methods for dealing with yawed flow.

4.3.4 The anti-symmetric pressure distributions

As has been determined in Section 4.1.11, there is a moment about the vertical diameter of a yawed wind turbine rotor disc, the restoring yaw moment. An axi-symmetric pressure distribution, however, is not capable of producing a yaw moment so more terms from the series solution of Equation 4.72 need to be included.

The only terms in Equation 4.72 which will yield a yawing moment are those for which $m = 1$ and for which $D_n^1 \neq 0$. Terms for which $m = 1$ and $C_n^1 \neq 0$ will cause a tilting moment. Recalling that $m + n$ must be odd to achieve a pressure discontinuity across the disc the values of n that may be combined with $m = 1$ must be even.

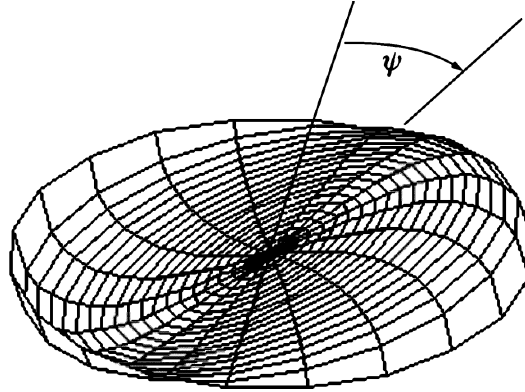


Figure 4.27 The form of the pressure distribution which yields a yawing moment

Because of the nature of the Legendre polynomials only one term in the series of Equation 4.72 will produce a net thrust and only one term will produce a yawing moment, which is a first moment. Similarly only one term will produce a second moment, and so on.

The unique term in Equation 4.72 which yields a yawing moment is that for which $m = 1$, $n = 2$ and $C_n^1 \neq 0$, therefore,

$$P_2^1(v) = 3v\sqrt{1-v^2} = 3\mu\sqrt{1-\mu^2} \quad (4.87)$$

and

$$Q_2^1(i\eta) = 3i\eta\sqrt{1+\eta^2}\tan^{-1}\frac{1}{\eta} - 3i\sqrt{1+\eta^2} + \frac{i}{\sqrt{1+\eta^2}} \quad (4.88)$$

so, at the disc,

$$Q_2^1(i0) = -2i \quad (4.88a)$$

A zero pressure gradient at the rotor axis is not appropriate in this case because the pressure distribution is anti-symmetric about the yaw axis, therefore,

$$p(\mu, \psi) = P_2^1(\mu)Q_2^1(i0)D_2^1\sin\psi = -6iD_2^1\mu\sqrt{1-\mu^2}\sin\psi \quad (4.89)$$

The pressure distribution is shown in Figure 4.27.

The yawing moment coefficient is defined by

$$C_{mz} = \frac{M_z}{\frac{1}{2}\rho \cdot U_\infty^2 \pi \cdot R^3} \quad (4.90)$$

As before, if the pressure in Equation 4.89 is non-dimensionalised by the free-stream dynamic pressure $\frac{1}{2}\rho \cdot U_\infty^2$, then

$$C_{mz} = \frac{1}{\pi} \int_0^{2\pi} \int_0^1 \mu \sin\psi \cdot p(\mu, \psi) \mu \cdot d\mu d\psi = -\frac{1}{\pi} 6i D_2^1 \int_0^1 \mu^3 \sqrt{1-\mu^2} d\mu \int_0^{2\pi} \sin^2\psi \cdot d\psi \quad (4.91)$$

which gives

$$iD_2^1 = -\frac{5}{4} \cdot C_{mz} \quad (4.92)$$

To establish a relationship between the yawing moment coefficient and the axial velocity induced by the pressure distribution of Equation 4.89 the velocity distribution has to be obtained by integrating Equations 4.58. Unfortunately, no analytical solution has been determined for the anti-symmetric case, as Mangler and Squire have done for the symmetric case. Numerical values of induced velocities need to be calculated from Equations 4.58 using the pressure distribution defined by Equations 4.87 and 4.88.

Pitt and Peters (1981) have determined the axial velocity distribution for values of the yaw angle from 0° to 90° : the yaw angle fixes the far upstream conditions where the integration commences. The velocity distribution found corresponds to that of Equation 4.80 for the axi-symmetric case. Pitt and Peters again impose the form of Equation 4.83 and determine the average value of the axial induced velocity a_0 and the value of a_s , using the same method of Equation 4.84: in both cases, of course, numerical integration is necessary.

The values of a_0 are not zero, as might have been expected from the anti-symmetric pressure distribution, but are equal and opposite to the values of a_s found for the corrected axi-symmetric pressure distribution of Equation 4.79. The variation of the two coefficients a_0 and a_s with yaw angle γ is determined numerically but, using the Mangler and Squire analytical forms for guidance, analytical variations can be inferred. Pitt and Peters found that the linearised axial induced velocity distribution is

$$a_0 = -\frac{15}{128}\pi \tan \frac{\gamma}{2} C_{mz} \quad (4.93)$$

and

$$a_s = -\left(1 - \tan^2 \frac{\gamma}{2}\right) C_{mz} \quad (4.94)$$

Pitt and Peters also include a cosine term in the linearised axial induced flow factor representation of Equation 4.83 which will only arise if $C_2^1 \neq 0$

$$a = a_0 + a_s \mu \sin \psi + a_c \mu \cos \psi \quad (4.95)$$

In which case there will be an additional pressure distribution given by

$$p(\mu, \psi) = P_2^1(\mu) Q_2^1(i0) C_2^1 \cos \psi = -6i C_2^1 \mu \sqrt{1 - \mu^2} \cos \psi \quad (4.96)$$

The tilting moment coefficient is given by

$$C_{my} = \frac{1}{\pi} \int_0^{2\pi} \int_0^1 \mu \cos \psi p(\mu, \psi) \mu \cdot d\mu \cdot d\psi = -\frac{1}{\pi} 6i C_2^1 \int_0^1 \mu^3 \sqrt{1 - \mu^2} d\mu \int_0^{2\pi} \cos^2 \psi \cdot d\psi \quad (4.97)$$

Therefore,

$$iC_2^1 = -\frac{5}{4} C_{my} \quad (4.98)$$

The axial induced velocity distribution resulting from the pressure field of Equation 4.96 is calculated by numerical integration of Equations 4.58 and is then matched with the linear velocity distribution of Equation 4.95 using again the same method for Equation 4.84.

$$\begin{aligned} & \int_0^{2\pi} \int_0^1 \mu \cos \psi (a_0 + a_c \mu \cos \psi) \mu d\mu d\psi \\ &= \int_0^{2\pi} \int_0^1 \mu \cos \psi C_T \left(\frac{1}{2} A_0(\mu, \gamma) + \sum_{k=1}^{\infty} A_k(\mu, \gamma) \cos k\psi \right) \mu d\mu d\psi \quad (4.99) \end{aligned}$$

The functions $A_n(\mu, \gamma)$ are determined numerically. Again, using the Mangler and Squire results as guidance, an expression for a_c is found.

$$a_c = -\sec^2 \frac{\gamma}{2} \cdot C_{my} \quad (4.100)$$

4.3.5 The Pitt and Peters model

Pitt and Peters (1981) have developed the linear theory that relates the axial induced flow factors to the thrust and moment coefficients given in Equations 4.82, 4.85, 4.93, 4.94 and 4.100, which collect together in matrix form

$$\begin{bmatrix} a_0 \\ a_c \\ a_s \end{bmatrix} = \begin{bmatrix} \frac{1}{4} & 0 & -\frac{15}{128}\pi \tan \frac{\gamma}{2} \\ 0 & -\sec^2 \frac{\gamma}{2} & 0 \\ \frac{15}{128}\pi \tan \frac{\gamma}{2} & 0 & -\left(1 - \tan^2 \frac{\gamma}{2}\right) \end{bmatrix} \begin{bmatrix} C_T \\ C_{my} \\ C_{mz} \end{bmatrix} \quad (4.101a)$$

$$(a) = [L](C) \quad (4.101b)$$

The solution procedure is to assume initial values for (a) from which the values of (C) can be calculated from the blade element theory. New values of (a) are then found from Equation 4.101 and an iteration proceeds.

For the wind turbine the value of a_0 may not be small compared with 1 and so the above procedure will converge on values of a_0 which are too small compared with what the momentum theory would deliver.

To produce more realistic results, that is, results in line with Glauert's momentum theory, the expression for C_T could be

$$C_T = 4a\sqrt{1 - a(2\cos\gamma - a)} = 4aA_G(a) \quad (4.18a)$$

Or, to use the Coleman theory,

$$C_T = 4a \left(\cos\gamma + \tan \frac{\chi}{2} \sin\gamma - a \sec^2 \frac{\chi}{2} \right) = 4aA_C(a) \quad (4.24a)$$

Also, it is usual for the wake skew angle to be used in matrix $[L]$ instead of the yaw angle.

The matrix $[L]$ should then be modified to become

$$[L] = \begin{bmatrix} \frac{1}{4A(a_0)} & 0 & -\frac{15}{128}\pi \tan \frac{\chi}{2} \\ 0 & -\sec^2 \frac{\chi}{2} & 0 \\ \frac{15}{128A(a_0)}\pi \tan \frac{\chi}{2} & 0 & -\left(1 - \tan^2 \frac{\chi}{2}\right) \end{bmatrix} \quad (4.102)$$

where $A(a_0)$ is chosen according to which momentum theory is to be used.

The Pitt and Peters method does not include any determination of induced velocities in the plane of the rotor disc and as a consequence it is not possible to account for wake rotation. However, it is possible that the Kinner solutions $Q_n^m(v)$ that were excluded from the analysis because they give infinite pressure at $v^2 = 1$, which lies along the axis of rotation, may give velocity distributions which provide for wake rotation; the momentum theory of Section 3.3 also predicts an infinite pressure at the axis of rotation because of wake rotation. In practice, of course, the rotor disc would not extend to the axis of rotation and the singularity would not occur.

With or without wake rotation a flow angle ϕ can be determined from which a torque can be found. If the normal force on an element of the rotor disc is equal to $\delta L \cos \phi$ then the tangential force will be $\delta L \sin \phi$.

4.3.6 The general acceleration potential method

Peters, with a number of associates, has developed the theory further and a reading of references (Pitt and Peters, 1981; Goanker and Peters, 1988; HaQuang and Peters, 1988) is recommended.

The acceleration potential method has been developed specifically for wind turbines by van Bussell (1995), where a much more comprehensive account of the theory is given.

If the velocity field at the rotor disc associated with any chosen Kinner pressure distribution, including the in-plane induced velocity components, was available then a solution could be determined for any rotor by matching the forces given by the combined pressure fields of a sufficient number of Kinner distributions with those forces given by the blade element theory.

4.3.7 Comparison of methods

A project to compare existing methods of predicting yaw behaviour, among other aspects of the aerodynamic behaviour of wind turbines, was reported upon (Snel and Schepers, 1995). Figure 4.28 shows results obtained by various methods for predicting the yawing moment of the 2 MW, three blade turbine at Tjæreborg in Denmark at a yaw angle of 32° and a wind speed of 8.5 m/s.

Most of the theoretical predictions in Figure 4.28 have the correct phasing and about the correct mean yawing moment but the amplitude of the yawing moment varies. In this example the second method bears the closest comparison with the measured data. Generally, the amplitude of the yawing moment variation is under-estimated by the theoretical predictions, whereas the mean yawing moment is quite well predicted.

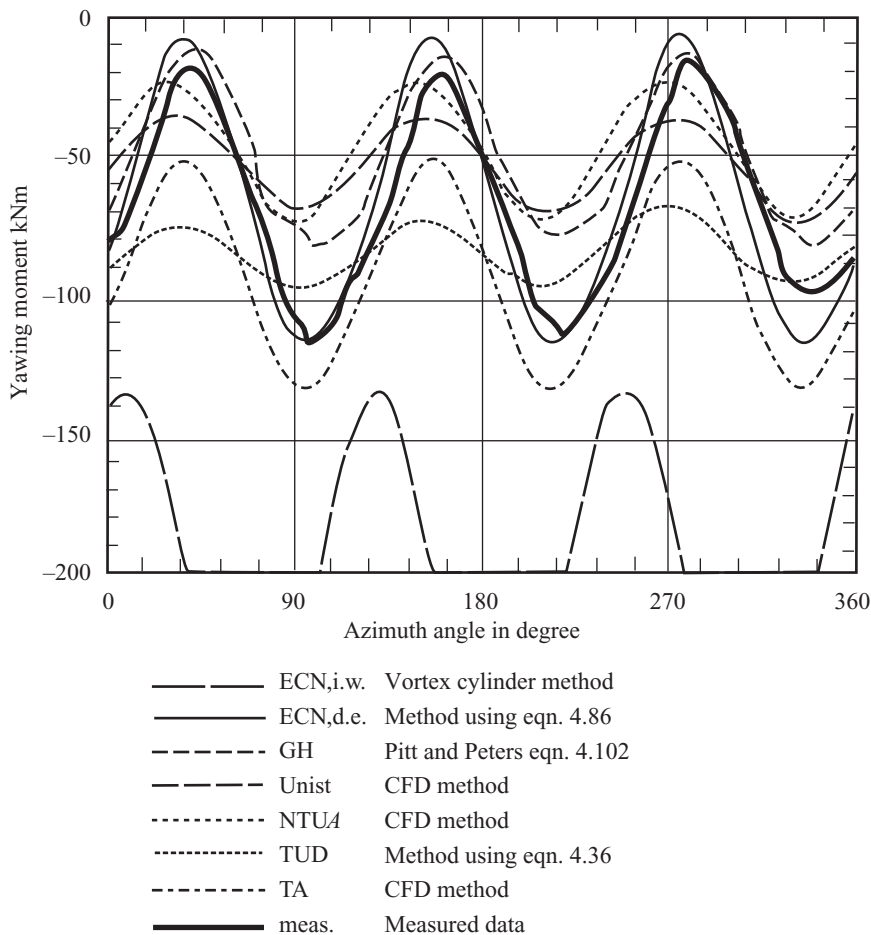


Figure 4.28 Yawing Moment on the Tjæreborg Turbine at 32° Yaw and 8.5 m/s

4.4 Unsteady flow

4.4.1 Introduction

Natural winds are almost never steady in either strength or direction and so it is seldom that the conditions for the momentum theory apply. It takes a finite time for the wind to travel from far upwind of a rotor to far downwind and in that time wind conditions will change so an equilibrium state is never achieved. Even if the ‘average’ wind speed changes only slowly, small scale turbulence will cause a continuous unsteadiness in the velocities impinging on a rotor blade.

Several approximate solutions offer themselves for the determination of the dynamic flow conditions at the rotor disc. It could be assumed that the induced velocity remains fixed at the level determined by the average wind speed blowing over a set period of time, that may be quite short. The wake remains frozen while the unsteady component of the wind passes

through the rotor disc unattenuated by the presence of the rotor. The unsteady forces would be determined by the blade element theory. Alternatively, the induced velocity through the rotor disc could be determined from the instantaneous wind velocity as if that velocity was steady. The induced velocity will change as the wind speed changes but it must be assumed that the entire wake changes instantaneously to remain in step. Equilibrium in the wake is maintained at all times.

The truth lies somewhere between the two scenarios given above, both of which rely on simple assumptions about the state of the wake.

The acceleration potential method avoids reference to the wake and allows the flow conditions at the rotor disc to be determined by the upwind flow field, which is much simpler to determine than that of the wake.

In steady flow conditions the velocity at a fixed point in the upwind flow field is constant; acceleration of the flow from point to point takes place (e.g. $U_\infty(\partial u/\partial x)$ in the x -direction, assuming u , the induced velocity, is much smaller than U_∞) but no rate of change of velocity with time (e.g. $\partial u/\partial t$) occurs at a single point. In unsteady flow, conditions at a fixed point do change with time and the total acceleration in the x -direction is then $\partial u/\partial t + U_\infty(\partial u/\partial x)$. The additional acceleration requires an additional inertia force, the reaction to which will change the force on the rotor disc. The additional force is often termed the added mass force because if the unsteadiness in the relative flow past a blade can be attributed not to flow turbulence but to an unsteady motion of the blade itself some of the air will be forced to move (accelerate) with the blade, effectively adding to the mass of the blade.

4.4.2 Adaptation of the acceleration potential method to unsteady flow

If the unsteady acceleration terms are added to equations (Equation 4.58), which are simplified to account for the induced velocities being very much smaller than the wind velocity, then those equations become

$$\begin{aligned}\rho \left(\frac{\partial u}{\partial t} + U_\infty \frac{\partial u}{\partial x} \right) &= -\frac{\partial p}{\partial x} \\ \rho \left(\frac{\partial v}{\partial t} + U_\infty \frac{\partial v}{\partial x} \right) &= -\frac{\partial p}{\partial y} \\ \rho \left(\frac{\partial w}{\partial t} + U_\infty \frac{\partial w}{\partial x} \right) &= -\frac{\partial p}{\partial z}\end{aligned}\tag{4.103}$$

As before, differentiating each equation with respect to its particular direction and adding together the results gives

$$\rho \left[\frac{\partial}{\partial t} \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) + U_\infty \frac{\partial}{\partial x} \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \right] = - \left(\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 p}{\partial z^2} \right)$$

but, for continuity of the flow,

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0$$

Therefore, the condition

$$\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 p}{\partial z^2} = 0$$

applies together with the Kinner pressure distributions of Section 4.3.2.

The accelerations ($\partial u/\partial t$, $\partial v/\partial t$, $\partial w/\partial t$) can be determined directly from Equations 4.103 without integration being necessary, but it is only the component $\partial u/\partial t$ that is required because it is normal to the rotor disc and so will give rise to a normal force.

The solutions for $\rho U_\infty(\partial u/\partial x) = -\partial p/\partial x$ have already been obtained in Sections 4.3.2, 4.3.3 and 4.3.4 and so it remains to determine the solutions for

$$\rho \frac{\partial u}{\partial t} = -\frac{\partial p}{\partial x} \quad (4.104)$$

Equation 4.104 cannot be solved for the velocity u because that is the solution of the complete equation that is the first of Equations 4.103. What can be determined from Equation 4.104 is the acceleration $\partial u/\partial t$ for which it is necessary to differentiate the chosen pressure distribution. The Kinner pressure distributions, that are solutions of Equations 4.62, are given as functions of the ellipsoidal co-ordinates (v , η , ψ), so to obtain the derivative with respect to x a co-ordinate transformation is required.

The relationships between the ellipsoidal co-ordinates and the Cartesian co-ordinates (x , y , z) are given in Equations 4.60 from which can be obtained the derivatives $\partial x/\partial v$, $\partial x/\partial \eta$, $\partial x/\partial \psi$ but what are really needed are the inverses of these derivatives.

We can find by appropriate differentiations of Equations 4.60

$$\frac{\partial}{\partial v} = \frac{\partial x''}{\partial v} \frac{\partial}{\partial x''} + \frac{\partial y'}{\partial v} \frac{\partial}{\partial y'} + \frac{\partial z}{\partial v} \frac{\partial}{\partial z},$$

for example, and so the complete Jacobian can be determined:

$$\begin{bmatrix} \frac{\partial}{\partial v} \\ \frac{\partial}{\partial \eta} \\ \frac{\partial}{\partial \psi} \end{bmatrix} = \begin{bmatrix} \frac{\partial x''}{\partial v} & \frac{\partial y'}{\partial v} & \frac{\partial z}{\partial v} \\ \frac{\partial x''}{\partial \eta} & \frac{\partial y'}{\partial \eta} & \frac{\partial z}{\partial \eta} \\ \frac{\partial x''}{\partial \psi} & \frac{\partial y'}{\partial \psi} & \frac{\partial z}{\partial \psi} \end{bmatrix} \begin{bmatrix} \frac{\partial}{\partial x''} \\ \frac{\partial}{\partial y'} \\ \frac{\partial}{\partial z} \end{bmatrix} \quad (4.105)$$

the inverse of which is what is required.

$$\begin{bmatrix} \frac{\partial}{\partial x''} \\ \frac{\partial}{\partial y'} \\ \frac{\partial}{\partial z} \end{bmatrix} = \begin{bmatrix} \frac{\partial v}{\partial x''} & \frac{\partial \eta}{\partial x''} & \frac{\partial \psi}{\partial x''} \\ \frac{\partial v}{\partial y'} & \frac{\partial \eta}{\partial y'} & \frac{\partial \psi}{\partial y'} \\ \frac{\partial v}{\partial z} & \frac{\partial \eta}{\partial z} & \frac{\partial \psi}{\partial z} \end{bmatrix} \begin{bmatrix} \frac{\partial}{\partial v} \\ \frac{\partial}{\partial \eta} \\ \frac{\partial}{\partial \psi} \end{bmatrix} \quad (4.106)$$

The Jacobian matrix of Equation 4.105 can be determined algebraically from Equations 4.60 and this can then be inverted algebraically to give the inverse Jacobian of Equation 4.106. From the inverse Jacobian it is found that

$$\frac{\partial}{\partial x''} = \frac{\eta(1 - v^2)}{R(\eta^2 + v^2)} \frac{\partial}{\partial v} + \frac{v(1 + \eta^2)}{R(\eta^2 + v^2)} \frac{\partial}{\partial \eta} \quad (4.107)$$

However, only the acceleration at the rotor disc itself is required and there the value of η is zero, so

$$\frac{\partial}{\partial x''} = \frac{1}{Rv} \frac{\partial}{\partial \eta} \quad (4.108)$$

If the corrected axi-symmetric pressure drop distribution of Equation 4.79 is chosen, to conform with the steady flow case then, for the whole flow field, the normalised pressure at any point is given by

$$\begin{aligned} p(v, \eta) = \frac{15}{32} v C_{TD} \left[-7\eta \tan^{-1} \frac{1}{\eta} + 4(1 - v^2) + 15v^2 \eta^2 \left(\eta \tan^{-1} \frac{1}{\eta} - 1 \right) \right. \\ \left. + 9\eta \left(\eta + (v^2 - \eta^2) \tan^{-1} \frac{1}{\eta} \right) \right] \end{aligned} \quad (4.109)$$

in which the pressure is normalised by $\frac{1}{2} \rho U_\infty^2$. The term C_{TD} is the contribution to the total thrust coefficient of the dynamic acceleration $\partial u / \partial t$. Note that, as explained at the end of Section 4.3.1, the pressure level just upwind of the rotor disc, as given by Equation 4.109, is half the magnitude of the pressure drop across the disc given by Equation 4.79.

By means of Equation 4.108, at the rotor plane, where $\eta = 0$, the pressure gradient is found to be

$$\frac{\partial p}{\partial x''} = \frac{1}{R} \frac{15\pi}{64} C_{TD} (9v^2 - 7)$$

Therefore, in terms of parameter μ , from Equation 4.104,

$$\rho \frac{\partial u}{\partial t} = - \frac{\partial p}{\partial x''} \frac{1}{2} \rho U_\infty^2 = \frac{1}{R} \frac{15\pi}{64} C_{TD} (9\mu^2 - 2) \frac{1}{2} \rho U_\infty^2 \quad (4.110)$$

It should be noted that the axial acceleration distribution is axi-symmetric and independent of the yaw angle.

The mean value of axial acceleration over the area of the disc is

$$\frac{\partial u_0}{\partial t} = \frac{75\pi}{256} \frac{U_\infty^2}{R} C_{TD} \quad (4.111)$$

The non-dimensional form of the acceleration can be expressed as

$$\frac{\partial a_o}{\partial \tau} = \frac{R}{U_\infty^2} \frac{\partial u_o}{\partial t} = \frac{75\pi}{256} C_{TD} \quad (4.112)$$

where $a_o = u_o/U_\infty$, axial flow factor and $\tau = tU_\infty/R$ which is called non-dimensional time.

The axial force on the disc is

$$T_x = \frac{1}{2} \rho U_\infty^2 \pi R^2 C_{TD}$$

Substituting for C_{TD} from Equation 4.112 gives

$$T_x = \frac{128}{75} \rho R^3 \frac{\partial u_o}{\partial t} \quad (4.113)$$

The added mass is, therefore, $\frac{128}{75} \rho R^3$

The added mass term associated with a solid disc given by (Tuckerman, 1925) is $\frac{8}{3}$, compared with $\frac{128}{75}$ given in Equation 4.113, and is in agreement with the value that is given by the uncorrected axi-symmetric Kinner pressure distribution of Equations 4.73. Although Pitt and Peters (1981) determine the value $\frac{128}{75}$, in subsequent papers by Peters and other workers the value $\frac{8}{3}$ is recommended and has come to be generally accepted. The use of the so-called 'corrected' pressure distribution for wind turbines has already been questioned in Section 4.3.3; there is no need to impose a zero pressure difference on the rotor disc at the rotation axis.

4.4.3 Unsteady yawing and tilting moments

For unsteady flow in yaw the normal unsteady acceleration distribution on the disc is required to have the same form of linear variation as the velocity, given in Equation 4.95. In terms of flow factors,

$$\frac{\partial a}{\partial \tau} = \frac{\partial a_0}{\partial \tau} + \frac{\partial a_s}{\partial \tau} \mu \sin \psi + \frac{\partial a_c}{\partial \tau} \mu \cos \psi \quad (4.114)$$

The condition that causes a yawing moment arises from the anti-symmetric pressure distribution of Section 4.3.4 can be obtained from Equations 4.87, 4.88. For the whole field surrounding the rotor disc the pressure distribution is

$$p(v, \eta, \psi) = 3D_2^1 v \sqrt{1-v^2} \left(3i\eta \sqrt{1+\eta^2} \tan^{-1} \frac{1}{\eta} - 3i\sqrt{1+\eta^2} + \frac{i}{\sqrt{1+\eta^2}} \right) \sin \psi$$

which, on the disc, produces the pressure shown in Figure 4.27. The coefficient D_2^1 is related to the yawing moment coefficient in Equation 4.92

$$iD_2^1 = -\frac{5}{4}C_{mz} \quad (4.92)$$

Therefore,

$$p(v, \eta, \psi) = -\frac{15}{4}\pi \cdot C_{mz} v \sqrt{1-v^2} \left(3\eta \sqrt{1+\eta^2} \tan^{-1} \frac{1}{\eta} - 3\sqrt{1+\eta^2} + \frac{1}{\sqrt{1+\eta^2}} \right) \sin \psi \quad (4.115)$$

As before, the pressure in Equation 4.115 is non-dimensionalised by the free-stream dynamic pressure $\frac{1}{2}\rho \cdot U_\infty^2$.

Applying the differential operator given in Equation 4.108 to Equation 4.115, from Equation 4.104 we get at the rotor disc, where $\eta = 0$,

$$\frac{\partial u_s}{\partial t} = \frac{45}{16}\pi \frac{U_\infty^2}{R} C_{mzD} \mu \sin \psi \quad (4.116)$$

In terms of non-dimensional time and velocity

$$\frac{\partial a_s}{\partial \tau} = \frac{45}{16}\pi C_{mzD} \quad (4.117)$$

Similarly, if there is a tilting moment then the corresponding acceleration is

$$\frac{\partial a_c}{\partial \tau} = \frac{45}{16}\pi C_{myD} \quad (4.118)$$

The radial variation is linear and so no linearisation adjustment is necessary as there is in the case of the velocity distribution. Again, the acceleration is independent of yaw angle. The mean acceleration is zero and so there is no coupling between the cases.

The relationship between accelerations and force coefficients is, therefore,

$$\begin{bmatrix} \frac{16}{3\pi} & 0 & 0 \\ 0 & \frac{16}{45\pi} & 0 \\ 0 & 0 & \frac{16}{45\pi} \end{bmatrix} \begin{bmatrix} \frac{\partial a_0}{\partial \tau} \\ \frac{\partial a_c}{\partial \tau} \\ \frac{\partial a_s}{\partial \tau} \end{bmatrix} = \begin{bmatrix} C_T \\ C_{my} \\ C_{mz} \end{bmatrix}_D \quad (4.119a)$$

or

$$[M] \left\{ \frac{\partial a}{\partial \tau} \right\} = \{C\}_D \quad (4.119b)$$

The complete equation of motion combines Equation 4.119 and the steady yaw Equation 4.101. The combination is achieved by adding the corresponding force coefficients, which means that both equations must be inverted.

$$[M] \left\{ \frac{\partial a}{\partial \tau} \right\} + [L]^{-1} \{a\} = \{C\}_D + \{C\}_S \quad (4.120)$$

The right hand side of Equation 4.120 can also be determined from blade element theory and will be a time dependent function of the axial induction factor. The blade forces will vary in a manner determined by the time-varying velocity of the oncoming wind and consequent dynamic structural deflections of the necessarily elastic rotor. Equation 4.120 applies to the whole rotor disc and the blade element forces need to be integrated along the blade lengths.

Numerical solutions to Equation 4.120 require a procedure for dealing with first order differential equations and the tried and tested fourth order Runge-Kutta method is recommended. Starting with a steady state solution the progress in time of the induced velocity as an unsteady flow passes through the rotor can be tracked. However, non-dimensionalising with respect to wind speed is not very useful if wind speed is changing dynamically and it is common to work directly in terms of induced velocity rather than flow factors.

Equation 4.120 really applies to the whole rotor and the only spatial variation of the induced velocity and acceleration that is permitted is as defined in Equations 4.95 and 4.114. However, a relaxation of the strict approach has been adopted by several workers, see, for example, Schepers and Snel (1995), where the induced velocities are determined for separate annular rings, as described in Section 4.1.8. The added mass term for an annular ring can be taken as a proportion of the whole added mass according to the appropriate acceleration distribution, Equations 4.112, 4.117 and 4.118.

Figure 4.29 shows measured and calculated flap-wise (out of the rotor plane) blade root bending moments for the Tjæreborg turbine caused by a pitch change from 0.070° to 3.716° with the reversed change 30 seconds later. The turbine was not in yaw and the wind speed was 8.7 m/s. The calculated results were made both according to the equilibrium wake method and with a differential equation method similar to that of Equation 4.120.

The comparison with the measured results clearly shows that the dynamic analysis predicts the initial overshoot in bending moment whereas the equilibrium wake method does not. Neither theory predicts the steady state bending moment achieved between the pitch changes. Figure 4.29 is taken from Lindenburg (1996), which describes the PHATAS III aero-elastic code developed at ECN in the Netherlands. Details of the Tjæreborg turbine, which is sited near Esbjerg in Denmark, can be obtained from Snel and Schepers (1995).

The solution procedure requires the time varying blade element force to determine the right hand side of Equation 4.120, but calculating the lift and drag forces on a blade element in unsteady flow conditions is not a straight forward process. The lift force on a blade element is dependent upon the circulation around the element but after a change in conditions the circulation takes time to settle at a new level (see Section 4.5.3) and in the interim the instantaneous lift cannot be determined via the instantaneous angle of attack. In a continuously changing situation the lift is not in phase with the angle of attack and does not have a magnitude that can be determined using static, two-dimensional aerofoil lift versus angle of attack data.

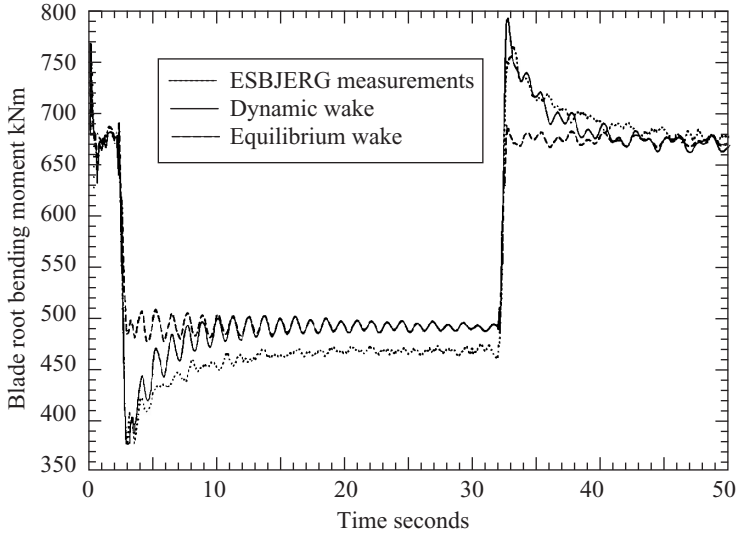


Figure 4.29 Measured and calculated blade root bending moment responses to blade pitch angle changes on the Tjæreborg turbine (from Lindenburg, 1996)

4.5 Quasi-steady aerofoil aerodynamics

4.5.1 Introduction

When the oncoming flow relative to an aerofoil is unsteady the angle of attack is continuously changing and so the lift also is changing with time. The simple, but incorrect, way of dealing with this problem is to assume that the instantaneous angle of attack corresponds to the same lift coefficient as if that angle of attack were to be constantly applied. The angle of attack is determined by the oncoming flow velocity and the velocity of the blade's motion. If the blade motion includes a torsional (pitching) component then the angle of attack will vary along the chord length: thin aerofoil theory (see J.D. Anderson, 1991) shows that the point $\frac{3}{4}$ of the chord length from the leading edge is where the angle of attack must be determined.

The velocities that determine the quasi-steady angle of attack for a rotor blade element are shown in Figure 4.30, the dot represents differentiation with respect to time t .

The flow velocity $W(t)$, which includes the rotational speed of the blade element, varies in magnitude and direction $\alpha_w(t)$ with the unsteady wind. $W(t)$ also includes the induced velocities caused by the rotor disc as might be determined by Equation 4.120. The elastic deflection velocities (subscript e) caused by blade vibration also influence the quasi-steady angle of attack, which is

$$\alpha(t) = \alpha_w(t) - \left(v_e(t) - \frac{\partial \beta_e}{\partial t} \left(\frac{3}{4} - h \right) c \right) \frac{1}{W(t)} \quad (4.121)$$

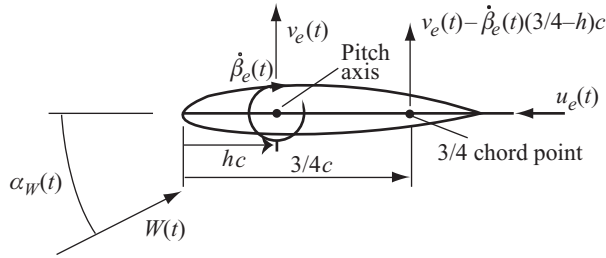


Figure 4.30 Unsteady flow and structural velocities adjacent to a rotor blade

The structural velocity caused by chord-wise (edge-wise) deflections of the blade will also influence the angle of attack but by a very small amount. The non-dimensional parameter h defines the position of the pitching axis (flexural axis, shear centre position) of the blade element.

Assuming the structural deflection velocities to be small the lift force is then

$$L_c(t) = \frac{1}{2} \rho W^2(t) \frac{dC_l}{d\alpha} \sin \alpha(t) \quad (4.122)$$

The lift-curve slope $dC_l/d\alpha$ is assumed to be the same as for the static case.

4.5.2 Aerodynamic forces caused by aerofoil acceleration

In addition to the circulatory forces there are forces on the aerofoil caused by the inertia of the surrounding air that is accelerated as the aerofoil accelerates in its motion. The additional terms are added mass forces. The added mass per unit span of blade can be shown to be that of a circular cylinder of air of diameter equal to the aerofoil chord $(\pi c^2/4)\rho$. There are two components to the added mass force, see Fung (1969).

1. A lift force with the centre of pressure at the mid-chord point of an amount equal to the apparent mass times the normal acceleration of the mid-chord point.

$$L_{m1}(t) = -\frac{1}{4} \pi c^2 \rho \left(\frac{\partial v_e}{\partial t} - c \left(\frac{1}{2} - h \right) \frac{\partial^2 \beta_e}{\partial t^2} \right) \quad (4.123)$$

2. A lift force with the centre of pressure at the $\frac{3}{4}$ chord point, of the nature of a centrifugal force, of an amount equal to the apparent mass times $W(t)(\partial \beta_e / \partial t)$

$$L_{m2}(t) = -\frac{1}{4} \pi c^2 \rho W(t) \frac{\partial \beta_e}{\partial t} \quad (4.124)$$

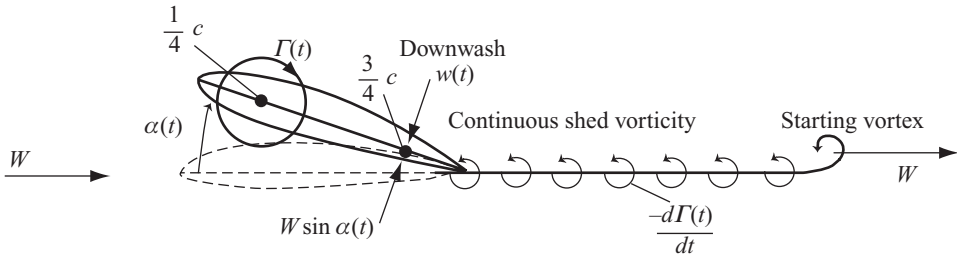


Figure 4.31 Wake development after an impulsive change of angle of attack

There is also a nose-down pitching moment equal to the apparent moment of inertia $(\pi/128)c^4\rho$ (which, actually, is only a quarter of the moment of inertia per unit length of the cylinder of air of diameter c) times the pitching acceleration $\partial^2\beta_e/\partial t^2$

$$M_m = \frac{\pi}{128}c^4\rho\frac{\partial^2\beta_e}{\partial t^2} \quad (4.125)$$

The added masses are determined by a process similar to that of Section 4.4.2.

4.5.3 The effect of the wake on aerofoil aerodynamics in unsteady flow

If the angle of attack of the flow relative to an aerofoil changes, the strength of the circulation also changes, but the process is not instantaneous because the circulation can only develop gradually. To determine how the lift on an aerofoil actually develops with time after an impulsive change of angle of attack occurs it is necessary to include the wake in the analysis. The sudden change of α causes a build up of circulation around the aerofoil that is matched by an equal and opposite vorticity being shed into the wake.

The bound circulation on an aerofoil is actually distributed along the chord but, for simplicity, can be assumed to be a concentrated vortex Γ at the aerodynamic centre ($\frac{1}{4}$ chord point). In steady flow conditions, the velocity induced by the vortex, normal to the chord-line, at the $\frac{3}{4}$ chord point is exactly equal and opposite to the component of the flow velocity normal to the chord-line. The two opposed velocities ensure that no flow passes through the aerofoil at the $\frac{3}{4}$ chord point, a condition which, of course, must be true everywhere along the chord-line but the $\frac{3}{4}$ chord point is used as a control point. The simplified situation assumes that the aerofoil can be represented geometrically by its chord-line, this is known as thin aerofoil representation.

In unsteady flow conditions the velocity induced at the $\frac{3}{4}$ chord point (often referred to as downwash) is caused jointly by the bound vortex and the wake vorticity, see Figure 4.31, but must still be equal and opposite to the component of the flow velocity normal to the chord-line.

After the impulsive change of angle of attack there is a sudden change in upwash ($W \sin \alpha$) which must be matched by a sudden change of downwash. The change of upwash implies an impulsive acceleration of the mass of the air that causes an added mass force on the aerofoil. The sudden change of downwash must come from a sudden increase in circulation that must be matched by an equal and opposite starting vortex being shed into the wake and then convects downstream. The influence of the starting vortex on the downwash gets gradually weaker as

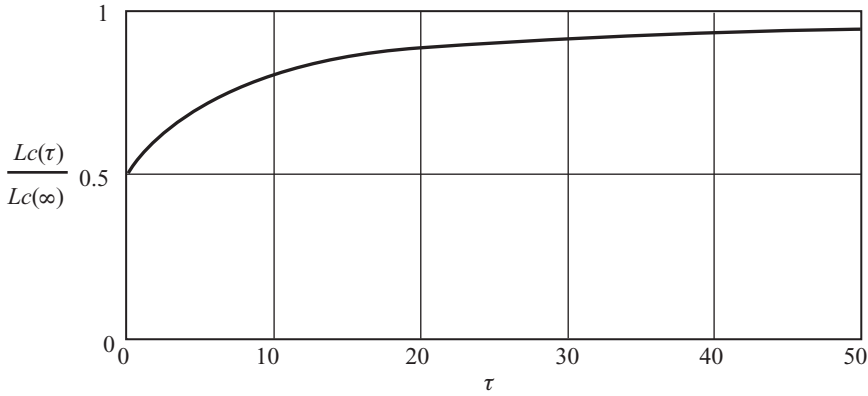


Figure 4.32 Lift development after an impulsive change of angle of attack

the vortex moves away so the bound vortex must increase in strength to maintain that the downwash matches the upwash. The increasing strength of the bound vortex means that, to keep the overall angular momentum contained in the vorticity zero (there was none before the impulse), continuous vorticity of the opposite sense must be shed into the wake and this also contributes to the downwash.

The rate of increase of the bound vortex strength gradually reduces with a corresponding reduction of the strength of the shed vorticity and eventually, asymptotically, the steady state bound circulation strength is developed.

The analytical solution to the problem was developed by Wagner (1925), it is complex and expressed in terms of Bessel functions but several approximations to the Wagner function exist, the most accurate of which is given by Jones (1945).

$$\frac{\Delta L_c(\tau)}{\frac{1}{2}\rho W^2 c \frac{dC_l}{d\alpha} \sin(\Delta\alpha)} = \Phi(\tau) = 1 - 0.165e^{-0.0455\tau} - 0.335e^{-0.30\tau} \quad (4.126)$$

where $\tau = 2Wt/c$ is the non-dimensional time based upon the half chord length $c/2$ of the aerofoil and $dC_l/d\alpha$ is the slope of the static lift versus angle of attack characteristic of the aerofoil. Non-dimensional time τ can also be regarded as the number of half-chord lengths travelled down-stream by the starting vortex after a time t has elapsed since the impulsive change of angle of attack. Equation 4.126 is an example of an indicial equation.

Figure 4.32 shows the progression of the growth of the lift as time proceeds from the original impulsive change of angle of attack. The added mass lift gradually dies away as the circulatory lift develops. Eventually, the steady state, full circulatory lift is achieved.

In the situation where the angle of attack is continuously changing, which is the case for the wind turbine blade, the circulation never reaches an equilibrium state and the added mass lift never dies away.

In an unsteady wind, if for each change of wind velocity over an increment of time δt there is a corresponding impulsive angle attack change then the lift on the aerofoil will subsequently be influenced by that change in the manner of Equation 4.126 and Figure 4.32. The accumulation of all such changes in a continuous manner will determine the unsteady lift force on the aerofoil.

Assume the flow has been in progress for a long time t_0 and let t be any time prior to t_0 , the lift at time t is then given by

$$L_c(\tau) = L_c(0) + \frac{1}{2}\rho \frac{dC_l}{d\alpha} c \int_0^\tau W(\tau_0)\Phi(\tau - \tau_0) \frac{dw(\tau_0)}{d\tau_0} d\tau_0 \quad (4.127)$$

Where $\delta w = (dw(\tau_0)/d\tau_0)\delta\tau_0$ is the change in upwash (downwash) determined by the change in $W(\tau)$ and the changes in blade motion during the time interval.

The use of non-dimensional time in the above equation poses a problem and it is more convenient to use actual time in a numerical integration.

Theodorsen (1935) solved Equation 4.127 for the case of an aerofoil oscillating sinusoidally in pitch and heave (flapping motion) at fixed frequency ω and immersed in a steady on-coming wind U . The unsteady lift on the aerofoil is also sinusoidal but not in phase with the angle of attack variation, nor is the amplitude of the lift variation related to the amplitude of the angle of attack by static aerofoil characteristics.

Theodorsen's solution shows that the circulatory lift on the aerofoil equals the quasi-steady lift of Equation 4.121 multiplied by Theodorsen's function $C(k)$ that has both real and imaginary parts. $k = \omega c/2U$ is called the *reduced frequency* and $\omega t = k\tau$. In addition there is the added mass lift given by Equations 4.123 and 4.124.

$$C(k) = \frac{1}{1 + A(k)} = \frac{1}{1 + \left(\frac{Y_0(k) + iJ_0(k)}{J_1(k) - iY_1(k)} \right)} \quad (4.128)$$

where $J_n(k)$ and $Y_n(k)$ are Bessel functions of the first and second kind, respectively, and n is an integer. Like the Legendre polynomials the Bessel functions are the solutions to a second order ordinary differential equation called Bessel's equation,

$$k^2 \frac{d^2 y}{dk^2} + k \frac{dy}{dk} + (k^2 - n^2) = 0$$

Unlike the Legendre polynomials, the Bessel functions cannot be expressed in closed form but only as an infinite series.

Theodorsen's function is often divided into two functions, one describing the real part and the other the imaginary part.

$$C(k) = F(k) + iG(k) \quad (4.129)$$

From Jones' approximation to the Wagner function, Equation 4.126, an approximation to Theodorsen's function is obtained

$$C(k) = 1 - \frac{0.165}{1 - i \frac{0.0455}{k}} - \frac{0.335}{1 - i \frac{0.30}{k}} = F(k) + iG(k) \quad (4.130)$$

The exact and approximate parts of $C(k)$ are shown in Figures 4.33a and 4.33b.

The real part of $C(k)$ gives the lift that is in phase with the angle of attack defined in Equation 4.122 and the imaginary part gives the lift that is 90° out of phase with the angle of attack.

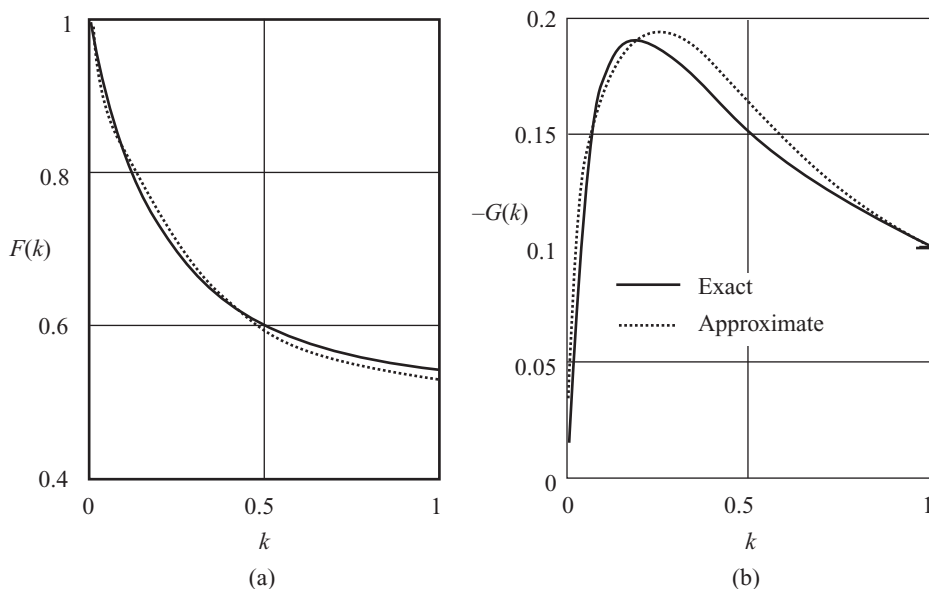


Figure 4.33 The real (a) and imaginary (b) parts of Theodorsen's function

The drawback of the Theodorsen function for rotor blade application is that the wake streams away from the blade in a straight line whereas the rotor blade wake is helical and the wakes of other blades will also be present. Loewy (1957) developed a theory for a rotor blade that accounts for the repeated wake in a similar manner to Prandtl (see Section 3.8.3). As Theodorsen had done, Loewy used two-dimensional, thin aerofoil theory and produced a modification to Theodorsen's function. In Equation 4.128 the Bessel function of the first kind $J_n(k)$ is multiplied by $(1 + W(k))$ where $W(k)$ is called the Loewy wake-spacing function.

$$W(k) = \frac{1}{e^{(\frac{2d}{c}k + i2\pi)} - 1} \quad (4.131)$$

d is the wake spacing defined in Equation 3.77 and c is the chord of the aerofoil.

Miller (1964) arrived at a very similar result to Loewy by using a discrete vortex wake model.

Loewy's and Miller's theories apply only to the non-yawed rotor but Peters, Boyd and He (1989) have developed a much more extensive theory based upon the method of acceleration potential. A sufficient number of Kinner pressure distributions are required to model both the radial and azimuthal pressure distribution on a helicopter rotor such that the pressure spikes of individual blades are present. The theory obviates the use of blade element theory and includes automatically unsteady effects and tip-losses: modelling of the blade geometry by this method does present some problems, however. Suzuki and Hansen (1999) have applied the theory of Peters, Boyd and He to wind turbine rotors and make comparisons with the blade-element/momentum theory. Van Bussel's theory (1995) is very similar to that of Peters, Boyd and He but is intended for application to wind turbines.

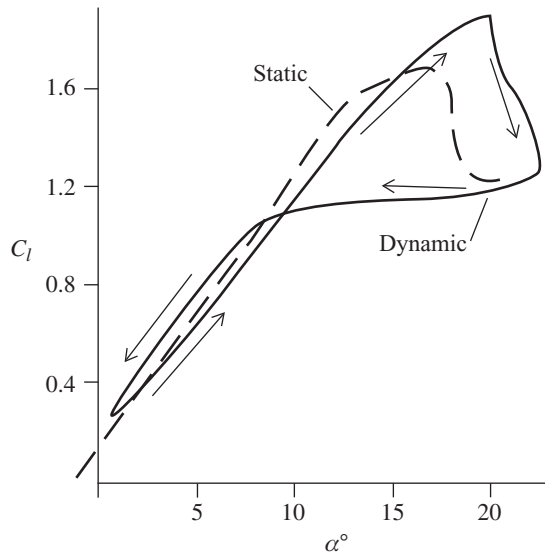


Figure 4.34 Typical dynamic stall behaviour

4.6 Dynamic stall

At high wind speeds, and often when a rotor is yawed to the wind, all, or part, of a blade will have separated flow on the downwind surface, that is, the flow will be stalled. Because of unsteadiness in the ambient flow, or because of the changing angle of attack that occurs with a yawed rotor, the flow about a blade may go into and out of stall. In such circumstances the nature of the stalling process is dynamic and experience shows that it is significantly different to so-called static stall. Actually, the very process of stalling is always dynamic.

In the case of static, leading-edge stall an increase in angle of attack beyond the stall angle initially gives rise to an adverse pressure gradient just behind the leading edge on the upper surface of the aerofoil sufficient to cause separation. The separation is not complete over the upper surface instantaneously. The separated flow forms a vortex which moves towards the trailing edge. While the vortex is still above the aerofoil the flow on the upper surface upstream of the vortex is separated but downstream the flow remains attached. Viscosity causes the vortex to dissipate rapidly and, although the low pressure in the vicinity of the vortex maintains some lift on the aerofoil, by the time the vortex reaches the trailing edge and leaves the aerofoil the stall is complete and the circulation has fallen. The process is transient. The pressure distribution on the aerofoil changes dramatically because there is a rear-ward movement of the centre of pressure causing a rise in the nose-down pitching moment and a rise in pressure drag.

If the angle of attack is changing continuously as the static stall angle is reached, during the finite time for the separated vortex to progress towards the trailing edge, the angle of attack is still increasing causing a further increase in lift and in the strength of the vortex, see Figure 4.34. Lift can, therefore, rise to values well above the static stall level. Once the vortex has passed the trailing edge the lift falls suddenly, even though the angle of attack may still be

increasing. Once the flow is fully stalled and with the angle of attack now decreasing the lift remains low and fairly constant until re-attachment of the flow. Re-attachment does not take place until the angle of attack is significantly lower than the static stall level.

Dynamic stall will occur on a wind turbine when the rotor is yawed and at a low tip speed ratio (high wind speed), when the rotor encounters a gust and on emerging from tower shadow. The loads experienced by a blade during dynamic stall can be large and can cause significant fatigue damage.

Various models have been developed to predict dynamic stall notably those of Gormont (1973) and Beddoes (1975). More recently Leishman and Beddoes (1989) have improved the original theory of Beddoes and this is now the preferred method for use with wind turbines and helicopter rotors. A study of the dynamic stall behaviour of a NREL wind turbine aerofoil is given by Gupta and Leishman (2006). A report from the Risø National Laboratory in Denmark by Hansen, Gaunaa and Madsen (2004) adapts the Leishmann and Beddoes model for wind turbines.

4.7 Computational fluid dynamics

The methods for analysing the flow through a wind turbine in various conditions developed in Chapter 3 and this chapter are all simplifications necessary to facilitate the calculations; to obtain accurate solutions to the flow conditions a much more complex method is required.

The analysis of the flow approaching a turbine rotor can be undertaken, with little loss of accuracy, by using the equations of potential flow known as the Euler equations, developed in the eighteenth century, and given in Section 4.4.2 as Equation 4.103. The Euler equations, together with the continuity equation, can form the basis of a numerical procedure to obtain a solution to the flow conditions. However, during the course of the flow through the rotor, and in the wake, the Euler equations are no longer adequate because they cannot deal with boundary layer flow close to a blade surface and with the separated flow conditions in the wake. In the nineteenth century the fully viscous flow equations of Navier and Stokes were developed and are used today to predict turbulent flows, a method generally known as computational fluid dynamics (CFD). The additional terms introduced in the Navier-Stokes equations include the effect of viscosity and are basically given by Newton's theory of viscous flow – Equation A3.1, in the Appendix to Chapter 3.

CFD is essentially a numerical solution of the equations. For the analysis of a wind turbine rotor the area (volume) around is divided into a mesh, two or three dimensional. At the rotor surfaces the mesh needs to be very fine in order to model attached flow boundary layers whereas in the wake a coarser mesh will suffice. Especial care needs to be taken in regions of high shear such as the wake boundary and close to the vicinity of shed vortices. The number of unknowns to be solved in most three dimensional problems is enormous and, because the solution process is iterative, the solution times are of long duration. The wind turbine rotor requires a further complication because the mesh must rotate but still, at each stage, match the non-rotating mesh that is used at a sufficient distance. Much of the skill and effort required to carry out an analysis is invested in the mesh generation. A principal advantage of the CFD method for wind turbine blades is that no experimentally based aerofoil data is required because the method calculates the flow conditions surrounding a blade surface.

CFD methods are currently used mainly for research but they are used by wind turbine designers to confirm the analyses using simpler, faster methods.

CFD packages called Ellipsys 3D and Ellipsys 2D have been specifically developed at Risø for application to wind turbines: Sørensen (2002) and Bertagnolio, Sørensen and Johansen (2006) give instructive accounts of the results that are achievable.

Commercial CFD packages can be used for wind turbine applications but they require the specialist skills of the wind turbine aerodynamicist and it is often found to be difficult to achieve both the high Reynolds number attached flow conditions and, elsewhere in the wake, the large eddy conditions.

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Design loads for horizontal axis wind turbines

5.1 National and international standards

5.1.1 Historical development

The preparation of national and international standards containing rules for the design of wind turbines began in the 1980s. The first publication was a set of regulations for certification drawn up by Germanischer Lloyd in 1986. These initial rules were subsequently considerably refined as the state of knowledge grew, leading to the publication by Germanischer Lloyd of the 'Regulation for the Certification of Wind Energy Conversion Systems' in 1993. Revised editions were published in 1999 and 2003. Meanwhile national standards were published in The Netherlands (NEN 6096) and Denmark (DS 472) in 1988 and 1992 respectively.

The International Electrotechnical Commission (IEC) began work on the first international standard in 1988, leading to the publication of IEC 1400–1 'Wind Turbine Generator Systems – Part 1: Safety Requirements' in 1994. Second and third editions, each containing some significant changes and bearing the new number IEC 61400–1, appeared in 1999 and 2005 respectively. IEC 61400–1 has now superseded the national standards referred to above.

The following sections describe the scope of the IEC 61400–1 and Germanischer Lloyd requirements in outline.

5.1.2 IEC 61400–1

IEC 61400–1 'Wind Turbines – Part 1: Design Requirements' identifies three different classes of wind turbines to suit differing site wind conditions, with increasing class designation number corresponding to reducing average and extreme wind speeds. The wind speed parameters for each class are given in Table 5.1, where the reference wind speed is defined as the 10 minute mean wind speed at hub height with a 50-year return period. Rigorous procedures

Table 5.1 Wind speed parameters for wind turbine classes

	Class I	Class II	Class III
Reference wind speed, U_{ref} (m/s)	50	42.5	37.5
Annual average wind speed, U_{ave} (m/s)	10	8.5	7.5
50-year return gust speed, $1.4U_{\text{ref}}$ (m/s)	70	59.5	52.5
1-year return gust speed, $1.12U_{\text{ref}}$ (m/s)	56	47.6	42.0

are laid down for demonstrating that the wind conditions at a particular wind turbine site conform to those of the designated wind turbine class. To allow for sites where conditions do not conform to any of these classes, a fourth class (Class S) is provided, in which the basic wind speed parameters are to be specified by the manufacturer.

The standard identifies a total of 22 different load cases (17 ultimate, 5 fatigue), which, as a minimum, require consideration in the design of the turbine. Each load case is defined in terms of a different combination of wind conditions and machine state – for example, extreme wind shear during power production. The standard does not extend to the prescription of particular methods of loading analysis.

Subsequent sections cover requirements for the control and protection systems, the various mechanical systems, the electrical system, installation, commissioning, operation and maintenance.

5.1.3 GL rules

Germanischer Lloyd’s ‘Guidelines for the Certification of Wind Turbines’, commonly referred to as the GL rules, adopts the same classification of wind turbines as IEC 61400–1, apart from the addition of a low wind speed Class IV with a reference wind speed of 30 m/s. A larger number of load cases are specified, but most of them parallel cases in IEC 61400–1.

The GL rules then go on to describe the design processes required for each component of the turbine in turn – beginning with the blades and ending with the foundation. This includes design load definition, analysis methods, material strengths and fatigue properties. The level of detail provided here sets the GL rules apart from the IEC standard, and is a consequence of their role in defining the design documentation required for certification.

There is a rigorous treatment of the requirements for the control and safety systems, and for the associated protection and monitoring devices. The centrality of these systems to the overall design process is emphasised by placing them at the start of the document. Final sections deal with lightning protection, operation and maintenance manuals, and wind turbine testing.

5.2 Basis for design loads

5.2.1 Sources of loading

The sources of loading to be taken into account may be categorised as follows:

- Aerodynamic loads.
- Gravitational loads.

- Inertia loads (including centrifugal and gyroscopic effects).
- Operational loads arising from actions of the control system (e.g. braking, yawing, blade pitch control, generator disconnection).

5.2.2 Ultimate loads

The load cases selected for ultimate load design must cover realistic combinations of a wide range of external wind conditions and machine states. It is common practice to distinguish between normal and extreme wind conditions on the one hand, and between normal machine states and fault states on the other. The load cases for design are then chosen from:

- Normal wind conditions in combination with normal machine states.
- Extreme wind conditions in combination with normal machine states.
- Machine fault states in combination with appropriate wind conditions.

Extreme wind conditions are generally defined in terms of the worst condition occurring with a 50-year return period. It is assumed that machine fault states arise infrequently, and are uncorrelated with extreme wind conditions, so that the occurrence of a machine fault in combination with the 50-year return wind condition is an event with such a high return period that it need not be considered as a load case. However, IEC 61400–1 wisely stipulates that if there *is* some correlation between an extreme external condition and a fault state, then the combination should be considered as a design case.

5.2.3 Fatigue loads

A typical wind turbine is subjected to a severe fatigue loading regime. The rotor of a 2 MW machine will rotate some 10^8 times during a 20-year life, with each revolution causing a complete gravity stress reversal in the low speed shaft and in each blade, together with a cycle of blade out-of-plane loading due to the combined effects of wind shear, yaw error, shaft tilt, tower shadow and turbulence. It is, therefore, scarcely surprising that the design of many wind turbine components is often governed by fatigue rather than by ultimate load.

The design fatigue load spectrum should be representative of the loading cycles experienced during power production over the full operational wind speed range, with the numbers of cycles weighted in accordance with the proportion of time spent generating at each wind speed. For completeness, load cycles occurring at start-up and shut-down, and, if necessary, during shut-down, should also be included.

It is generally assumed that the extreme load cases occur so rarely that they will not have a significant effect on fatigue life.

5.2.4 Partial safety factors

Partial safety factors for loads

Limit state design requires characteristic loads to be multiplied by appropriate partial safety factors when calculating the design loads. Although, traditionally, different partial safety factor values have been assigned to different kinds of load in static analyses, IEC 61400–1 Edition 3 specifies a single partial safety factor for aerodynamic, operational, gravity and

Table 5.2 Partial safety factors for loads, IEC 61400–1 Edition 3

Unfavourable loads			Favourable loads
Class of design load case			
Normal 1.35*	Abnormal 1.1	Transport & erection 1.5	0.9

*Exceptionally, the partial factor for Design Load Case 1.1 (see Section 5.4.1) is set at 1.25, because, in this case, the loads are determined using statistical load extrapolation

inertia loads for each class of load case. This avoids the pitfalls inherent in any attempt to formulate equations of motions for a dynamic analysis in which different terms have been distorted relative to one another by the application of different load factors.

Ultimate load cases are divided into three classes, normal, abnormal and transport/erection, with a different partial safety factor for each, as set out in Table 5.2. Most load cases are assigned to the normal class, with the abnormal class reserved for the more unlikely fault conditions.

The partial factor for fatigue loads is unity.

Partial safety factors for the consequences of failure

In addition to the partial safety factors for loads and materials intrinsic to limit state design, IEC 61400–1 also specifies the use of a partial safety factor for the consequences of failure, which varies according to the nature of the component under consideration. Three classes of component are identified, as follows:

- Component class 1 – used for ‘fail-safe’ structural components whose failure does not result in the failure of a major part of a wind turbine.
- Component class 2 – used for ‘non fail-safe’ structural components whose failure may lead to the failure of a major part of a wind turbine.
- Component class 3 – used for ‘non fail-safe’ mechanical components that link non-redundant actuators and brakes required for turbine protection to main structural components.

Recommended minimum values for the partial safety factor for the consequences of failure are given in Table 5.3 below.

Table 5.3 Partial safety factors for the consequences of failure, IEC 61400–1 Edition 3

Type of strength assessment	Ultimate strength	Fatigue strength
Component class 1	0.9	1.0
Component class 2	1.0	1.15
Component class 3	1.3	1.3

IEC 61400–1 requires the partial safety factor for the consequences of failure to be introduced at the stage of assessing component design strength, but as it could equally well have been applied in the derivation of the design load, it is instructive to introduce the concept here.

5.2.5 Functions of the control and safety systems

A primary function of the control system is to maintain the machine operating parameters within their normal limits. The purpose of the safety system (referred to as ‘protection system’ in IEC 61400–1) is to ensure that, should a critical operating parameter exceed its normal limit as a result of a fault or failure in the wind turbine or the control system, the machine is maintained in a safe condition. Normally the critical operating parameters are:

- Turbine rotational speed.
- Power output.
- Vibration level.
- Twist of pendant cables running up into nacelle.

For each parameter it is necessary to set an activation level at which the safety system is triggered. This has to be set at a suitable margin above the normal operating limit to allow for overshooting by the control system, but sufficiently far below the maximum safe value of the parameter to allow scope for the safety system to rein it in. The rotor speed at which the safety system is activated is a key input to the design load case involving rotor overspeed.

5.3 Turbulence and wakes

Fluctuation of the wind speed about the short-term mean, or turbulence, naturally has a major impact on the design loadings, as it is the source of both the extreme gust loading and a large part of the blade fatigue loading. The latter is exacerbated by the gust slicing effect, in which a blade will slice through a localised gust repeatedly in the course of several revolutions.

The nature of free-stream turbulence, and its mathematical description in statistical terms, form the subject of Section 2.6. IEC 61400–1 Edition 3 addresses the variation in turbulence intensity from site to site arising from different terrain types by defining three turbulence categories, according to the expected value, I_{15} , of the hub height turbulence intensity at a reference mean wind speed, $\bar{U}$, of 15 m/s. These are categories A, B and C for I_{15} values of 0.16, 0.14 and 0.12 respectively.

Measurements have shown (see, for example, Risø paper R-1111 (1999) by Larsen *et al*) that there is significant variability in turbulence intensity at a particular site, even at a particular mean wind speed. Accordingly, IEC 61400–1 specifies that the design value is to be taken as the 90% quantile (i.e. the value with a 10% exceedance probability), defined as

$$I_u = \sigma_u / \bar{U} = I_{15} (0.75 + 5.6 / \bar{U}) \quad (5.1)$$

where σ_u is the standard deviation of the turbulent wind speed fluctuations, $\bar{U}$ is the hub height mean wind speed and I_{15} is defined above. This relationship results in reducing turbulence

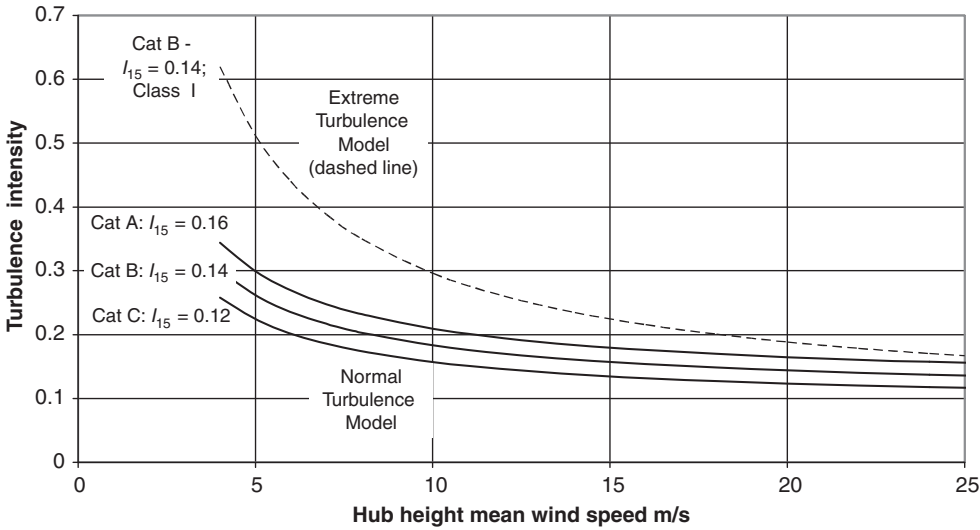


Figure 5.1 Variation of turbulence intensity with wind speed for the normal and extreme turbulence models

intensity with increasing wind speed, as illustrated in Figure 5.1, and is termed the ‘Normal Turbulence Model’. Note that σ_u does not vary with height, so I_u reduces with increasing height, because of wind shear.

There is also a requirement to consider the *maximum* turbulence expected to occur, in load case 1.3 – see Section 5.4.1. This is defined by the extreme turbulence model, in which the turbulence intensity is given by

$$I_u = \sigma_u / \overline{U} = I_{15} (0.036 (U_{ave} + 6) (1 - 8/\overline{U}) + 20/\overline{U}) \tag{5.2}$$

where U_{ave} is the hub height annual average wind speed in m/s. This relationship is shown by the dashed line on Figure 5.1.

For any particular candidate wind farm site, ambient turbulence levels have to be determined from site wind speed measurements and used to derive an estimate of the augmented turbulence levels including wake effects at each turbine location – for example, using the method suggested in IEC 61400–1 Annex D. An appropriate design turbulence category (A, B or C) may then be identified by showing that the 90% quantile of the estimated turbulence intensity including wake effects is less than the 90% quantile value given by Equation 5.1 for that category, for all wind speeds between 60% of turbine rated speed and the cut-out speed.

Computer simulation of the turbulent wind field also requires the definition of the power spectra of the fluctuations of the three orthogonal velocity components and their spatial correlation, which comprise the turbulence model. Edition 2 of the standard details two spectra – those due to Von Karman (1948) and Kaimal (1972) – and their corresponding coherence functions (see Sections 2.6.4 and 2.6.7). However, the Von Karman spectrum is omitted from Edition 3 and replaced by the Mann (1994) uniform shear turbulence model (see Section 2.6.8), although the use of the Von Karman spectrum is still permitted, as is the use of other spectra, provided the power spectral density is asymptotic to $n^{-5/3}$ at high frequencies.

5.4 Extreme loads

5.4.1 Operational load cases

A variety of load cases have to be investigated in this category, so that the effects of extremes of gust loading, wind direction change and wind shear – with or without faults – can be evaluated in turn. The IEC 61400–1 load cases can be divided into two distinct types, depending on whether the wind field is specified in deterministic or stochastic terms. In the deterministic load cases, simple mathematical expressions are used to define the wind speed variation over time, direction changes and wind shears. In the stochastic load cases the statistical properties of the wind are defined and the cases have to be analysed using repeated time domain simulations of the wind field incident on the rotor.

The deterministic load cases have the merit of simplicity, but are open to the criticism that they fail to model the behaviour of the real wind accurately. Accordingly, the number of load cases defined by deterministic discrete gusts or sudden direction changes has been reduced in IEC 61400–1 Edition 3 compared with earlier editions, in favour of greater reliance on cases requiring simulation of the turbulent wind field. In the longer term it may be possible to eliminate more deterministic load cases in favour of stochastic ones, by constraining the turbulent wind simulations to model a particular gust profile – see Section 5.4.3.

IEC 61400–1 Edition 3 requires account to be taken of the following in all operational load cases:

- Wind shear according to the power law $U(z) \propto z^{0.2}$, which is termed the ‘normal wind profile model’.
- Tower shadow (described in Section 5.7.2).
- Inclination of the mean air flow of up to 8° with respect to the horizontal plane.
- Rotor aerodynamic imbalance (e.g. due to blade pitch and twist deviations) and rotor mass imbalance.
- Yaw tracking errors.

Air density is to be taken as 1.225 kg/m^3 .

The individual ultimate load cases defined in Edition 3 of IEC 61400–1 are described below in turn. Note that, except where indicated otherwise, the full range of mean wind speeds between cut-in and cut-out are to be investigated in each case. (The annotated acronyms in capitals are those used by the code to identify the different wind conditions.)

Power production load cases – normal machine state

Load case 1.1: Operation in turbulent wind field defined by the normal turbulence model [NTM] – see Section 5.3. Wind speeds at 2 m/s intervals between cut-in wind speed and cut-out wind speed, U_o , to be investigated. Normal partial load factor (exceptionally 1.25).

A minimum of 15 10-minute simulations are required for each wind speed between 2 m/s below the rated wind speed, U_r , and the cut-out wind speed. (The rated wind speed is defined as the uniform, steady wind speed at which the turbine’s rated power is reached.) The characteristic load, that is, that having a 50-year recurrence period, in each element of

the structure is to be determined by statistical extrapolation of the extreme value distribution. See Section 5.14.

[Load case 1.2 is a fatigue load case]

Load case 1.3: Operation in turbulent wind field defined by the extreme turbulence model [ETM]. Normal partial load factor.

As this load case is intended to capture the loading arising from the *maximum* anticipated turbulence intensity, no extrapolation is required. In practice the case will rarely be critical, as the variability in loading resulting at ‘typical’ turbulence intensities over the long term is likely to be greater than that resulting from an extreme value of turbulence intensity over a short period.

Load case 1.4: Gust and direction change [ECD]. Hub height wind speed equal to the rated wind speed, U_r , ± 2 m/s plus a 15 m/s rising gust, in conjunction with a simultaneous wind direction change of $720/U_r^\circ$, that is, 60 degrees for a rated wind speed of 12 m/s, for example. The gust rise time and the period over which the direction change takes place are both specified as ten seconds. Normal partial load factor.

Load case 1.5: Extreme wind shear [EWS]. Additional vertical or horizontal transient wind shear superimposed on the ‘normal wind profile model’. Normal partial load factor. The additional wind shears are specified as:

$$\left(\frac{z - z_{\text{hub}}}{D}\right) \left(2.5 + 0.2\beta\sigma_u \left(\frac{D}{\Lambda_1}\right)^{0.25}\right) \left(1 - \cos\left(\frac{2\pi t}{T}\right)\right) \text{ m/s for } 0 < t < T, \text{ for vertical shear} \quad (5.3a)$$

$$\left(\frac{y}{D}\right) \left(2.5 + 0.2\beta\sigma_u \left(\frac{D}{\Lambda_1}\right)^{0.25}\right) \left(1 - \cos\left(\frac{2\pi t}{T}\right)\right) \text{ m/s for } 0 < t < T, \text{ for horizontal shear} \quad (5.3b)$$

where

z is the height above ground

y is the lateral co-ordinate with respect to the hub

D is the rotor diameter

$\beta = 6.4\sigma_u$ is as defined in Section 5.3

Λ_1 is the longitudinal turbulence scale parameter of $0.7z_{\text{hub}}$, or 42 m, whichever is the lesser

T is the duration of the transient wind shear, set at 12 seconds.

The two shears are to be applied independently as separate cases, not simultaneously. In the case of a 60 m hub height, 80 m dia machine operating in a 25 m/s wind speed, the resulting maximum additional wind speed at the tip of a blade is 8.36 m/s, assuming category A turbulence.

Fault occurrence during power production

This group of load cases covers faults both internal and external to the machine, including grid loss. If the connection to the grid is lost, then the aerodynamic torque will no longer meet with any resistance from the generator – which, therefore, experiences ‘loss of load’ – and so

the rotor will begin to accelerate until the braking systems are brought into action. Depending on the speed of braking response, grid loss may well result in critical rotor loadings. The fault cases are as follows:

Load case 2.1: Operation in turbulent wind field defined by the normal turbulence model [NTM], together with control system fault or loss of the electrical network. Normal partial load factor.

Twelve ten-minute simulations are to be carried out, with the characteristic load defined as the mean of the six largest ten-minute extremes.

Load case 2.2: Operation in turbulent wind field defined by the normal turbulence model [NTM], together with protection system fault or preceding internal electrical fault. These faults are considered to be rare events, so the abnormal partial load factor is applied. The characteristic load is to be calculated as for load case 2.1.

Load case 2.3: Extreme operating gust [EOG], superimposed on hub height wind speed of $U_r \pm 2$ m/s or the cut-out wind speed, U_o , in conjunction with external or internal electrical system fault, including loss of electrical network. The wind speed variation is defined as

$$U(z, t) = \bar{U}(z) - \frac{1.221\sigma_u}{1 + 0.1(D/\Lambda_1)} \sin(3\pi t/T) [1 - \cos(2\pi t/T)] \quad (5.4)$$

where t is the time elapsed since the onset of the gust and T is the gust duration, specified at 10.5 seconds. Although termed an extreme operating gust, the gust magnitude is only about 70% of the one-year return extreme operating gust defined in IEC 61400-1 Edition 2, so the return period of the current EOG is likely to be only a few days. The gust profile, which incorporates a dip in the wind speed both before and after the main gust, is illustrated in Figure 5.2 for a hub height wind speed of 25 m/s, Class A turbulence, a turbine diameter of 80 m and a turbulence length scale of 42 m. The combination of the gust and the electrical system fault is considered to be a rare event, so the abnormal partial load factor is to be applied.

[Load case 2.4 is a fatigue load case]

Start-up load cases

[Load case 3.1 is a fatigue load case]

Load case 3.2: Extreme operating gust [EOG, as defined for load case 2.3, above] during start-up, superimposed on hub height wind speed of $U_r \pm 2$ m/s or U_o . Normal partial safety factor.

The magnitude of the gust has been chosen so that its recurrence period in conjunction with a start-up or shut-down is one in 50 years.

Load case 3.3: Extreme direction change [EDC] during start-up, for steady hub height wind speed of $U_r \pm 2$ m/s or U_o . The direction change, θ_e , is defined as:

$$\theta_e = \pm 4 \arctan \left(\frac{\sigma_u}{U_{\text{hub}} [1 + 0.1(D/\Lambda_1)]} \right) \quad (5.5a)$$

with the direction varying over time according to the relation:

$$\theta(t) = 0.5\theta_e \{1 - \cos(\pi t/T)\} \quad \text{for } 0 < t < T \quad (5.5b)$$

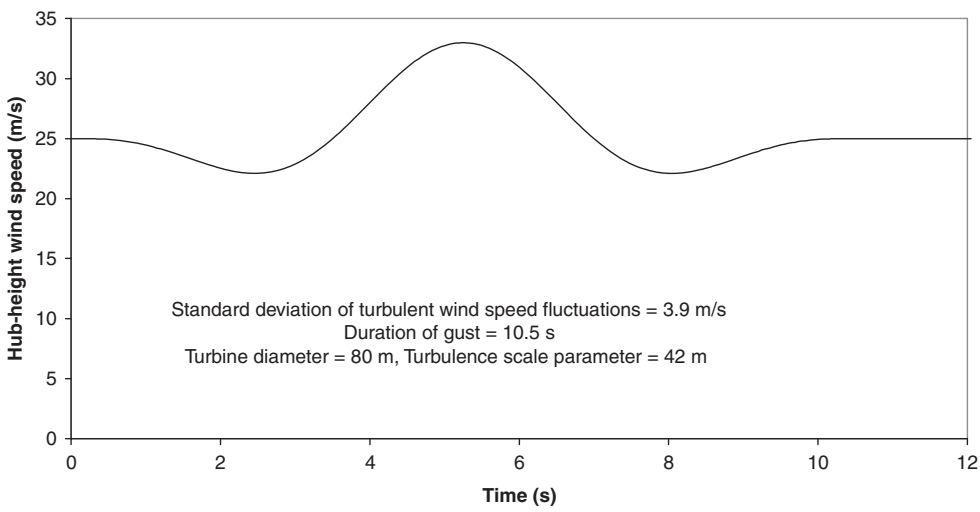


Figure 5.2 IEC 61400–1 Extreme rising and falling gust with 50-year return period for steady wind speed of 25 m/s and Category A turbulence

The direction change takes place over a period T of six seconds. Normal partial safety factor.

For a hub-height wind speed of 12 m/s, Class A turbulence, a turbine diameter of 80 m and a turbulence length scale of 42 m, the direction change is 37° , with a lower value applying at the cut-out wind speed because of the reduced turbulence intensity.

Shut-down load cases

[Load case 4.1 is a fatigue load case]

Load case 4.2: Extreme operating gust [EOG – as defined for load case 2.3, above] during shut-down, superimposed on hub-height wind speed of $U_r \pm 2$ m/s or U_o . Normal partial safety factor.

Load case 5.1: Emergency shut down during operation in turbulent wind field defined by the normal turbulence model [NTM], for steady hub height wind speed of $U_r \pm 2$ m/s or U_o . Normal partial safety factor.

The characteristic load is to be calculated as for load case 2.1.

5.4.2 Non-operational load cases

Normal machine state

When non-operational, a turbine is either stationary, that is, ‘parked’, or idling. In this condition it is exposed to the full range of wind speeds and is therefore required to survive the extreme wind conditions defined for the applicable wind class. IEC 61400–1 permits these wind conditions to be described in terms of either a steady wind speed corresponding to the three second gust with a 50-year return period or a turbulent wind with a ten minute mean equal to the 50-year return value (the ‘reference wind speed’) and a fixed turbulence intensity of 0.11.

The 50-year return gust value is defined as 1.4 times the 50-year return ten minute mean. Both gust and ten minute mean are specified at hub height, and are to be used in conjunction with a reduced wind shear exponent of 0.11.

The magnitude of the 50-year return gust depends on the gust duration chosen, which in turn should be based on the size of the loaded area. For example, the withdrawn British Standard CP3, Chapter V, Part 2 'Code of basic data for the design of buildings: wind loading', states that a three second gust can envelope areas up to 20 m, but advises that for larger areas up to 50 m across, a five second gust is appropriate. However, IEC 61400–1 and the GL rules specify the use of gust durations of three seconds regardless of the turbine size.

Recognising that the turbulent wind speed fluctuations are likely to excite blade and tower natural frequencies, IEC 61400–1 requires turbine dynamic response to be accounted for, whether using the deterministic or turbulent extreme wind models.

It is worth noting that Eurocode 1, Part 1–4 (2005) bases extreme loads on the dynamic pressure resulting from the extreme ten minute mean wind speed rather than a three second gust. The loads resulting from the extreme ten minute mean wind speed are augmented by a factor which takes into account both wind gusting and the excitation of resonant oscillations thereby.

The IEC 61400–1 non-operational load cases in the absence of faults or grid loss cater for varying yaw misalignments and may be summarised as follows:

Load case 6.1: Extreme wind with 50-year return period [EWM_{50}] Yaw misalignment up to $\pm 15^\circ$ using the steady wind model, or up to $\pm 8^\circ$ using the turbulent model, provided resistance against yaw slippage can be guaranteed. Normal partial safety factor.

Load case 6.3: Extreme wind with one-year return period [EWM_1] and extreme yaw misalignment of up to $\pm 30^\circ$ using the steady wind model, or up to $\pm 20^\circ$ using the turbulent model. The extreme wind speed with a return period of one year is to be taken as 80% of the 50-year return value. Normal partial safety factor.

Loss of grid connection can prevent the yaw system tracking any subsequent changes in wind direction, unless back-up is provided for the operation of the yaw system. Accordingly, IEC 61400–1 specifies loss of grid connection as a separate load case, which is summarised below. The use of the abnormal safety factor indicates that the combination of extreme wind and grid loss is considered rare.

Load case 6.2: Extreme wind with 50-year return period [EWM_{50}] and loss of grid connection. Yaw misalignment due to wind direction change up to $\pm 180^\circ$ unless back-up power for yawing provided. Abnormal partial safety factor.

The consequences of yaw slippage should be investigated in all non-operational load cases, if this is a possibility.

[Load Case 6.4 is a fatigue load case]

Machine fault state

Examples of faults in this category are ones involving the failure of the yaw or pitch mechanisms. On the assumption that there is no correlation between such a failure and extreme winds, the design wind condition for this load case is normally taken as the extreme wind condition with a return period of one year.

The load case in this category is summarised below.

Load case 7.1: Extreme wind with one-year return period [EWM_1] and machine fault. Yaw misalignment up to $\pm 180^\circ$ in case of yaw system fault. Abnormal partial safety factor.

5.4.3 Blade/tower clearance

In addition to checking the acceptability of stresses arising from the above load cases, the designer is also required to check that none can result in a collision between the blade and the tower, even after multiplying the blade tip deflection by the appropriate partial load factor and the partial safety factor for elasticity of the blade material. For load case 1.1 the characteristic blade tip deflection is to be determined by extrapolation in the same manner as the characteristic load.

5.4.4 Constrained stochastic simulation of wind gusts

Although time domain simulations of the turbulent wind field are able to accurately model the behaviour of the wind, very long simulations indeed are required if the extreme events required for design purposes are to be encountered by chance. Rather than select the desired extreme event from an extremely long wind speed time series, it is clearly desirable to ‘precipitate’ the desired event by constraining the time series in such a way that the statistical properties of the time series are unaffected.

A method of constraining gusts in wind simulations has been set out by Bierbooms (2009). In its simplest form, it consists in superposing a gust profile having the form of the wind speed autocorrelation function on a local maximum of the simulated wind time series. This is illustrated in Figure 5.3.

In this example, the gust magnitude of 19.5 m/s approximates to the 50-year return value for a Class 1A site, based on exposure to a 24–26 m/s wind speed for about four hours per annum. (The peak factor of five is calculated using Equation A5.42 in the Appendix, taking $\nu = 0.25$ Hz) The wind speed auto correlation function, derived from the power spectrum,

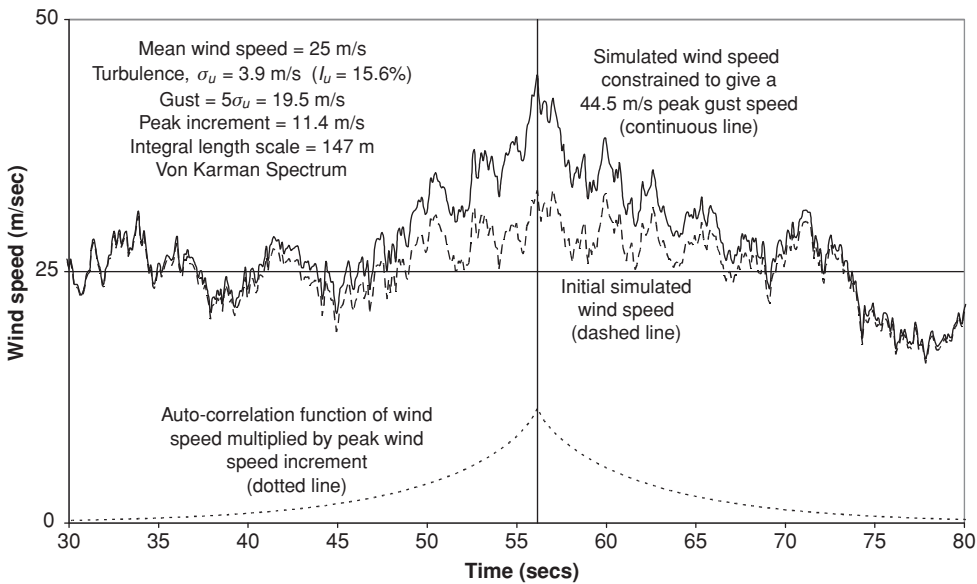


Figure 5.3 Simulated wind speed time series constrained to give a 44.5 m/s peak wind speed

is multiplied by the desired peak wind speed increment and added to the initial simulated wind speed time series, with the peak of the auto correlation function centred on the selected maximum of the initial time series.

There is no need for the timing of the peak of the constrained gust to coincide with a maximum of the initial time series. In the general case, the time series of the constrained gust, $u_c(t)$, can be obtained from the initial time series, $u(t)$ as follows:

$$u_c(t) = u(t) + \kappa(t - t_0)(A - u(t_0)) - \frac{\dot{\kappa}(t - t_0)}{\ddot{\kappa}(0)}\dot{u}(t_0) \quad (5.6)$$

where $\kappa(t - t_0)$ is the wind speed autocorrelation function and A the desired peak wind speed at time $t = t_0$.

The method can also be extended to gusts constrained to rise by a prescribed amount within a certain time (Bierbooms, 2009). Such gusts are important in relation to the design of pitch-controlled machines.

Although constrained stochastic stimulation appears to be a promising method of modelling extreme events, it is not, as yet, recognised by design standards.

5.5 Fatigue loading

5.5.1 Synthesis of fatigue load spectrum

The complete fatigue load spectrum for a particular wind turbine component has to be built up from separate load spectra derived for turbine operation at different wind speeds and from the load cycles experienced at start-up, normal shut-down and while the machine is parked or idling. Firstly the cycle counts for each stress range for one hour's operation in a particular wind speed band are calculated and scaled up by the predicted number of hours of operation in that band over the machine lifetime. This prediction can be based on the Weibull distribution (see Section 2.4), with the annual mean wind speed set according to the turbine class (see Section 5.1.2). Finally, the lifetime cycle counts obtained for operation in the different wind speed bands are combined and added to those calculated for start-ups, shut-downs and periods of non-operation.

5.6 Stationary blade loading

5.6.1 Lift and drag coefficients

Maximum blade loadings are in the out-of-plane direction and occur when the wind direction is either approximately normal to the blade, giving maximum drag, or at an angle of between 12° and 16° to the plane of the blade when the angle of attack is such as to give maximum lift.

In the absence of data on drag coefficients for airflow normal to the blade, designers formerly utilised the drag coefficient for an infinitely long flat plate of 2.0, with an adjustment downwards based on the aspect ratio. Thus, on a typical blade with a mean chord equal to one fifteenth of the radius, the length to width ratio would be taken as 30, because free flow cannot take place around the inboard end of the blade. Following EN 1991-1-4: 2005

‘Eurocode 1: Actions on structures – Part 1–4: Wind actions’, this would give a drag coefficient of 1.64. However, field measurements have shown that such an approach is unduly conservative, with drag coefficients of 1.24 being reported for the LM 17.2 m blade (Risø) and 1.25 for the Howden HWP-300 blade (Jamieson and Hunter, 1985). The 1992 edition of DS 472 ‘Loads and Safety of Wind Turbine Construction’ stipulated a minimum value of 1.3 for the drag coefficient.

The choice of lift coefficient value is more straightforward, because aerofoil data for low angles of attack is more generally available, and is, in any case, required for assessing rotor performance. The maximum lift coefficient rarely exceeds 1.6, but values down to as low as 1.1 will obtain on the thicker, inboard portion of the blade. The minimum value of lift coefficient of 1.5 specified in the 1992 edition of DS 472 for the calculation of blade out-of-plane loads is, therefore, probably conservative.

5.6.2 Critical configuration for different machine types

It was shown in the preceding section that the maximum lift coefficient is likely to exceed the maximum drag coefficient for a wind turbine blade, so consequently the maximum loading on a stationary blade will occur when the air flow is in a plane perpendicular to the blade axis and the angle of attack is such as to produce maximum lift. For a stall regulated machine, this will be the case when the blade is vertical and the wind direction is 75° – 80° to the nacelle axis, whereas for a pitch regulated machine the blade only needs to be approximately vertical with a wind direction at 10° – 20° to the nacelle axis.

5.6.3 Dynamic response

Tip displacement

Wind fluctuations at frequencies close to the first flapwise mode blade natural frequency excite resonant blade oscillations and result in additional, inertial loadings over and above the quasistatic loads that would be experienced by a completely rigid blade. As the oscillations result from fluctuations of the wind speed about the mean value, the standard deviation of resonant tip displacement can be expressed in terms of the wind turbulence intensity and the normalised power spectral density at the resonant frequency, $R_u(n_1) = n \cdot S_u(n_1)/\sigma_u^2$, as follows:

$$\frac{\sigma_{x1}}{\bar{x}_1} = 2 \frac{\sigma_u}{\bar{U}} \frac{\pi}{\sqrt{2\delta}} \sqrt{R_u(n_1)} \sqrt{K_{sx}(n_1)}. \quad (5.7)$$

Here $\bar{x}_1$ is the first mode component of the steady tip displacement, $\bar{U}$ is the mean wind speed (usually averaged over ten minutes), δ is the logarithmic decrement of damping and $K_{sx}(n_1)$ is a size reduction factor, which results from the lack of correlation of the wind along the blade at the relevant frequency. Note that the dynamic pressure, $\frac{1}{2}\rho U^2 = \frac{1}{2}\rho(\bar{U} + u)^2 = \frac{1}{2}\rho(\bar{U}^2 + 2\bar{U}u + u^2)$, is linearised to $\frac{1}{2}\rho\bar{U}(\bar{U} + 2u)$ to simplify the result. See Sections A5.2–4 in Appendix 5 for the derivations of Equation 5.7 and the expression for $K_{sx}(n_1)$.

Damping

It is evident from Equation 5.7 that a key determinant of resonant tip response is the level of damping present. Generally the damping consists of two components, aerodynamic and structural. In the case of a vibrating blade flat on to the wind, the aerodynamic force per unit length is given by $\frac{1}{2}\rho(\bar{U} - \dot{x})^2 C_D \cdot c(r) \cong \rho\bar{U}\dot{x}C_D \cdot c(r)$, where $\dot{x}$ is the blade flatwise velocity, C_D the drag coefficient and $c(r)$ the local blade chord. Hence, the aerodynamic damping per unit length, $\hat{c}_a(r)$, is $\rho\bar{U}C_D c(r)$ and the first mode aerodynamic damping ratio,

$$\xi_{a1} = c_{a1}/2m_1\omega_1 = \int_0^R \frac{\hat{c}_a(r)\mu_1^2(r)dr}{2m_1\omega_1}$$

is given by

$$\xi_{a1} = \rho\bar{U}C_D \int_0^R \frac{\mu_1^2(r)c(r)dr}{2m_1\omega_1}.$$

Here $\mu_1(r)$ is the first mode shape,

$$m_1 = \int_0^R m(r)\mu_1^2(r)dr$$

is the generalised mass, and ω_1 is the first mode natural frequency in radians per second. The logarithmic decrement is obtained by multiplying the damping ratio by 2π .

When the wind direction is angled to the blade so as to generate maximum lift, the blade will be approaching stall, with the result that the aerodynamic damping is effectively zero. In this situation tip deflections are limited only by the blade structural damping. Structural damping is discussed in Section 5.8.4, and values for typical blade materials given.

Root bending moment

The standard deviation of tip displacement in combination with the blade mode shape yields an inertial loading distribution from which the standard deviation of the resulting bending moment at any position along the blade may be calculated. In particular, the standard deviation of the root bending moment may be expressed in terms of the mean root bending moment as follows:

$$\frac{\sigma_{M1}}{\bar{M}} = 2\frac{\sigma_u}{\bar{U}} \frac{\pi}{\sqrt{2\delta}} \sqrt{R_u(n_1)} \sqrt{K_{Sx}(n_1)} \cdot \lambda_{M1} = \frac{\sigma_{x1}}{\bar{x}_1} \cdot \lambda_{M1} \quad (5.8a)$$

where

$$\lambda_{M1} = \frac{\int_0^R m(r)\mu_1(r)rdr}{m_1 \cdot \int_0^R c(r)rdr} \cdot \int_0^R c(r)\mu_1(r)dr \quad (5.8b)$$

See section A5.5 in the Appendix for the derivation of the expression for λ_{M1} .

The standard deviation of the quasistatic root bending moment fluctuation, or root bending moment *background* response is expressed in terms of the mean root bending moment by

$$\frac{\sigma_{MB}}{\bar{M}} = 2 \frac{\sigma_u}{U} \sqrt{K_{SMB}} \quad (5.9)$$

where K_{SMB} is a size reduction factor to take account of lack of correlation of wind fluctuations along the blade. As shown in Section A5.6 of the Appendix, K_{SMB} is usually only slightly less than unity because the blade length is small compared with the integral length scale of longitudinal turbulence measured in the across wind direction.

The variance of the total root bending moment fluctuations is equal to the sum of the resonant and background response variances, that is,

$$\sigma_M^2 = \sigma_{M1}^2 + \sigma_{MB}^2$$

Hence,

$$\frac{\sigma_M}{\bar{M}} = 2 \frac{\sigma_u}{U} \sqrt{K_{SMB} + \frac{\pi^2}{2\delta} R_u(n_1) K_{Sx}(n_1) \cdot \lambda_{M1}^2} \quad (5.10)$$

The design extreme root bending moment is typically calculated as that due to the 50-year return, ten minute mean wind speed plus the number of standard deviations of the root bending moment fluctuations corresponding to the likely peak excursion in a ten minute period. Thus,

$$M_{\max} = \bar{M} + g \cdot \sigma_M \quad (5.11)$$

where g is known as the peak factor, and depends on the number of cycles of root bending moment fluctuations in ten minutes, according to the formula

$$g = \sqrt{2 \ln(600\nu)} + \frac{0.577}{\sqrt{2 \ln(600\nu)}} \quad (5.12)$$

Here, ν is the mean zero-upcrossing frequency of the root bending moment fluctuations, which will be intermediate between that of the quasistatic wind loading and the blade natural frequency, n_1 – see Section A5.7 of the Appendix. (Note that, as g varies relatively slowly with frequency, it is a reasonable approximation to set g at an upper limit of 3.9, which corresponds to a frequency of about 1.9 Hz.)

Substituting Equation 5.10 into Equation 5.11 yields

$$M_{\max} = \bar{M} \left[1 + g \frac{\sigma_M}{\bar{M}} \right] = \bar{M} \left[1 + g \left(2 \frac{\sigma_u}{U} \right) \sqrt{K_{SMB} + \frac{\pi^2}{2\delta} R_u(n_1) K_{Sx}(n_1) \cdot \lambda_{M1}^2} \right] \quad (5.13)$$

The expression in square brackets is similar to the numerator of the structural factor, $c_s c_d$, in EN 1991-1-4: 2005

$$c_s c_d = \frac{1 + 2k_p I_1(z_s) \sqrt{B^2 + R^2}}{1 + 7I_1(z_s)} \quad (5.14)$$

in which $R^2 = (\pi^2/2\delta) R_u(n_1) K_{Sx}(n_1)$.

It is often necessary to express the maximum moment in terms of the quasistatic moment due to the 50-year return gust speed, U_{e50} . In order to do this, we equate the latter quantity to the quasistatic component of Equation 5.13, obtaining

$$C_f \cdot \frac{1}{2} \rho U_{e50}^2 \int_0^R c(r) \cdot r \cdot dr = \overline{M} \left(1 + g_0 \cdot 2 \frac{\sigma_u}{\overline{U}} \sqrt{K_{SMB}} \right) \quad (5.15)$$

Here, the peak factor g takes a lower value, g_0 , corresponding to the lower frequency of the quasistatic root bending moment fluctuations. Equation 5.15 can then be combined with Equation 5.13 to yield

$$M_{\max} = C_f \cdot \frac{1}{2} \rho U_{e50}^2 \int_0^R c(r) \cdot r \cdot dr \cdot Q_D \quad (5.16)$$

where Q_D is a dynamic factor given by

$$Q_D = \frac{1 + g \left(2 \frac{\sigma_u}{\overline{U}} \right) \sqrt{K_{SMB} + \frac{\pi^2}{2\delta} R_u(n_1) K_{Sx}(n_1) \cdot \lambda_{M1}^2}}{1 + g_0 \left(2 \frac{\sigma_u}{\overline{U}} \right) \sqrt{K_{SMB}}} \quad (5.17)$$

The dynamic factor Q_D equates to the EN 1991-1-4 dynamic factor, c_d , if g_0 is 3.5 and the ratio λ_{M1} is unity. The EN 1991-1-4 structural factor, c_s , given in Equation 5.14 is the product of the dynamic factor, c_d , and the size factor, c_s , given by $c_s = (1 + 7 \cdot I_v(z_s) \sqrt{B^2}) / (1 + 7 \cdot I_v(z_s))$.

There is considerable advantage in starting with the extreme gust speed and calculating the extreme root moment as the product of $C_f \cdot \frac{1}{2} \rho U_{e50}^2 \int_0^R c(r) \cdot r \cdot dr$ and Q_D , because it eliminates most of the error associated with linearising the formula for dynamic pressure. For example, if the extreme gust is 1.4 times the extreme ten minute mean wind speed, as postulated in IEC 61400-1 (which implies that the product $g_0(\sigma_u/\overline{U})$ is 0.4), then the dynamic pressure due to the gust will be $1.4^2 = 1.96$ times that due to the ten minute mean, rather than 1.8 times as given by the formula $1 + g_0(2(\sigma_u/\overline{U}))$.

Example 5.1

Evaluate the dynamic factor, Q_D , for the blade root bending moment for a 40 m long stationary blade under extreme loading.

Consider a trial 40 m long blade design (designated Blade T40) utilising NACA 632XX aerofoil sections with the chord and thickness distributions shown in Figure 5.4a. The thickness distribution has a pronounced knee near mid span to minimise the thickness to chord ratio in the outer half of the span. Assuming a uniform skin thickness along the blade, apart from local thickening at the root, the resulting mass and stiffness distributions are as shown in Figure 5.4b. Modal analysis as described in Section 5.8.2 yields the first and second mode shapes shown in Figure 5.4c. The first mode shape for a blade of constant cross-section (i.e. a uniform cantilever) is also shown for comparison purposes, and it is evident that the high stiffness of the inboard portion of the tapered blade results in dramatically reduced deflections there as a proportion of tip deflection. For a fibreglass blade, typical values of the Young's modulus and material density would be 40,000 N/sqmm and 1.7 Tonnes/m³ respectively, resulting in a first mode natural frequency of 0.824 Hz, and a second mode natural frequency of 2.86 Hz.

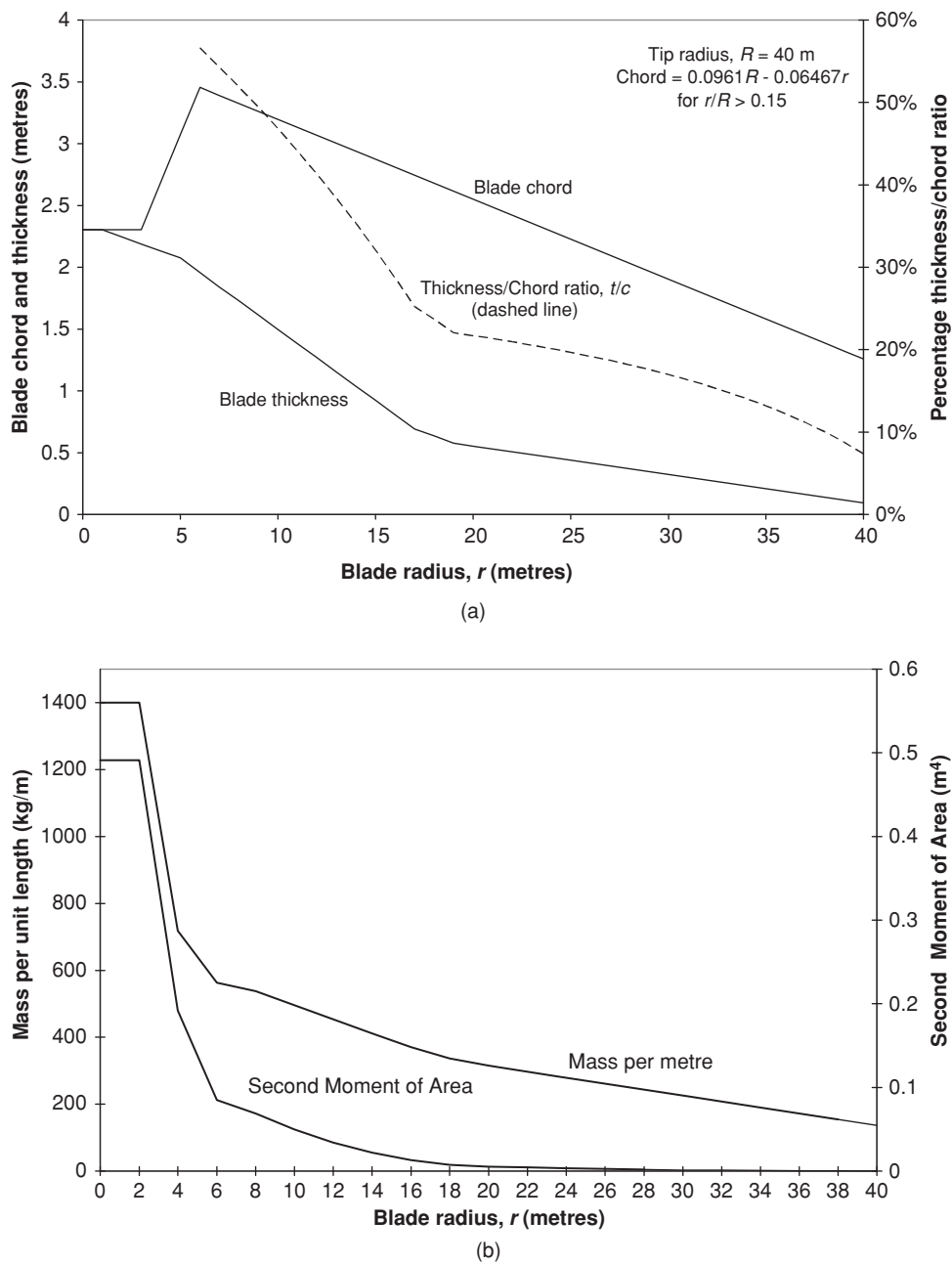


Figure 5.4 (a) Blade 'T40' chord and thickness distributions; (b) Blade 'T40' mass and stiffness distribution; (c) Blade 'T40' 1st and 2nd mode shapes

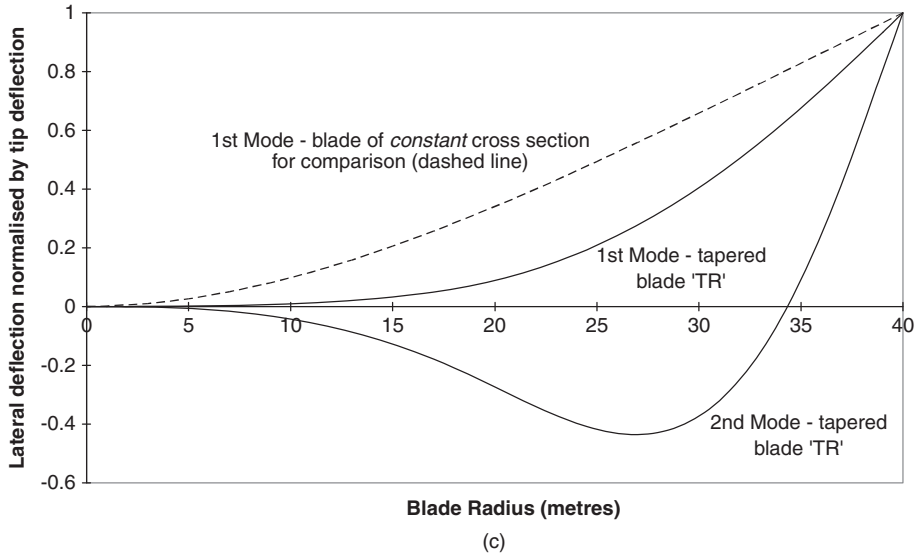


Figure 5.4 (Continued)

Values of the other parameters assumed are:

Blade height, z	70 metres
50 year return ten minute mean wind speed at blade height, $\bar{U}$	50 m/s
Eurocode 1 terrain category	I (roughness length, $z_o = 0.01$ m)
Turbulence intensity, $I(z) = 1/\ln(z/z_o)$	0.113

The corresponding integral length scale for longitudinal turbulence is 189 m according to EN 1991-1-4: 2005.

The values of the parameters in Equation 5.7 governing the resonant tip response are determined as follows:

- The aerodynamic damping is assumed to be zero, so the damping logarithmic decrement is taken as 0.05, corresponding to the structural damping value for fibreglass.
- The non-dimensional power spectral density of longitudinal wind turbulence, $R_u(n) = n \cdot S_u(n_1)/\sigma_u^2$, is calculated at the blade first mode natural frequency of 0.824 Hz according the Kaimal power spectrum defined in EN 1991-1-4 (Equation A5.8 in the Appendix) as 0.0631.
- A value of ten is taken for the non-dimensional decay constant, C , in the exponential expression for the normalised co-spectrum used in the derivation of the size reduction factor, $K_{sx}(n_1)$, in Equation A5.25.

The various stages in the derivation of the extreme root bending moment and the dynamic factor, Q_D , are set out below. The figures in square brackets are the corresponding values obtained using the method of Annex C of EN1991-1-4: 2005, which are included for comparison.

Size reduction factor for resonant response, $K_{Sx}(n_1)$	0.431	(Equation A5.25)	[0.322]
Ratio of standard deviation of resonant tip displacement to the first mode component of steady tip displacement, $\frac{\sigma_{y1}}{\bar{y}_1} = 2 \frac{\sigma_u}{U} \frac{\pi}{\sqrt{2\delta}} \sqrt{R_u(n_1)} \sqrt{K_{Sx}(n_1)}$ $= 2 \times 0.113 \times 9.935 \times \sqrt{0.0631} \times \sqrt{0.431}$ $= 2 \times 0.113 \times 9.935 \times 0.251 \times 0.657$ $= 2 \times 0.113 \times 1.638$		(Equation A5.7)	[N/A]
Root moment factor, λ_{M1}	0.584	(Equation 5.8b)	[N/A]
Ratio of standard deviation of resonant root moment to mean value, $\frac{\sigma_{M1}}{\bar{M}} = \frac{\sigma_{x1}}{\bar{x}_1} \lambda_{M1}$	0.216	(Equation 5.8a)	[0.320]
Size reduction factor for quasistatic or background response, K_{SMB}	0.829	(Equation A5.40)	[0.871]
Ratio of standard deviation of quasistatic root moment response to mean value, $\frac{\sigma_{MB}}{\bar{M}} = 2 \frac{\sigma_u}{U} \sqrt{K_{SMB}} = 2 \times 0.113 \times 0.910$	0.206	(Equation 5.9)	[0.197]
Ratio of standard deviation of total root moment response to mean value, $\frac{\sigma_M}{\bar{M}} = \sqrt{\left(\frac{\sigma_{MB}}{\bar{M}}\right)^2 + \left(\frac{\sigma_{M1}}{\bar{M}}\right)^2} = \sqrt{0.206^2 + 0.216^2}$	0.298		[0.376]
Zero up-crossing frequency of quasistatic response, n_0	0.343 Hz	(Equation A5.57)	[N/A]
Zero up-crossing frequency of total root moment response, ν	0.642 Hz	(Equation A5.54)	[N/A]
Peak factor, g , based on ν	3.62	(Equation 5.12)	[3.64]
Ratio of extreme moment to mean value, $\frac{M_{\max}}{\bar{M}} = 1 + g \left(\frac{\sigma_M}{\bar{M}} \right) = 1 + 3.62(0.298)$	2.08	(Equation 5.13)	[2.57]
Peak factor, g_0 , based on n_0	3.44		[3.5]
Ratio of quasistatic component of extreme moment to mean value $= 1 + g_0 \frac{\sigma_{MB}}{\bar{M}} = 1 + 3.44(0.206)$	1.708	(Equation 5.15)	[1.79]
Dynamic factor, $Q_D = 2.08/1.708$	1.218	(Equation 5.17)	[1.40]

It is apparent that the EN 1991–1–4 method yields a significantly larger value of the extreme root bending moment. However, the EN 1991–1–4 ratio of extreme to mean bending moment is intended to apply at all points along the blade, so a conservative value at the root is inescapable, as is shown in the next section which examines the variation of bending moment along the blade.

Spanwise variation of bending moment

The resonant and quasistatic components of bending moments at intermediate positions along the blade can be related to those at the root in a straightforward way.

As far as the quasistatic bending moment fluctuations are concerned, the variation along the blade follows closely the bending moment variation due to the steady loading, although slight changes in the size reduction factor have a small effect. The bending moment diagram for the resonant oscillations is, however, of a very different shape, because of the dominance of the inertia loading on the tip. An expression for the resonant bending moment variation along the blade is given in section A5.8 of the Appendix, and it is plotted out for the example above in Figure 5.5a, with the quasistatic bending moment variation alongside for comparison. It is seen that the resonant bending moment diagram is closer to linear than the quasistatic one, which approximates to a parabola.

A consequence of the much slower decay of the resonant BM out towards the tip is an increase in the ratio of the resonant bending moment standard deviation to the local steady moment with radius. This results in an increase in the dynamic magnification factor, Q_D , from 1.22 at the root to 1.96 near the tip for the example above (see Figure 5.5b).

5.7 Blade loads during operation

5.7.1 Deterministic and stochastic load components

It is normal to separate out the loads due to the steady wind on the rotating blade from those due to wind speed fluctuations and analyse them in different ways. The periodic loading on the blade due to the steady spatial variation of wind speed over the rotor swept area is termed the deterministic load component, because it is uniquely determined by a limited number of parameters, that is, the hub-height wind speed, the rotational speed, the wind shear and so on. On the other hand, the random loading on the blade due to wind speed fluctuations (i.e. turbulence) has to be described probabilistically, and is therefore termed the stochastic load component.

In addition to wind loading, the rotating blade is also acted on by gravity and inertial loadings. The gravity loading depends simply on blade azimuth and mass distribution, and is thus deterministic, but the inertial loadings will be affected by turbulence – as, for example, in the case of a teetering rotor, or a flexing blade – and so will contain stochastic as well as deterministic components.

5.7.2 Deterministic aerodynamic loads

Steady, uniform flow perpendicular to plane of rotor

The application of momentum theory to a blade element, which is described in Section 3.5.3, enables the aerodynamic forces on the blade to be calculated at different radii. Equations 3.54 and 3.55 are solved iteratively for the flow induction factors, a and a' , at each radius,

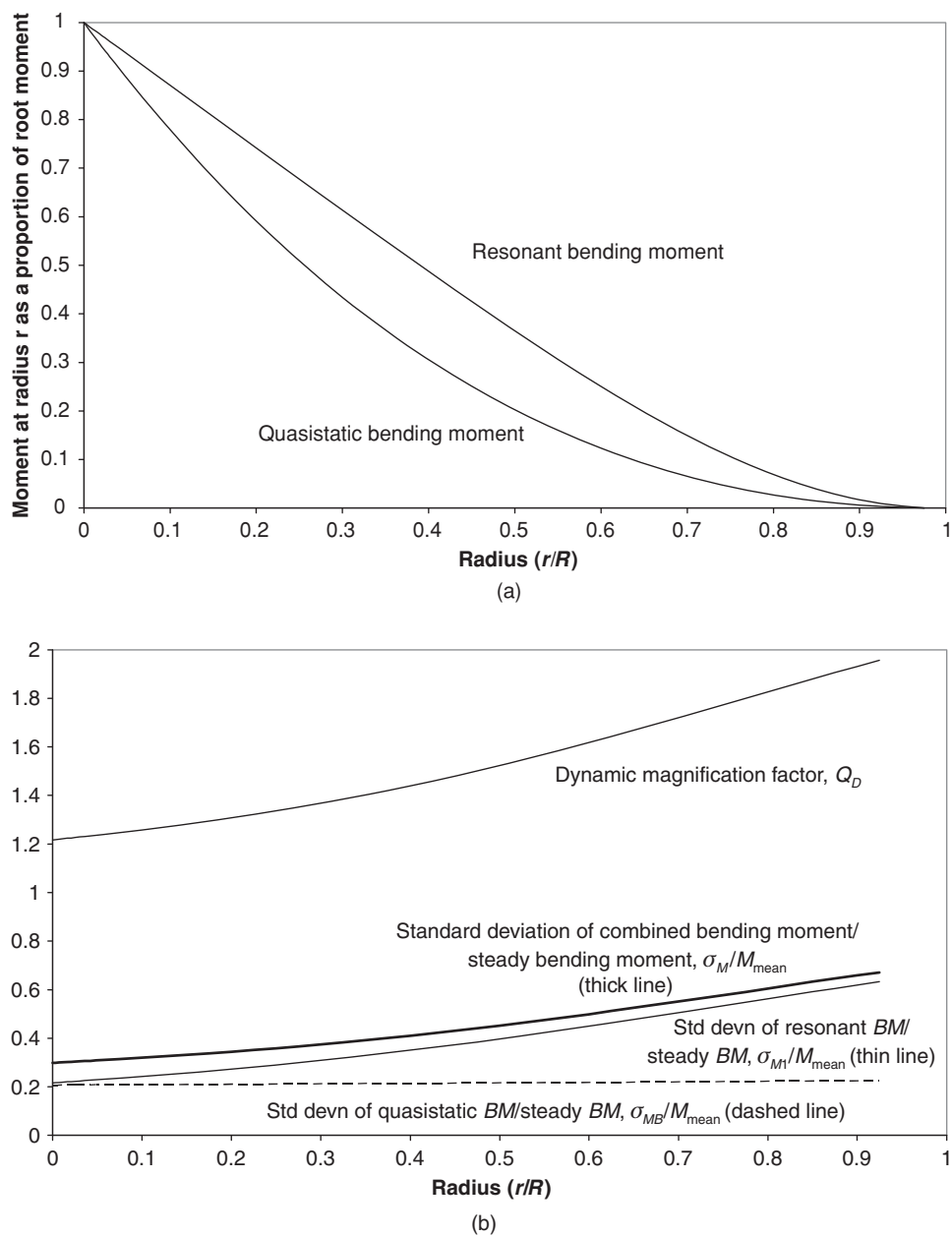


Figure 5.5 (a) Spanwise variation of resonant and quasistatic moments – Blade ‘T40’; (b) Spanwise variation of (i) fluctuating bending moment distributions in terms of local steady bending moment and (ii) dynamic magnification factor – for Blade ‘T40’

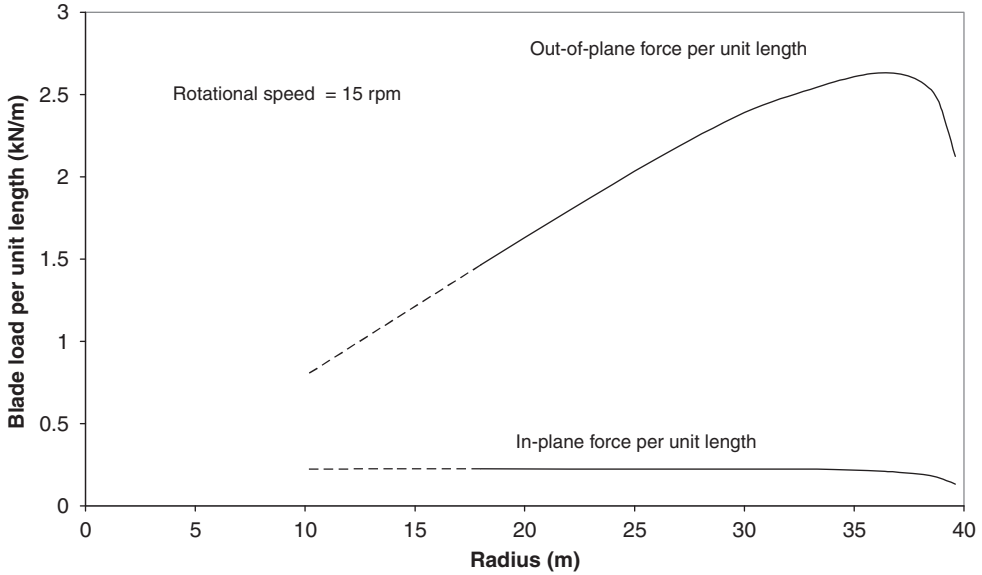


Figure 5.6 Distribution of blade in-plane and out-of-plane aerodynamic loads during operation of typical 80 m diameter stall-regulated machine in a steady, uniform 8 m/s wind

enabling the flow angle, ϕ , the angle of attack, α , and hence the lift and drag coefficients to be determined.

For loadings on the outboard portion of the blade, allowance for tip loss must be made, so Equations 3.54 and 3.55 are replaced by Equations 3.54b and 3.55a in Section 3.8.6. These equations can be arranged to give the following expressions for the forces per unit length on an element perpendicular to the plane of rotation and in the direction of blade motion, known as the out-of-plane and in-plane forces respectively:

Out-of-plane force per unit length,

$$F_X = C_x \cdot \frac{1}{2} \rho W^2 \cdot c = 4\pi\rho \cdot U_\infty^2 (1 - af)a \cdot \frac{f}{N}r \quad (5.18)$$

In-plane force per unit length,

$$F_Y = C_y \cdot \frac{1}{2} \rho W^2 \cdot c = 4\pi\rho \cdot \Omega U_\infty (1 - af)a' \cdot \frac{f}{N}r^2 \quad (5.19)$$

The parameters in the expressions are as defined in Chapter 3. f is the tip loss factor, and N is the number of blades.

The variation of the in-plane and out-of-plane forces with radius is shown in Figure 5.6 for a typical machine operating in a steady 8 m/s wind speed. The 80 m stall-regulated turbine considered in this example is fitted with three ‘T40’ blades as described in Example 5.1 and rotates at 15 rpm. The blade twist distribution is linear, and selected to produce the maximum energy yield for an annual mean wind speed of 7.5 m/s. It is evident that the out-of-plane load per unit length increases approximately linearly with radius, in spite of the reducing blade chord until the effects of tip loss are felt beyond about 80% of tip radius. Note that the form of the variation would be the same for any combination of rotational speed, wind speed and

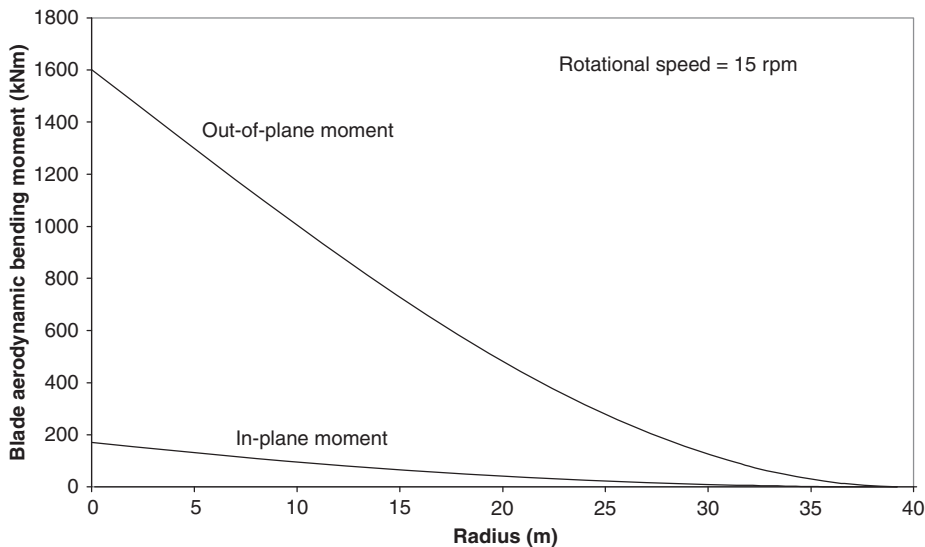


Figure 5.7 Distribution of blade in-plane and out-of-plane aerodynamic bending moments during operation of typical 80 m diameter stall-regulated machine in a steady, uniform 8 m/s wind

tip radius yielding the same tip speed ratio, because it is the tip speed ratio that determines the radial distribution of flow angle ϕ , and of the induction factors a and a' .

Integration of these forces along the blade then yields in-plane and out-of-plane aerodynamic blade bending moments. The variation of these moments with radius is shown in Figure 5.7 for the example above. The blade bending moments effectively decrease linearly with increasing radius over the inboard third of the blade because of the concentration of loading outboard.

The variation of the blade root out-of-plane bending moment with wind speed is illustrated in Figure 5.8 for the 80 m diameter example machine described above. As explained in Section 3.9, the phenomenon of stall delay results in significantly increased values of the lift coefficient at higher wind speeds on the inboard section of the rotating blade than predicted by static aerofoil data, such as that reproduced in Figure 3.45. Accordingly, Figure 5.8 and the other figures referred to in this section have been derived using realistic aerofoil data for a rotating LM-19.0 blade reported in Petersen *et al.* (1998), which is based on an empirical modification of static or 2D aerofoil data. The modified data is reproduced in Figure 5.9, and displays much higher lift coefficients for the thicker, inboard blade sections at high angles of attack than for the thinner, outboard blade sections because of stall delay at the inboard sections.

Figure 5.8 shows the blade root out-of-plane bending moment increasing nearly linearly with wind speed at first and then levelling off, becoming almost constant for winds between 12 m/s and 16 m/s, as the blade goes into stall. Thereafter the root moment increases again, but much more gently than before.

Also shown in Figure 5.8 is the variation of blade root out-of-plane bending moment with wind speed for the same machine with pitch regulation to limit the power output to 1600 kW. It is evident that the bending moment drops away rapidly at wind speeds above rated.

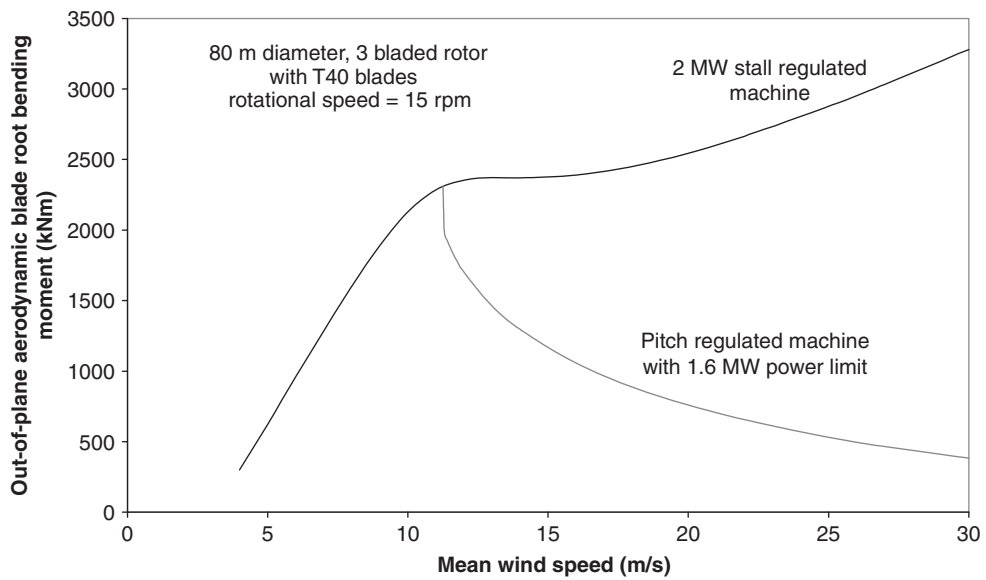


Figure 5.8 Blade out-of-plane root bending moment during operation is steady, uniform wind – variation with wind speed for similar stall-regulated and pitch-regulated machines

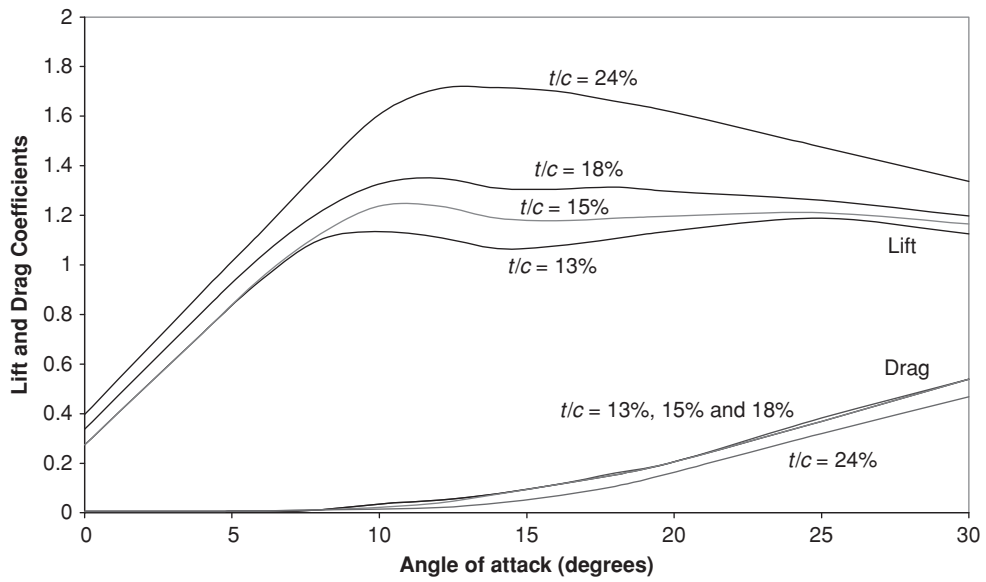


Figure 5.9 Aerofoil data for LM 19.0 blade for various thickness/chord ratios (from Petersen *et al.* (1998))

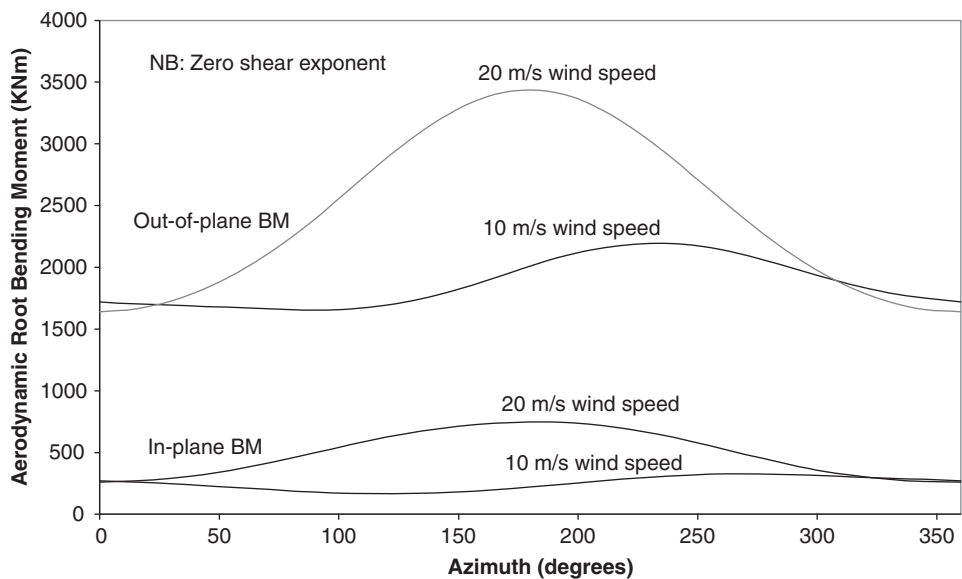


Figure 5.10 Variation of blade root bending moment with azimuth, for typical 80 m diameter stall-regulated machine operating at a steady 30° yaw

Yawed flow

The application of blade-element momentum theory to steady yawed flow is described in Section 4.2.8. This methodology has been used to derive Figure 5.10, which shows the variation of the blade root out-of-plane and in-plane moments with azimuth for the 80 m diameter stall-regulated machine described above, operating at a steady yaw angle of +30°. Note that the blade azimuth is measured in the direction of blade rotation, from a zero value at top dead centre, and the yaw angle is defined as positive when the lateral component of air flow with respect to the rotor disc is in the same direction as the blade movement at zero azimuth.

Figure 5.10 reveals a distinct difference between the behaviour at 10 m/s and 20 m/s. In the latter case, the bending moment variation is sinusoidal with a maximum value at 180° azimuth, indicating that the variation is dominated by the effect of the fluctuation of the air velocity relative to the blade, W . At 10 m/s, however, the maximum out-of-plane bending moment occurs at about 240° azimuth, suggesting that the non-uniform component of induced velocity, u_1 (Equation 4.20) is also significant. As wind speed increases, of course, the induction factor, a , becomes small, reducing the impact of u_1 .

Shaft tilt

Upwind machines, that is, wind turbines with the rotor positioned between the tower and the oncoming wind, – normally have the rotor shaft tilted upwards by several degrees in order to increase the clearance between the rotor and the tower. Thus, as for the case of yaw misalignment, the flow is inclined to the rotor shaft axis, but tilted upwards rather than sideways, so the treatment of shaft tilt mirrors that of yawed flow.

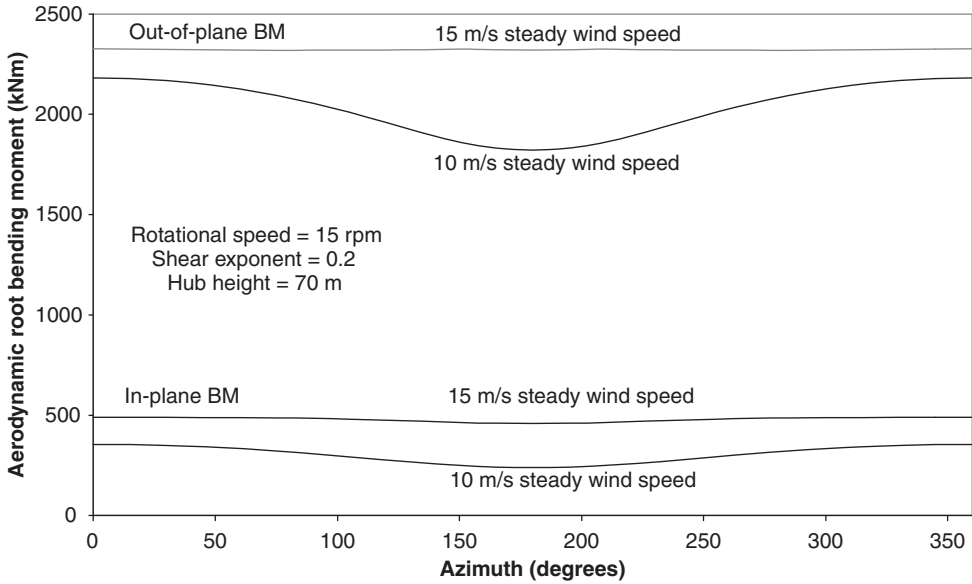


Figure 5.11 Variation of blade root bending moments with azimuth due to wind shear, for typical 80 m diameter stall-regulated machine operating in steady hub height winds of 10 m/s and 15 m/s (shear exponent = 0.2)

Wind shear

The increase of wind speed with height is known as wind shear. The theoretical logarithmic profile, $U(z) \propto \ln(z/z_0)$, is usually approximated by the power law, $U(z) \propto (z/z_{\text{ref}})^\alpha$ for wind turbine design purposes. The appropriate value of the exponent α increases with the surface roughness, z_0 , with a figure of 0.14 typically quoted for level countryside, although the speed up of airflow close to the ground over rounded hills usually results in a lower value at hill tops. As already noted, IEC 61400–1 specifies a conservative value of 0.20.

In applying momentum theory to this case, the velocity component at right angles to the plane of rotation is expressed as $U_\infty (1 + r \cos \psi / z_{\text{hub}})^\alpha (1 - a)$. The variation of blade root bending moments with azimuth due to wind shear is illustrated in Figure 5.11 for the example 80 m diameter stall regulated machine, taking the exponent as 0.20, the hub-height as 70 m and considering hub-height wind speeds of 10 m/s and 15 m/s. In the former case, the variation is nearly sinusoidal, but in the latter case the root bending moments are effectively constant, as the blade is in stall.

Tower shadow

The blocking of the air flow by the tower results in regions of reduced wind speed both upwind and downwind of the tower. This reduction is more severe for tubular towers than for lattice towers and, in the case of tubular towers, is larger on the downwind side because of flow separation. As a consequence, designers of downwind machines usually position the rotor plane well clear of the tower to minimise the interference effect.

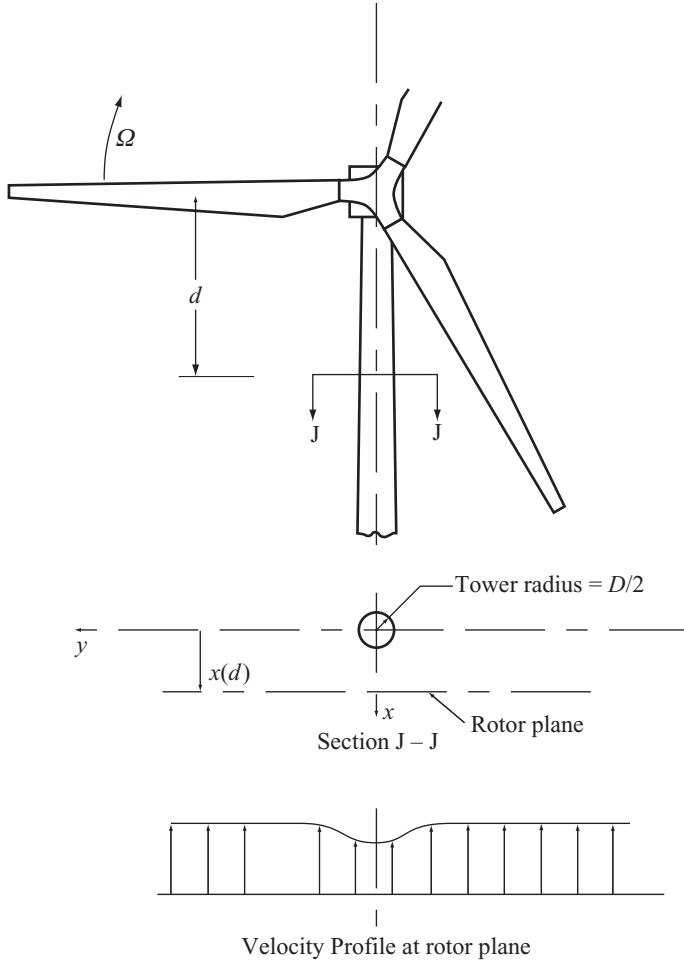


Figure 5.12 Tower shadow parameters

The velocity deficits upwind of a tubular tower can be modelled using potential flow theory. The flow around a cylindrical tower is derived by superposing a doublet, that is, a source and sink at very close spacing, on a uniform flow, U_∞ , giving the stream function:

$$\psi = U_\infty y \left(1 - \frac{(D/2)^2}{x^2 + y^2} \right) \quad (5.20)$$

where D is the tower diameter, and x and y are the longitudinal and lateral co-ordinates with respect to the tower centre – see Figure 5.12. Differentiation of ψ with respect to y yields the following expression for the flow velocity in the x direction:

$$U = U_\infty \left(1 - \frac{(D/2)^2(x^2 - y^2)}{(x^2 + y^2)^2} \right) \quad (5.21)$$

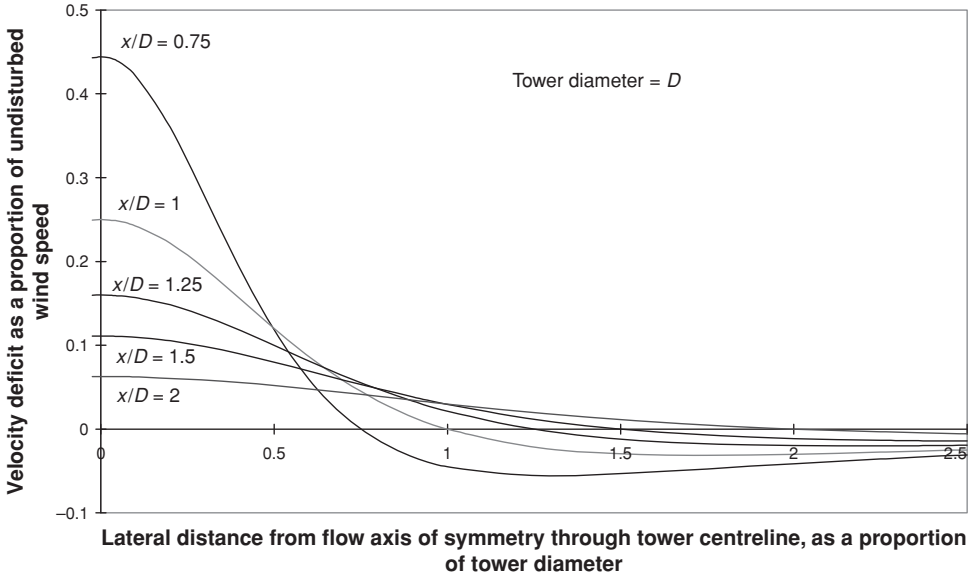


Figure 5.13 Profile of velocity deficit due to tower shadow at different distances x/D upwind of tower centreline

The second term within the brackets, which is the velocity deficit as a proportion of the undisturbed wind speed, is plotted out against the lateral co-ordinate, y , divided by tower diameter, for a range of upwind distances, x , in Figure 5.13. The velocity deficit on the flow axis of symmetry is equal to $U_{\infty}(D/2x)^2$ and the total width of the deficit region is twice the upwind distance. Consequently the velocity gradient encountered by a rotating blade decreases rapidly as the upwind distance, x , increases.

The effect of tower shadow on blade loading can be estimated by setting the local velocity component at right angles to the plane of rotation equal to $U(1 - a)$ in place of $U_{\infty}(1 - a)$, and applying blade element theory as usual. Results for blade root bending moments for the example 80 m diameter stall-regulated machine are given in Figure 5.14, assuming a tower diameter of 4 m and ignoring dynamic effects. The plots show the variation of in-plane and out-of-plane root moments with azimuth during operation in wind speeds of 10 m/s and 20 m/s, for a blade-tower clearance equal to the tower radius, that is, for $x/D = 1$. Note that the dip in out-of-plane bending moment is more severe at the lower wind speed. Also shown are 10 m/s plots for $x/D = 1.5$, which exhibit a much less severe disturbance.

In the case of downwind turbines, the flow separation and generation of eddies which take place are less amenable to analysis, so empirical methods are used to estimate the mean velocity deficit. Commonly the profile of the velocity deficit is assumed to be of cosine form, so that

$$U = U_{\infty} \left(1 - k \cos^2 \left(\frac{\pi y}{\delta} \right) \right) \quad (5.22)$$

where δ is the total width of the deficit region. The slight enhancement of velocities beyond the deficit region is usually ignored. See also Section 6.13.2.

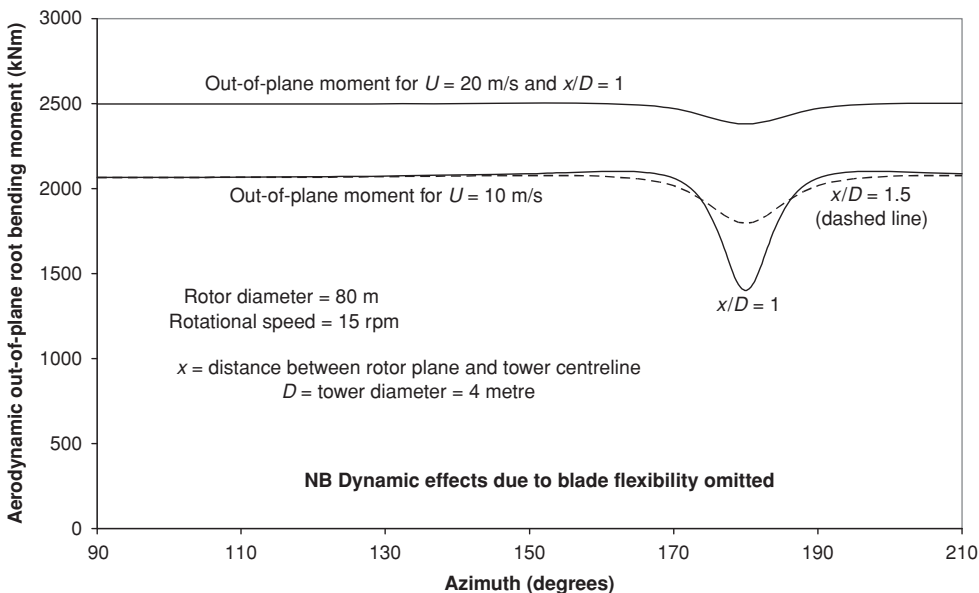


Figure 5.14 Variation of blade root out-of-plane bending moment with azimuth due to tower shadow, for typical 80 m diameter stall-regulated upwind machine operating in steady, uniform winds of 10 m/s and 20 m/s

The sharp dip in blade loading caused by tower shadow is more prone to excite blade oscillations than the smooth variations in load due to wind shear, shaft tilt and yaw, and this aspect is considered in the section on blade dynamic response.

Wake effects

Within a wind farm it is common for one turbine to be operating wholly or partly in the wake of another. In the latter case, which is more severe, the downwind turbine is effectively subjected to horizontal wind shear, and the blade load fluctuations can be analysed accordingly.

5.7.3 Gravity loads

Gravity loading on the blade results in a sinusoidally varying edgewise bending moment which reaches a maximum when the blade is horizontal, and which changes sign from one horizontal position to the other. It is, thus, a major source of fatigue loading. For the blade ‘T40’ (see Example 5.1), the maximum gravity moment, $\int_0^R m(r)rdr$ is 2140 kNm, so the edgewise bending moment range due to gravity is 4280 KNm. This dwarfs the variations in edgewise moment due to yaw or wind shear, which are typically one tenth this value or less. The spanwise distribution of gravity bending moment is shown in Figure 5.15 for blade ‘T40’.

5.7.4 Deterministic inertia loads

Centrifugal loads

For a rigid blade rotating with its axis perpendicular to the axis of rotation, the centrifugal forces generate a simple tensile load in the blade which at radius r^* is given by the expression

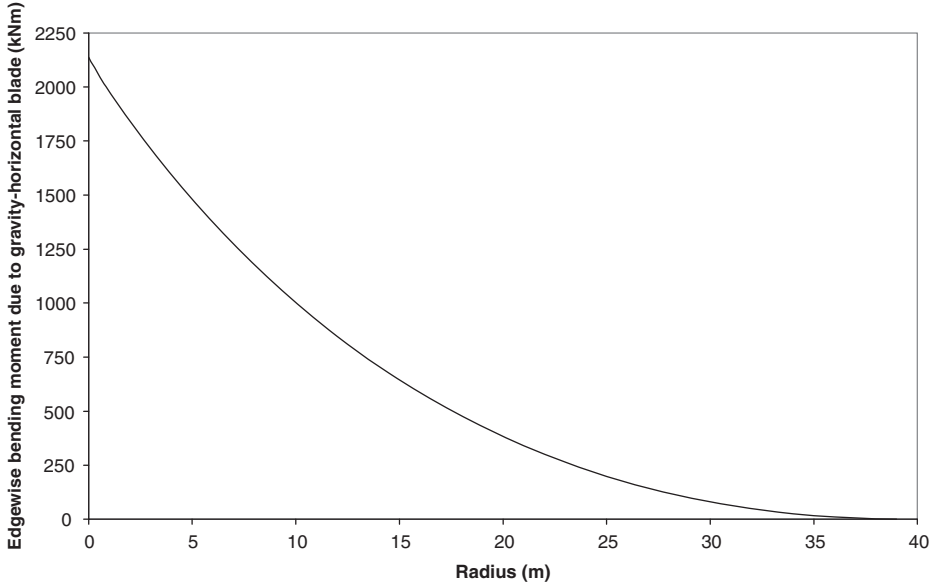


Figure 5.15 Blade ‘T40’ gravity bending moment distribution

$\Omega^2 \int_{r_*}^R m(r) r dr$. As a result the fluctuating stresses in the blade arising from all loading sources always have a tensile bias during operation. For blade ‘T40’ rotating at 15 rpm, the centrifugal force at the root amounts to 540 kN – approximately three and a half times its weight.

Thrust loading causes flexible blades to deflect downwind, with the result that the centrifugal forces generate blade out-of-plane moments in opposition to those due to the thrust. This reduction of the moment due to thrust loading is known as centrifugal relief. The phenomenon is non-linear, so iterative techniques are required to arrive at a solution. Greater centrifugal relief can be obtained by coning the rotor so that the blades are inclined downwind in the first place. A balance can be struck so that the maximum forward out-of-plane moment due centrifugal loads in very low wind is approximately equal to the maximum rearward out-of-plane moment due to the thrust loading in combination with centrifugal loads during operation in rated wind.

Gyroscopic loads

When an operating machine yaws, the blades experience gyroscopic loads perpendicular to the plane of rotation. Consider the point A on a rotor rotating clockwise at a speed of Ω rad/sec, as illustrated in Figure 5.16. The instantaneous horizontal velocity component of point A due to rotor rotation is Ωz , where z is the height of the point above the hub. If the machine is yawing clockwise in plan at a speed of Λ rad/sec, then it can be shown that point A accelerates at $2\Omega \Lambda z$ towards the wind, assuming the rotor is rigid. Integrating the resulting inertial force over the blade length gives the following expression for blade root out-of-plane bending moment:

$$M_Y = \int_0^R 2\Omega \Lambda z r \cdot m(r) dr = 2\Omega \Lambda \cos \psi \int_0^R r^2 m(r) dr = 2\Omega \Lambda \cos \psi \cdot I_B \quad (5.23)$$

where I_B is the blade inertia about the root.

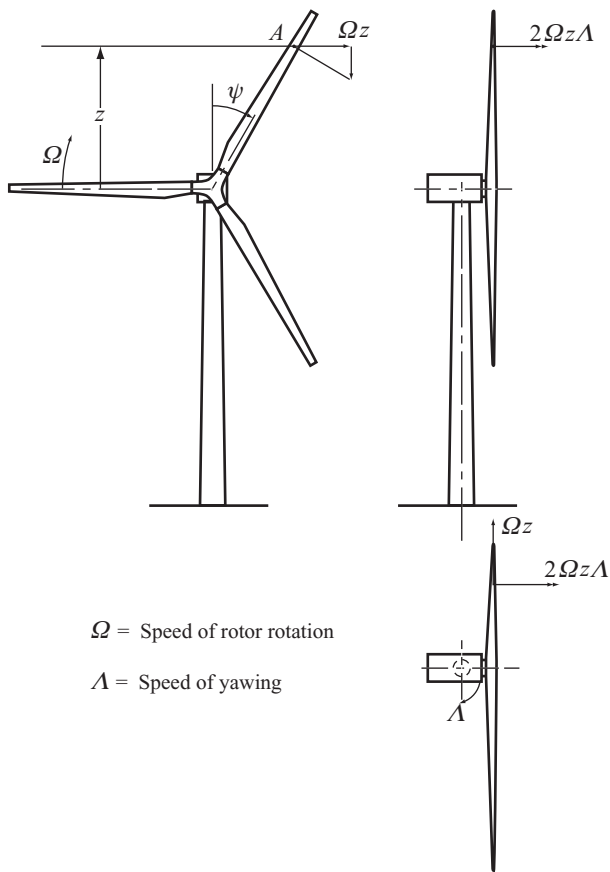


Figure 5.16 Gyroscopic acceleration of a point on a yawing blade

As an example, consider an 80 m diameter machine with ‘T40’ blades yawing at 1° per second during operation at 15 rpm. The blade root inertia is 4910 Tm^2 , so the maximum value of M_Y is $2(\pi/2)(0.0175)4910 = 270 \text{ KNm}$. This is only about 10% of the maximum out-of-plane moment due to aerodynamic loads.

Braking loads

Rotor deceleration due to mechanical braking introduces edgewise blade bending moments which are additive to the gravity moments on a descending blade.

Teeter loads

Blade out-of-plane root bending moments can be eliminated entirely by mounting each blade on a hinge so that it is free to rotate in the fore-aft direction. Although centrifugal forces are effective in controlling the cone angle of each blade at normal operating speeds, the need for alternative restraints during start-up and shut-down means that such hinges are rarely used. However, in the case of two-bladed machines, it is convenient to mount the whole

rotor on a single shaft hinge allowing fore-aft rotation or ‘teetering’, and this arrangement is frequently adopted in order to reduce out-of-plane bending moment fluctuations at the blade root, and to prevent the transmission of blade out-of-plane moments to the low speed shaft. As teetering is essentially a dynamic phenomenon, consideration of teeter behaviour is deferred to Section 5.8.

5.7.5 Stochastic aerodynamic loads: analysis in the frequency domain

As noted in Section 5.7.1, the random loadings on the blade due to short-term wind speed fluctuations are known as stochastic aerodynamic loads. The wind speed fluctuations about the mean at a *fixed* point in space are characterised by a probability distribution – which, for most purposes, can be assumed to be normal – and by a power spectrum which describes how the energy of the fluctuations is distributed between different frequencies (see Sections 2.6.3 and 2.6.4).

The stochastic loads are most conveniently analysed in the frequency domain, but in order to facilitate this, it is usual to assume a linear relation between the fluctuation, u , of the wind speed incident on the aerofoil and the resultant loadings. This is a reasonable assumption for an unstalled blade at high tip speed ratio, as will be shown. The fluctuating aerodynamic lift per unit length, L , is $\frac{1}{2}\rho W^2 C_L c$, where W is the air velocity relative to the blade, C_L is the lift coefficient and the drag term is ignored. Because the flow angle, ϕ , is small at high tip-speed ratio, λ , the relative air velocity, W , can be assumed to be changing much more slowly with the wind speed than C_L , so that dW/du can be ignored. As a result,

$$\frac{dL}{du} = \frac{1}{2}\rho W^2 c \frac{dC_L}{d\alpha} \frac{d\alpha}{du} \quad \text{where } \alpha, \text{ the angle of attack} = \phi - \beta \quad (5.24)$$

If the blades are not pitching, then the local blade twist, β , is constant, so that $d\alpha/du = d\phi/du$. To preserve linearity, it is necessary to assume that the rate of change of lift coefficient with angle of attack, $dC_L/d\alpha$ is constant, which is tenable only if the blade remains unstalled. Assuming for simplicity that the wake is frozen, that is, that the induced velocity, $\bar{U}a$, remains constant, despite the wind speed fluctuations, u , we obtain

$$\tan \phi \cong (\bar{U}(1 - a) + u)/\Omega r,$$

so that, for ϕ small,

$$\frac{d\phi}{du} \cong \frac{1}{\Omega r} \quad \text{and} \quad W \cong \Omega r,$$

leading to

$$\Delta L = L - \bar{L} = u \frac{dL}{du} = \frac{1}{2}\rho(\Omega r)^2 c \frac{dC_L}{d\alpha} \frac{u}{\Omega r} = \frac{1}{2}\rho\Omega r c \frac{dC_L}{d\alpha} u \quad (5.25)$$

Hence,

$$\sigma_L = \left(\frac{1}{2}\rho\Omega^2 \frac{dC_L}{d\alpha} \right) r c \sigma_u$$

Theoretically $dC_L/d\alpha$ is equal to 2π , but see Section A3.8 in Appendix A3.

If the turbulence integral length scale is large compared to the blade radius, then the expression for the standard deviation of the blade root bending moment – assuming a completely rigid blade – approximates to

$$\sigma_M = \int_0^R \sigma_L r dr = \frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \sigma_u \int_0^R c(r) \cdot r^2 dr \quad (5.26)$$

where σ_u is the standard deviation of the wind speed incident on the rotor disc, which, by virtue of the ‘frozen wake’ assumption, equates to the standard deviation of the wind speed in the undisturbed flow. If, as will be the case in practice, the longitudinal wind fluctuations are not perfectly correlated along the length of the blade, then

$$\sigma_M^2 = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_0^R \int_0^R \kappa_u(r_1, r_2, 0) c(r_1) c(r_2) r_1^2 r_2^2 dr_1 dr_2 \quad (5.27)$$

where $\kappa_u(r_1, r_2, 0)$ is the cross correlation function $\kappa_u(r_1, r_2, \tau)$ between the wind fluctuations at radii r_1 and r_2 with the time lag τ set equal to zero, that is,

$$\kappa_u(r_1, r_2, 0) = \left[\frac{1}{T} \int_0^T u(r_1, t) u(r_2, t) dt \right] \quad (5.28)$$

In reality, of course, the blade will not be completely rigid, so the random wind loading will excite the natural modes of blade vibration. In order to quantify these excitations, it is first necessary to know the energy content of the incident wind fluctuations *as seen by each point on the rotating blade* at the blade natural frequencies – information which is provided by the ‘rotationally sampled spectrum’. This spectrum is significantly different from the fixed point spectrum, because a rotating blade will often slice through an individual gust (defined as a volume of air travelling at above average speed) several times, as the gust dimensions are frequently large compared with the distance travelled by the air in one turbine revolution. This phenomenon, known as ‘gust slicing’, considerably enhances the frequency content at the rotational frequency and, to a lesser extent, at its harmonics also.

The method for deriving the rotational spectrum is described below. The dynamic response of a flexible blade to random wind loading is explored in Section 5.8.

Rotationally sampled spectrum

The derivation of the power spectrum of the wind seen by a point on a rotating blade is based on the Fourier transform pairs:

$$S_u(n) = 4 \int_0^\infty \kappa_u(\tau) \cos 2\pi n \tau d\tau \quad (5.29)$$

$$\kappa_u(\tau) = \int_0^\infty S_u(n) \cos 2\pi n \tau dn \quad (5.30)$$

where $S_u(n)$ is the single sided spectrum of wind speed fluctuations in terms of frequency in Hz. Firstly, the latter equation is used to obtain the autocorrelation function, $\kappa_u(\tau)$, for

the along wind turbulent fluctuations at a fixed point in space from the corresponding power spectrum. Secondly, $\kappa_u(\tau)$ is used to derive the related autocorrelation function, $\kappa_u^0(r, \tau)$, for a point on the rotating blade at radius r . Finally this function is transformed using Equation 5.29 to yield the rotationally sampled spectrum. The three steps are set out in more detail below. Note that three key simplifying assumptions are made – that the turbulence is homogeneous and isotropic, and that the flow is incompressible.

Step 1 – Derivation of the autocorrelation function at a fixed point: The von Karman spectrum is chosen as the input spectrum, because it is isotropic and homogeneous. The power spectrum of the along-wind wind speed fluctuations at a fixed point in space is given by Equation 2.25:

$$\frac{S_u(n)}{\sigma_u^2} = \frac{4L/\bar{U}}{(1 + 70.8(nL/\bar{U})^2)^{\frac{5}{6}}} \quad (5.31)$$

where L is the isotropic integral length scale of turbulence. It can be shown that Equation 5.30 yields the following expression for the corresponding auto correlation function:

$$\kappa_u(\tau) = \frac{2\sigma_u^2}{\Gamma\left(\frac{1}{3}\right)} \left(\frac{\tau/2}{T'}\right)^{\frac{1}{3}} K_{1/3}\left(\frac{\tau}{T'}\right) \quad (5.32)$$

where T' is related to the integral length scale, L , by the formula

$$T' = \frac{\Gamma\left(\frac{1}{3}\right)}{\Gamma\left(\frac{5}{6}\right)\sqrt{\pi}} \frac{L}{\bar{U}} \cong 1.34 \frac{L}{\bar{U}} \quad (5.33)$$

$\Gamma()$ is the gamma function and $K_{1/3}(x)$ is a modified Bessel function of the second kind and order $\nu = \frac{1}{3}$. The general definition of $K_\nu(x)$ is as follows:

$$K_\nu(x) = \frac{\pi}{2 \sin \pi \nu} \sum_{m=0}^{\infty} \frac{(x/2)^{2m}}{m!} \left[\frac{(x/2)^{-\nu}}{\Gamma(m - \nu + 1)} - \frac{(x/2)^{\nu}}{\Gamma(m + \nu + 1)} \right] \quad (5.34)$$

Step 2 – Derivation of the autocorrelation function at a point on the rotating blade: This derivation makes use of Taylor's 'frozen turbulence' hypothesis, by which the instantaneous wind speed at point C at time $t = \tau$ is assumed to be equal to that at a point B a distance $\bar{U}\tau$ upwind of C at time $t = 0$, $\bar{U}$ being the mean wind speed. Thus, referring to Figure 5.17, the autocorrelation function $\kappa_u^0(r, \tau)$ for the along-wind wind fluctuations seen by a point Q at radius r on the rotating blade is equal to the cross correlation function $\kappa_u(\vec{s}, 0)$ between the simultaneous along-wind wind fluctuations at points A and B. Here A and C are the positions of point Q at the beginning and end of time interval τ respectively, B is $\bar{U}\tau$ upwind of C and $\vec{s}$ is the vector BA. (Note that the superscript ^{0} denotes that the autocorrelation function relates to a point on a rotating blade rather than a fixed point. The same convention will be adopted in relation to power spectra.)

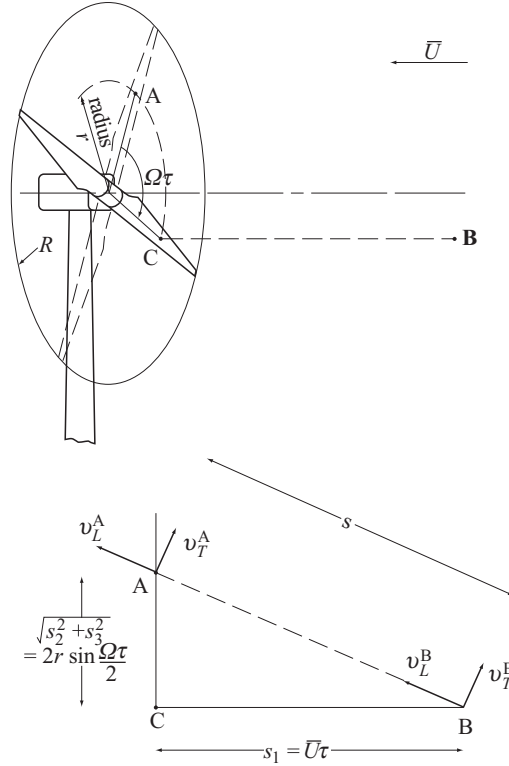


Figure 5.17 Geometry for the derivation of the velocity autocorrelation function for a point on a rotating blade

Batchelor (1953) has shown that, if the turbulence is assumed to be homogeneous and isotropic, the cross correlation function, $\kappa_u(\vec{s}, 0)$, is given by:

$$\kappa_u(\vec{s}, 0) = (\kappa_L(s) - \kappa_T(s)) \left(\frac{s_1}{s} \right)^2 + \kappa_T(s) \quad (5.35)$$

where $\kappa_L(s)$ is the cross correlation function between velocity components at points A and B, s apart, in a direction *parallel* to AB (v_L^A and v_L^B in Figure 5.17), and $\kappa_T(s)$ is the corresponding function for velocity components (v_T^A and v_T^B) in a direction *perpendicular* to AB. s_1 is the separation of points A and B measured in the along wind direction, that is, $\bar{U}\tau$. Noting that the distance between points A and C on the rotor disc is $2r \sin(\Omega\tau/2)$, we have

$$s^2 = \bar{U}^2 \tau^2 + 4r^2 \sin^2(\Omega\tau/2) \quad (5.36)$$

Hence,

$$\kappa_u(\vec{s}, 0) = \kappa_L(s) \left(\frac{\bar{U}\tau}{s} \right)^2 + \kappa_T(s) \left[1 - \left(\frac{\bar{U}\tau}{s} \right)^2 \right] = \kappa_L(s) \left(\frac{\bar{U}\tau}{s} \right)^2 + \kappa_T(s) \left(\frac{2r \sin(\Omega\tau/2)}{s} \right)^2 \quad (5.37)$$

For incompressible flow, it can also be shown (Batchelor 1953) that

$$\kappa_T(s) = \kappa_L(s) + \frac{s}{2} \frac{d\kappa_L(s)}{ds} \quad (5.38)$$

Substitution of Equation 5.38 in Equation 5.37 gives

$$\kappa_u(\vec{s}, 0) = \kappa_L(s) + \frac{s}{2} \frac{d\kappa_L(s)}{ds} \left(\frac{2r \sin(\Omega\tau/2)}{s} \right)^2 \quad (5.39)$$

When the vector $\vec{s}$ is in the along-wind direction, $\kappa_L(s)$ translates to $\kappa_u(s_1)$, which, by Taylor's 'frozen turbulence' hypothesis, equates to the autocorrelation function at a fixed point, $\kappa_u(\tau)$ (Equation 5.32), with $\tau = s_1/\bar{U}$. Thus,

$$\kappa_L(s_1) = \frac{2\sigma_u^2}{\Gamma\left(\frac{1}{3}\right)} \left(\frac{s_1/2}{T'\bar{U}} \right)^{\frac{1}{3}} K_{1/3} \left(\frac{s_1}{T'\bar{U}} \right) \quad (5.40)$$

Because the turbulence is assumed to be isotropic, $\kappa_L(s)$ is independent of the direction of the vector $\vec{s}$, so we can write, with the aid of Equation 5.33:

$$\kappa_L(s) = \frac{2\sigma_u^2}{\Gamma\left(\frac{1}{3}\right)} \left(\frac{s/2}{T'\bar{U}} \right)^{\frac{1}{3}} K_{1/3} \left(\frac{s}{T'\bar{U}} \right) = \frac{2\sigma_u^2}{\Gamma\left(\frac{1}{3}\right)} \left(\frac{s/2}{1.34L} \right)^{\frac{1}{3}} K_{1/3} \left(\frac{s}{1.34L} \right) \quad (5.41)$$

Noting that $(d/dx)[x^\nu K_\nu(x)] = -x^\nu K_{(1-\nu)}(x)$, the following expression for the autocorrelation function for the along-wind fluctuations at a point at radius r on the rotating blade is obtained by substituting Equation 5.41 in Equation 5.39:

$$\begin{aligned} \kappa_u^o(r, \tau) = \kappa_u(\vec{s}, 0) &= \frac{2\sigma_u^2}{\Gamma\left(\frac{1}{3}\right)} \left(\frac{s/2}{1.34L} \right)^{\frac{1}{3}} \left[K_{1/3} \left(\frac{s}{1.34L} \right) - \frac{s}{2(1.34L)} K_{2/3} \left(\frac{s}{1.34L} \right) \right. \\ &\quad \left. \times \left(\frac{2r \sin(\Omega\tau/2)}{s} \right)^2 \right] \end{aligned} \quad (5.42)$$

where s is defined in terms of τ by Equation 5.36 above.

Step 3 – Derivation of the power spectrum seen by a point on the rotating blade: the rotationally sampled spectrum is obtained by taking the Fourier transform of $\kappa_u^o(r, \tau)$ from Equation 5.42:

$$\begin{aligned} S_u^o(n) &= 4 \int_0^\infty \kappa_u^o(r, \tau) \cos 2\pi n\tau \, d\tau \\ &= 2 \int_{-\infty}^\infty \kappa_u^o(r, \tau) \cos 2\pi n\tau \, d\tau \quad \text{as } \kappa_u^o(r, \tau) = \kappa_u^o(r, -\tau) \end{aligned} \quad (5.43)$$

As no analytical solution has been found for the integral, a solution has to be obtained numerically using a discrete Fourier transform (DFT). First the limits of integration are reduced to $-T/2, +T/2$, as $\kappa_u^o(r, \tau)$ tends to zero for large τ . Then the limits of integration are altered to $0, T$ with $\kappa_u^o(r, \tau)$ set equal to $\kappa_u^o(r, T - \tau)$ for $\tau > T/2$, as $\kappa_u^o(r, \tau)$ is now assumed to be periodic with period T . Thus,

$$S_u^o(n) = 2 \int_0^T \kappa_u^{*o}(r, \tau) \cos 2\pi n \tau d\tau \quad (5.44)$$

where the asterisk denotes that $\kappa_u^o(r, \tau)$ is ‘reflected’ for $T > T/2$. The discrete Fourier transform then becomes

$$S_u^o(n_k) = 2T \left[\frac{1}{N} \sum_{p=0}^{N-1} \kappa_u^{*o}(r, pT/N) \cos(2\pi kp/N) \right] \quad (5.45)$$

Here, N is the number of points taken in the time series of $\kappa_u^{*o}(r, pT/N)$, and the power spectral density is calculated at the frequencies $n_k = k/T$ for $k = 0, 1, 2 \dots N - 1$. The expression in square brackets can be evaluated using a standard fast Fourier transform (FFT), provided N is chosen equal to a power of 2. Clearly N should be as large as possible if a wide range of frequencies is to be covered at high resolution. Just as $\kappa_u^{*o}(r, \tau)$ is symmetrical about $T/2$, the values of $S_u^o(n_k)$ obtained from the FFT are symmetrical about the mid-range frequency of $N/2T$, and the values above this frequency have no real meaning. Moreover, the values of power spectral density calculated by the DFT at frequencies approaching $N/(2T)$ will be in error as a result of aliasing, because these are falsely distorted by frequency components above $N/2T$ which contribute to the $\kappa_u^{*o}(r, pT/N)$ series. Assuming that the calculated spectral densities are valid up to a frequency of $N/4T$, then the selection of $T = 200$ sec and $N = 4096$ would enable the FFT to give useful results up to a frequency of about 5 Hz at a frequency interval of 0.005 Hz.

Example 5.2

As an illustration, results have been derived for points on a 40 m radius T40 blade rotating at 15 rpm in a mean wind speed of 8 m/s. Following the recommendations for the use of the Von Karman isotropic turbulence model in IEC 61400–1 Edition 2 (1999), the isotropic integral length scale, L , is taken as 3.5 times the IEC 61400–1 turbulence scale parameter, Λ_1 . However, Λ_1 is taken as 42 m, as recommended in Edition 3 (2005) for a hub height exceeding 60 m, giving $L = 147$ m. (These values compare with $\Lambda_1 = 21$ m and $L = 73.5$ m recommended in Edition 2). Figure 5.18 shows how the normalised autocorrelation function, $\rho_u^o(r, \tau) (= \kappa_u^o(r, \tau)/\sigma_u^2)$, for the longitudinal wind fluctuations varies with the number of rotor revolutions at 40 m, 20 m and 0 m radii. For $r = 20$ m, and even more so for $r = 40$ m, these curves display pronounced peaks after each full revolution, when the blade may be thought of as encountering the initial gust or lull once more.

Figure 5.19a shows the corresponding rotationally sampled power spectral density function, $R_u^o(r, n) (= nS_u^o(r, n)/\sigma_u^2)$, plotted against frequency, n , using a logarithmic scale for the latter. It is clear that there is a substantial shift of the frequency content of the spectrum to the frequency of rotation and, to a lesser degree to its harmonics, with the extent of the shift

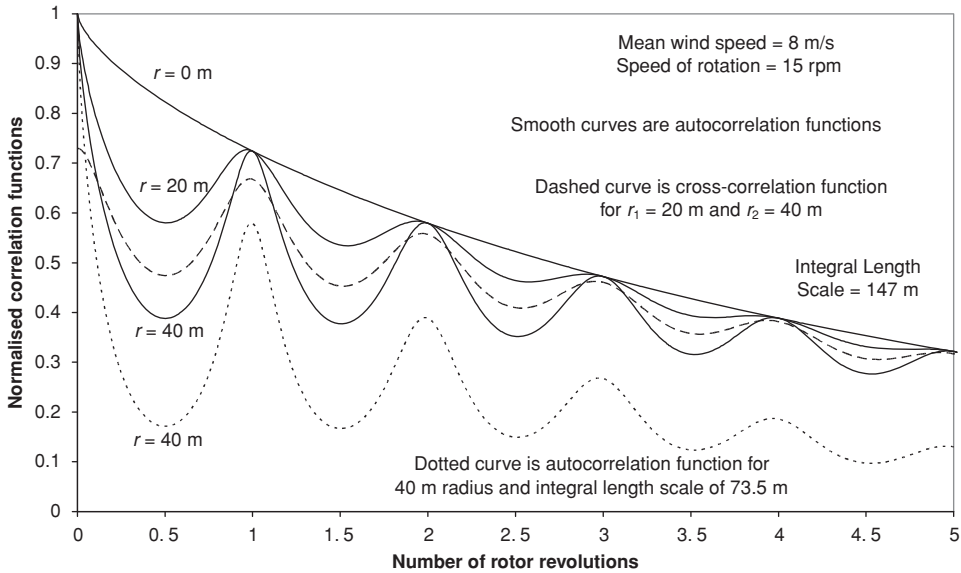


Figure 5.18 Normalised autocorrelation and cross correlation functions for along-wind wind fluctuations as seen by points on a rotating blade at different radii

increasing with radius. Figure 5.19b is a repeat of Figure 5.19a, but with a logarithmic scale used on both axes.

It is instructive to consider how the various input parameters affect the shift of energy to the rotational frequency. As $\kappa_u^o(r, \tau) = \kappa_u(\vec{s}, 0)$ decreases monotonically with increasing s , Equation 5.36 indicates that the depth of the troughs in this function – and hence the transfer of energy to the rotational frequency – increases roughly in proportion to the tip speed ratio, $\Omega r / \bar{U}$, and will thus be most significant for fixed-speed two-bladed machines (which generally rotate faster than three-bladed ones) in low wind speeds.

Effect of reduced length scale

The effect of adopting a reduced isotropic integral length scale, L , of 73.5 m as opposed to 147 m on the auto correlation function and the rotationally sampled power spectrum is illustrated for a radius of 40 m in Figures 5.18 and 5.20 respectively.

It is seen that, despite the more rapid attenuation of auto-correlation function, the reduction of length scale has negligible effect on the spectral peak at the rotational frequency.

Rotationally sampled cross spectra

The expressions for the spectra of blade bending moments and shears are normally functions of entities known as rotationally sampled cross spectra for pairs of points along the blade, which are analogous to the rotationally sampled ordinary spectra for single points described

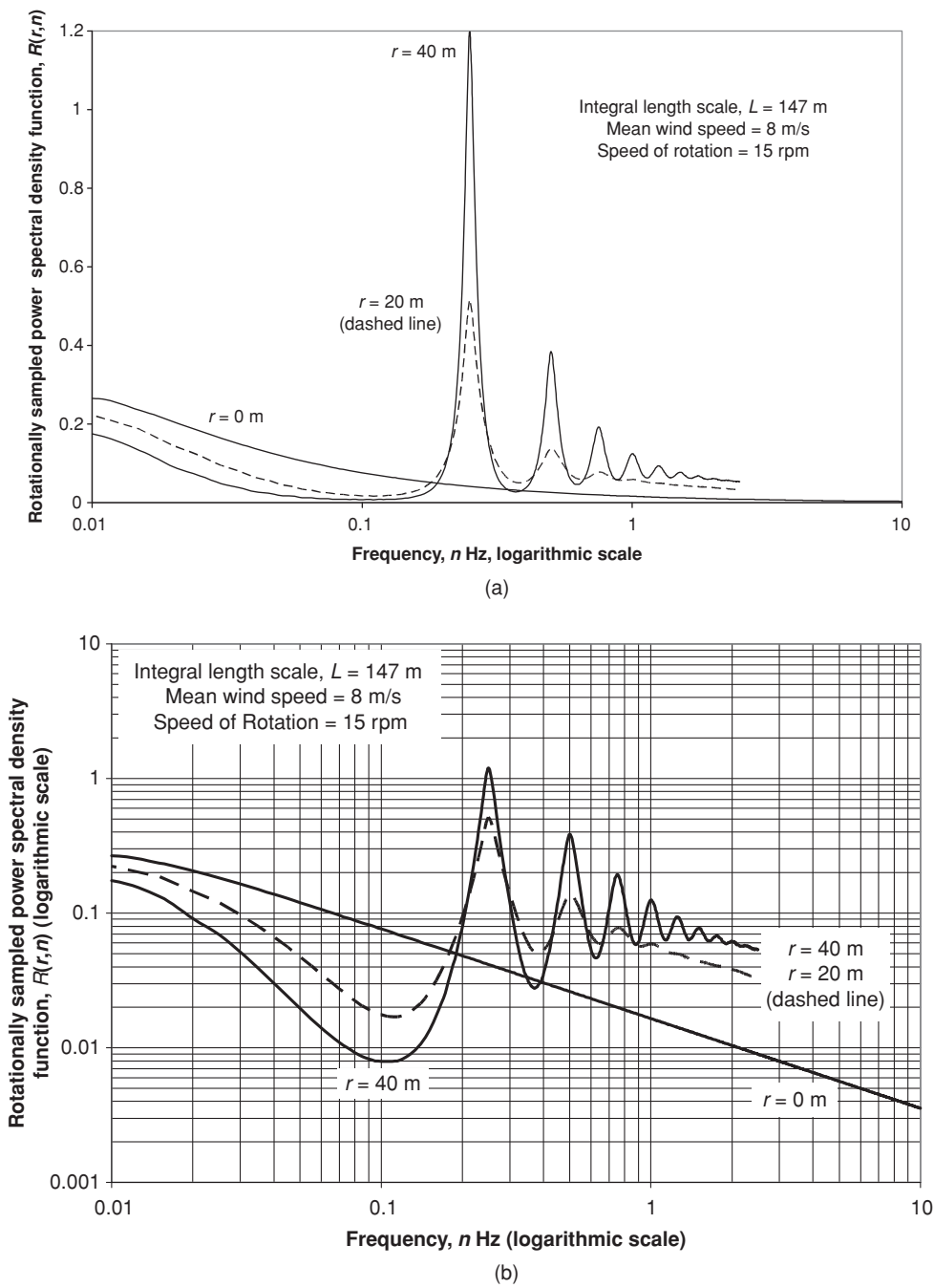


Figure 5.19 (a) Rotationally sampled power spectra of longitudinal wind speed fluctuations at different radii; (b) Rotationally sampled power spectra of longitudinal wind speed fluctuations at different radii: log-log plot

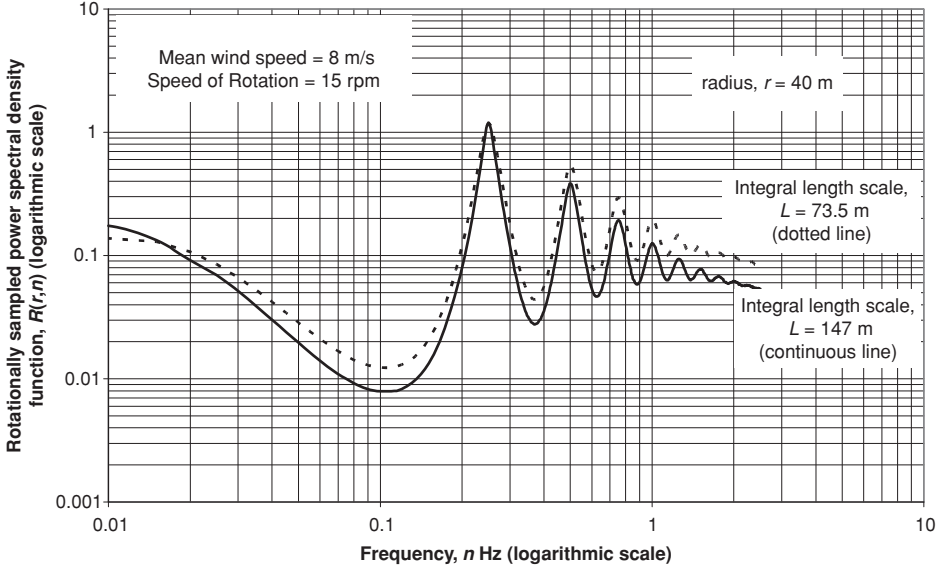


Figure 5.20 Comparison of rotationally sampled power spectra at 40 m radius for different integral length scales

above. The cross spectrum for a pair of points at radii r_1 and r_2 on a rotating blade is thus related to the corresponding cross correlation function by the Fourier transform pair

$$S_u^o(r_1, r_2, n) = 4 \int_0^\infty \kappa_u^o(r_1, r_2, \tau) \cos 2\pi n \tau d\tau \quad (5.46a)$$

$$\kappa_u^o(r_1, r_2, \tau) = \int_0^\infty S_u^o(r_1, r_2, n) \cos 2\pi n \tau dn \quad (5.46b)$$

Setting $\tau = 0$ in Equation 5.46b gives

$$\kappa_u^o(r_1, r_2, 0) = \int_0^\infty S_u^o(r_1, r_2, n) dn \quad (5.47)$$

which, when substituted into the expression for the standard deviation of the blade root bending moment in Equation 5.27 gives

$$\sigma_M^2 = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_0^R \int_0^R \left[\int_0^\infty S_u^o(r_1, r_2, n) dn \right] c(r_1) c(r_2) r_1^2 r_2^2 dr_1 dr_2 \quad (5.48)$$

From this, it can be deduced that the power spectrum of the blade root bending moment is

$$S_M(n) = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_0^R \int_0^R S_u^o(r_1, r_2, n) c(r_1) c(r_2) r_1^2 r_2^2 dr_1 dr_2 \quad (5.49)$$

The derivation of the rotationally sampled cross spectrum, $S_u^o(r_1, r_2, n)$, exactly parallels the derivation of the rotationally sampled single point spectrum given above, with the cross correlation function $\kappa_u^o(r_1, r_2, \tau)$ between the longitudinal wind fluctuations at points at radii r_1 and r_2 on the rotating blade replacing the autocorrelation function in step 2. Here the expression for the separation distance, s , given in Equation 5.36, is replaced by

$$s^2 = \bar{U}^2 \tau^2 + r_1^2 + r_2^2 - 2r_1 r_2 \cos \Omega \tau \quad (5.50)$$

The expression for the cross correlation function thus becomes:

$$\begin{aligned} \kappa_u^o(r_1, r_2, \tau) = & \frac{2\sigma_u^2}{\Gamma\left(\frac{1}{3}\right)} \left(\frac{s/2}{1.34L}\right)^{\frac{1}{3}} \left[K_{1/3}\left(\frac{s}{1.34L}\right) - \frac{s}{2(1.34L)} K_{2/3}\left(\frac{s}{1.34L}\right) \right. \\ & \left. \times \left(\frac{r_1^2 + r_2^2 - 2r_1 r_2 \cos(\Omega \tau)}{s^2} \right) \right] \end{aligned} \quad (5.51)$$

with s defined by Equation 5.50.

The form of the resulting normalised cross correlation function, $\rho_u^o(r_1, r_2, \tau) = \kappa_u^o(r_1, r_2, \tau)/\sigma_u^2$, is illustrated in Figure 5.18 for the case considered in Example 5.2, taking $r_1 = 20$ m and $r_2 = 40$ m. In Figure 5.21, the rotationally sampled cross spectrum for this case is compared with the rotationally sampled single point spectra or ‘autospectra’ at these radii. It can be seen that the form of the cross spectrum curve is similar to that of the autospectra with a pronounced peak at the rotational frequency roughly midway between the

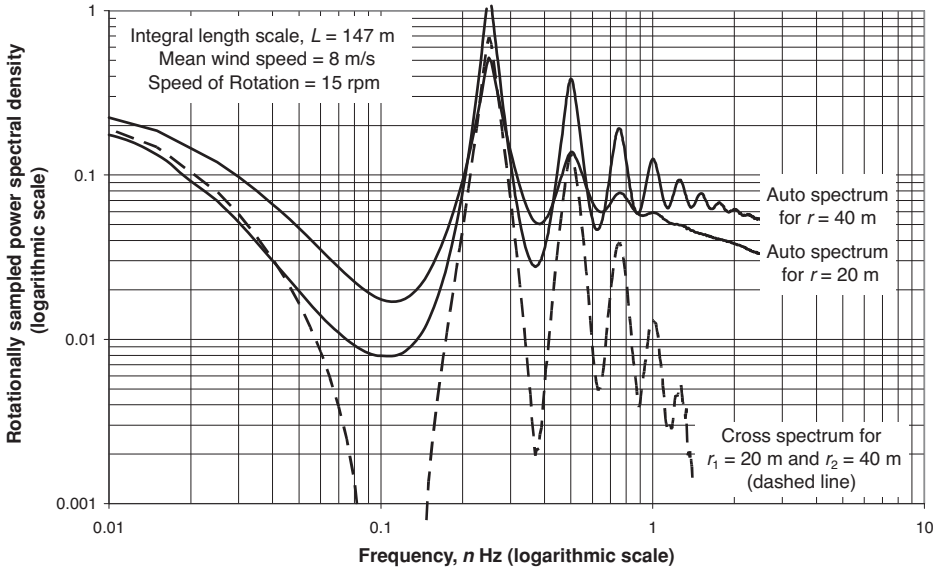


Figure 5.21 Rotationally sampled cross spectrum of longitudinal wind speed fluctuations at 20 m and 40 m radii compared with auto spectra: log-log plot

peaks of the two autospectra. At higher frequencies, however, the cross spectrum falls away much more rapidly.

The evaluation of the power spectrum of the blade root bending moment is, in practice, carried out using summations to approximate to the integrals in Equation 5.49, as follows:

$$S_M(n) = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \sum_j \sum_k S_u^o(r_j, r_k, n) c(r_j) c(r_k) r_j^2 r_k^2 (\Delta r)^2 \quad (5.52)$$

Limitations of analysis in the frequency domain

As noted at the beginning of this section, analysis of stochastic aerodynamic loads in the frequency domain depends for its validity on a *linear* relationship between the incident wind speed and the blade loading. Thus, the method becomes increasingly inaccurate for pitch regulated machines as winds approach the cut-out value and breaks down completely for stall-regulated machines once the wind speed is high enough to cause stall. In order to avoid these limitations, it is necessary to carry out the analysis in the time domain.

5.7.6 Stochastic aerodynamic loads: analysis in the time domain

Wind simulation

The analysis of stochastic aerodynamic loads in the time domain requires, as input, a simulated wind field extending over the area of the rotor disc and over time. Typically, this is obtained by generating simultaneous time histories at points over the rotor disc, which have appropriate statistical properties both individually, and in relation to each other. Thus, the power spectrum of each time history should conform to one of the standard power spectra (e.g. Von Karman or Kaimal), and the normalised cross spectrum (otherwise known as the coherence function) of the time histories at two different points should equate to the coherence function corresponding to the chosen power spectrum and the distance separating the points. For example, the coherence of the longitudinal component of turbulence corresponding to the Kaimal power spectrum for points j and k separated by a distance Δs_{jk} perpendicular to the wind direction is:

$$C_{jk}(n) = C(\Delta s_{jk}, n) = \frac{S_{jk}(n)}{S_u(n)} = \exp \left(-H \cdot \Delta s_{jk} \sqrt{\left(\frac{n}{U} \right)^2 + \left(\frac{0.12}{L} \right)^2} \right) \quad (5.53)$$

The constant H was specified as 8.8 in IEC 61400–1 Edition 2, but increased to 12 in Edition 3. The $(0.12/L)$ term is negligible except at frequencies below 0.01 Hz. (Note that coherence is sometimes termed coherency, and that some authors define coherence as the *square* of the normalised cross spectrum). See Section 2.6.6 for details of the coherence corresponding to the Von Karman spectrum.

Three distinct approaches have been developed for generating simulation time histories as follows:

1. The transformational method, based on filtering Gaussian white noise signals.

2. The correlation method, in which the velocity of a small body of air at the end of a time step is calculated as the sum of a velocity correlated with the velocity at the start of the time step and a random, uncorrelated increment.
3. The harmonic series method, involving the summation of a series of cosine waves at different frequencies with amplitudes weighted in accordance with the power spectrum.

This last method is probably now the one in widest use, and is described in more detail below. The description is based on that given in Veers (1988).

Wind simulation by the harmonic series method

The spectral properties of the wind speed fluctuations at N points can be described by a spectral matrix, $\mathbf{S}$, in which the diagonal terms are the *double-sided* single point power spectral densities at each point, $S_{kk}(n)$, and the off-diagonal terms are the cross-spectral densities, $S_{jk}(n)$, also double sided. This matrix is equated to the product of a triangular transformation matrix, $\mathbf{H}$, and its transpose, $\mathbf{H}^T$, as follows:

$$\begin{bmatrix} S_{11} & S_{21} & S_{31} & \dots \\ S_{21} & S_{22} & S_{32} & \dots \\ S_{31} & S_{32} & S_{33} & \dots \\ \dots & \dots & \dots & S_{NN} \end{bmatrix} = \begin{bmatrix} H_{11} & & & \\ H_{21} & H_{22} & & \\ H_{31} & H_{32} & H_{33} & \\ \dots & \dots & \dots & H_{NN} \end{bmatrix} \begin{bmatrix} H_{11} & H_{21} & H_{31} & \dots \\ & H_{22} & H_{32} & \dots \\ & & H_{33} & \dots \\ & & & H_{NN} \end{bmatrix}$$

resulting in a set of $N(N + 1)$ equations linking the elements of the $\mathbf{S}$ matrix to the elements of the $\mathbf{H}$ matrix as follows:

$$\begin{aligned} S_{11} &= H_{11}^2 & S_{21} &= H_{21} \cdot H_{11} & S_{22} &= H_{21}^2 + H_{22}^2 & S_{31} &= H_{31} \cdot H_{11} \\ S_{32} &= H_{31} \cdot H_{21} + H_{32} \cdot H_{22} & S_{33} &= H_{31}^2 + H_{32}^2 + H_{33}^2 \end{aligned}$$

$$S_{jk} = \sum_{l=1}^k H_{jl} \cdot H_{kl} \quad S_{kk} = \sum_{l=1}^k H_{kl}^2 \quad (5.54)$$

As with the elements of the $\mathbf{S}$ matrix, the elements of the $\mathbf{H}$ matrix are all double-sided functions of frequency, n .

Noting that the expression for the power spectral density S_{kk} resembles that for the variance of the sum of group of k independent variables, it is apparent that the elements of the $\mathbf{H}$ matrix can be considered as the weighting factors for the linear combination of N independent, unit magnitude, white noise inputs to yield N correlated outputs with the correct spectral matrix. Thus, the elements in the j th row of $\mathbf{H}$ are the weighting factors for the inputs contributing to the output at point j . The formula for the linear combination is

$$u_j(n) = \sum_{k=1}^j H_{jk}(n) \cdot \Delta n \cdot \exp(-i\theta_k(n)) \quad (5.55)$$

where $u_j(n)$ is the complex coefficient of the discretised frequency component at n Hz of the simulated wind speed at point j . The frequency bandwidth is Δn . $\theta_k(n)$ is the phase angle associated with the n Hz frequency component at point k , and is a random variable uniformly distributed over the interval $0 - 2\pi$.

The values of the weighting factors, H_{jk} , which are $N(N + 1)$ in number are derived from the Equations 5.54, giving:

$$H_{11} = \sqrt{S_{11}} \quad H_{21} = S_{21}/H_{11} \quad H_{22} = \sqrt{S_{22} - H_{21}^2} \quad H_{31} = S_{31}/H_{11} \quad \text{etc.} \quad (5.56)$$

Hence

$$\begin{aligned} u_1(n) &= \sqrt{S_{11}(n)} \cdot \Delta n \cdot \exp(-i\theta_1(n)) \\ u_2(n) &= \sqrt{S_{22}(n)} \cdot \Delta n \left[C_{21}(n) \cdot \exp(-i\theta_1(n)) + \sqrt{1 - C_{21}^2(n)} \cdot \exp(-i\theta_2(n)) \right] \quad \text{etc.} \end{aligned} \quad (5.57)$$

Time series for the wind speed fluctuations are obtained by taking the inverse discrete Fourier transform of the coefficients $u_j(n)$ at each point j . Lateral and vertical wind speed fluctuations can also be simulated, if desired, using the same method. As an illustration, examples of time series derived by this method for two points 10 m apart are shown in Figure 5.22, based on the Von Karman spectrum. The mean wind speed and integral length scale $^x L_u$ in this example are taken as 10 m/s and 73.5 m respectively, giving an integral time scale, $^x L_u / \bar{U}$ of 7.35 seconds.

In his 1988 paper, Veers pointed out that computation time required can be reduced by arranging for the simulated wind speed to be calculated at each point only at those times

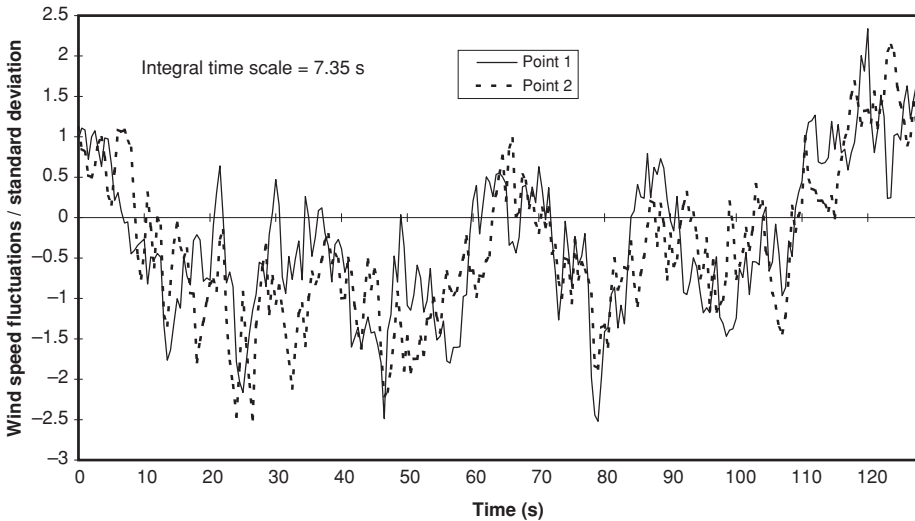


Figure 5.22 Simulated time series of wind speed fluctuations at two points 10 m apart for mean wind speed of 10 m/s

when a blade is passing – i.e. at a frequency of $\Omega B/2\pi$, where B is the number of blades. This is achieved by applying a phase shift to each frequency component at each point of $\psi_j \cdot n \cdot 2\pi/\Omega$, where ψ_j is the azimuth angle of point j .

Blade load time histories

Once the simulated wind speed time histories have been generated across the grid, the calculation of blade load histories at different radii can begin. If the wake is assumed to be ‘frozen’, then the axial induced velocity, $a\bar{U}$, and the tangential induced velocity, $a'\Omega r$, are taken as remaining constant over time, at each radius, at the values calculated for a steady wind speed of $\bar{U}$. The instantaneous value of the flow angle, ϕ , and, hence, the values of the lift and drag coefficients, may then be calculated directly from the instantaneous value of the wind speed fluctuation (including lateral and vertical components, if calculated) by means of the velocity diagram.

Alternatively, an equilibrium wake may be assumed. In this case, the induced velocities are taken to vary continuously so that the momentum equations are satisfied at each blade element at all times. Obviously, this requires that these equations are solved afresh at each time step, which is computationally much more demanding.

Neither the equilibrium wake model nor the frozen wake model provide an accurate description of wake behaviour. A better model is provided by unsteady flow theory, which assumes that there is some delay before induced velocities react to changes in the incident wind field. See Section 4.4.

Note that, if desired, the spatial wind variations causing deterministic loading can be included in the simulated wind field, enabling the combined deterministic and stochastic loading on the blade to be calculated in a single operation.

5.7.7 Extreme loads

The derivation of extreme loads should properly take into account dynamic effects, which form the subject of the next section. However, in the interests of clarity, this section will be restricted to the consideration of extreme loads in the absence of dynamic effects.

As described in Section 5.4.1, it was customary for wind turbine design codes to specify extreme operating load cases in terms of deterministic gusts. The extreme blade loadings are then evaluated at intervals over the duration of the gust, using blade element and momentum theory as described in Section 5.7.2.

Although deterministic gusts have the advantage of clarity of definition, they are essentially arbitrary in nature. The alternative approach of employing a stochastic representation of the wind provides a much more realistic description of the wind itself, and has been adopted to a greater extent in IEC 61400–1 Edition 3. Although stochastic representation of the wind lends itself to analysis in the frequency domain, non-linearities introduced by stall invalidate the method, so the standard specifies analysis in the time domain using simulations. Nevertheless, analysis in the frequency domain can provide useful insights in the absence of stall and is considered further in the paragraphs which follow.

Normally the loading under investigation, for example the blade root bending moment, will contain both periodic and random components. Although it is straightforward to predict the extreme values of each component independently, the prediction of the extreme value

of the combined signal is quite involved. Madsen *et al.* (1984) have proposed the following simple, approximate approach, and have demonstrated that it is reasonably accurate.

The periodic component, $z(t)$, is considered as an equivalent 3 level square wave, in which the variable takes the maximum, mean (μ_z), and minimum values of the original waveform, for proportions ε_1 , ε_2 and ε_3 of the wave period respectively. It is easy to show that:

$$\varepsilon_1 = \frac{\sigma_z^2}{(z_{\max} - \mu_z)(z_{\max} - z_{\min})} \quad \varepsilon_3 = \frac{\sigma_z^2}{(\mu_z - z_{\min})(z_{\max} - z_{\min})} \quad (5.58)$$

Extreme values of the combined signal are only assumed to occur during the proportion of the time, ε_1 , for which the square wave representation of the periodic component is at the maximum value, $z_{\max}$.

Davenport (1964) gives the following formula for the extreme value of a random variable over a time interval T :

$$\frac{x_{\max}}{\sigma_x} = \sqrt{2 \ln(\nu T)} + \frac{\gamma}{\sqrt{2 \ln(\nu T)}} \quad (5.59)$$

where ν is the zero up-crossing frequency (i.e. the number of times per second the variable changes from negative to positive) given by Equation A5.46 and $\gamma = 0.5772$ (Euler's constant). Thus, the extreme value of the combined periodic and random components is taken to be

$$z_{\max} + x_{\max} = z_{\max} + \sigma_x \left(\sqrt{2 \ln(\nu \varepsilon_1 T)} + \frac{\gamma}{\sqrt{2 \ln(\nu \varepsilon_1 T)}} \right) = z_{\max} + g_1 \cdot \sigma_x \quad (5.60)$$

where g_1 is termed the peak factor.

The variation of $x_{\max}/\sigma_x$ with exposure time, T , is shown in Table 5.4 for a zero up-crossing frequency of 1 Hz. The periodic component is assumed to be a simple sinusoid, giving $\varepsilon_1 = 0.25$.

The method for determining the extreme load described above has to be applied with caution when the wind fluctuations exceed the rated wind speed. In the case of a stall-regulated machine, the linearity assumption breaks down completely, invalidating the method. With pitch-regulated machines, however, the blade pitch will respond to wind fluctuations at frequencies below, say, half the rotor rotational frequency in order to limit power, causing a parallel reduction in blade loading. This will modify the spectrum of blade loading dramatically, effectively removing the frequency components below the pitch system cut-off frequency, and consequently reducing the magnitude of σ_x to be substituted in Equation 5.60.

Table 5.4 Extreme values of random component for different exposure times

T	1 minute	10 minutes	1 hour	10 hours	100 hours	1000 hours	1 year
T (secs)	60	600	3,600	36,000	360,000	3,600,000	31,536,000
$\varepsilon_1 T$ (secs)	15	150	900	9,000	90,000	900,000	7,884,000
$x_{\max}/\sigma_x$	2.57	3.35	3.84	4.40	4.90	5.35	5.74

To illustrate the method, the procedure for calculating the extreme flapwise blade root bending moment of a pitch regulated machine operating at rated wind speed is described below:

- 1) Equation 5.48 for the standard deviation of the random component of blade root bending moment is first modified to eliminate the contribution of frequencies below half the rotational speed to account for the blade pitching response, and then discretised to give:

$$\sigma_M^2 = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \sum_{j=1}^m \sum_{k=1}^m \left[\int_{\Omega/2}^{\infty} S_u^o(r_j, r_k, n) dn \right] c(r_j) c(r_k) r_j^2 r_k^2 \cdot (\Delta r)^2 \quad (5.61)$$

Here the blade is assumed to be divided up into m sections of equal length $\Delta r = R/m$.

- 2) After evaluation of the integrals of the $m(m+1)/2$ different curtailed rotational spectra, the standard deviation of the blade root bending moment is obtained from Equation 5.61.
- 3) The time, T , that the machine spends in a wind speed band centred on the rated wind speed is estimated using the Weibull curve, and multiplied in turn by the factor ε_1 appropriate to the waveform of the periodic component of blade root bending moment and by the zero up-crossing frequency of the random root bending moment fluctuations, to give the effective number of peaks, $\nu \varepsilon_1 T$.
- 4) The predicted extreme value of the total moment is calculated by substituting the standard deviation of the blade root bending moment, $\sigma_M (= \sigma_x)$, the effective number of peaks, $\nu \varepsilon_1 T$, and the extreme value of the periodic moment into Equation 5.60.

In the case of a machine with a rated wind speed of 13 m/s operating at a site with an annual mean of 7 m/s, the expected proportion of the time spent operating within a 2 m/s wide band centred on the rated wind speed is 5.6%. Taking the machine lifetime as 20 years, a zero up-crossing frequency equal to the rotational speed of 15 rpm (0.25 Hz) and $\varepsilon_1 = 0.25$, this results in a peak factor, g_1 , of 5.5. For the 80 m diameter machine considered in Section 5.7.2, a turbulence intensity of 20% and a turbulence isotropic length scale of 147 m, the standard deviation of the random component of blade root bending moment given by Equation 5.60 is 315 kNm, resulting in a peak value of about 1730 KNm. This compares with the extreme value of the periodic component, including wind shear, of about 1830 KNm. It should be emphasised that the peak value of the random component quoted is a theoretical one – that is, it assumes the linearity assumptions are maintained even for the large wind speed fluctuation needed to generate this moment. In practice, a machine operating in a steady wind speed equal to rated is usually not all that far from stall, so the larger fluctuations may induce stall. In this example, the square root of the weighted mean of the integrals of all the curtailed rotational spectra is about $0.5\sigma_u$, so the idealised uniform wind speed fluctuation equivalent to the extreme root moment is about $0.20 \times 13 \times 0.5 \times 5.5 = 7.2$ m/s.

The method outlined above has more validity at higher wind speeds, when the blades are pitched back, and are operating further away from stall. However, it is important to note that the other linearity assumption used in deriving Equation 5.26, namely that ϕ is small, becomes increasingly in error.

It will be now be evident that the calculation of stochastic extreme loads is fraught with difficulties because non-linearities are likely to arise as the extremes are approached. In so

far as lift forces ‘saturate’ due to stall, or even drop back, as wind speed increases, a crude and simple approach to extreme out-of-plane operational loads is to calculate an upper bound based on the maximum lift coefficient for the local aerofoil section and the relative air velocity, W . The induction factors will be small, and can be ignored.

The most sophisticated approach, however, is to analyse the loads generated by a simulated wind field. As computing costs normally restrict the length of simulated ‘campaigns’ to a few hundred seconds or less, statistical methods have to be used to extrapolate from the extreme values of loadings calculated during the campaign to the extreme values to be expected over the machine design life.

One method, which is discussed by Thomsen and Madsen (1997), is to use Equation 5.60 with T set equal to the appropriate exposure period over the machine design life, and values of $z_{\max}$ and σ_x , abstracted from the simulation time history with the aid of azimuthal binning to separate the periodic and stochastic components. The danger of this approach with simulations of short duration is that the azimuthal binning process treats some load fluctuations due to the slicing of low frequency gusts as periodic rather than stochastic, so that the standard deviation of the stochastic component, σ_x , is underestimated.

Extrapolation techniques are considered further in Section 5.14.

5.8 Blade dynamic response

5.8.1 Modal analysis

Although dynamic loads on the blades will, in general, also excite the tower dynamics, tower head motion will initially be excluded from consideration in order to focus on the blade dynamic behaviour itself. The treatment is further limited to the response of blades in unstalled flow because of the inherent difficulty in predicting stalled behaviour.

The equation of motion for a blade element at radius r subject to a time varying load $q(r, t)$ per unit length in the out-of-plane direction is

$$m(r)\ddot{x} + \hat{c}(r)\dot{x} + \frac{\partial^2}{\partial r^2} \left[EI(r) \frac{\partial^2 x}{\partial r^2} \right] = q(r, t) \quad (5.62)$$

where the terms on the left hand side are the loads on the element due to inertia, damping and flexural stiffness respectively. $I(r)$ is the second moment of area of the blade cross-section about the weak principal axis (which for this purpose is assumed to lie in the plane of rotation) and x is the out-of-plane displacement. The expressions $m(r)$ and $\hat{c}(r)$ denote mass per unit length and damping per unit length respectively.

The dynamic response of a cantilever blade to the fluctuating aerodynamic loads upon it is most conveniently investigated by means of modal analysis, in which the excitations of the various different natural modes of vibration are computed separately and the results superposed, as follows:

$$x(t, r) = \sum_{j=1}^{\infty} f_j(t) \mu_j(r) \quad (5.63)$$

where $\mu_j(r)$ is the j th mode shape, arbitrarily assumed to have a value of unity at the tip, and $f_j(t)$ is the variation of tip displacement with time. Equation 5.62 then becomes

$$\sum_{j=1}^{\infty} \left\{ m(r)\mu_j(r)\ddot{f}_j(t) + \hat{c}(r)\mu_j(r)\dot{f}_j(t) + \frac{d^2}{dr^2} \left[EI(r)\frac{d^2\mu_j(r)}{dr^2} \right] f_j(t) \right\} = q(r, t) \quad (5.64)$$

For low levels of damping the beam natural frequencies are given by

$$m(r)\omega_j^2\mu_j(r) = \frac{d^2}{dr^2} \left[EI(r)\frac{d^2\mu_j(r)}{dr^2} \right] \quad (5.65)$$

so Equation 5.64 becomes

$$\sum_{j=1}^{\infty} \left\{ m(r)\mu_j(r)\ddot{f}_j(t) + \hat{c}(r)\mu_j(r)\dot{f}_j(t) + m(r)\omega_j^2\mu_j(r)f_j(t) \right\} = q(r, t) \quad (5.66)$$

Multiplying both sides by $\mu_i(r)$, and integrating over the length of the blade, R , gives:

$$\begin{aligned} \sum_{j=1}^{\infty} \left\{ \int_0^R m(r)\mu_i(r)\mu_j(r)\ddot{f}_j(t)dr + \int_0^R \hat{c}(r)\mu_i(r)\mu_j(r)\dot{f}_j(t)dr \right. \\ \left. + \int_0^R m(r)\omega_j^2\mu_i(r)\mu_j(r)f_j(t)dr \right\} = \int_0^R \mu_i(r)q(r, t)dr \end{aligned} \quad (5.67)$$

The undamped mode shapes are orthogonal as a result of Bettis's law, so they satisfy the orthogonality condition:

$$\int_0^R m(r)\mu_i(r)\mu_j(r)dr = 0 \quad \text{for } i \neq j \quad (5.68)$$

If we assume that the variation of the damping per unit length along the blade, $\hat{c}(r)$, is proportional to the variation in mass per unit length, $m(r)$, i.e. $\hat{c}(r) = a \cdot m(r)$, then

$$\int_0^R \hat{c}(r)\mu_i(r)\mu_j(r)dr = 0 \quad \text{for } i \neq j \quad (5.69)$$

As a result, all the cross terms on the left hand side of Equation 5.67 drop out, and it reduces to

$$m_i\ddot{f}_i(t) + c_i\dot{f}_i(t) + m_i\omega_i^2f_i(t) = \int_0^R \mu_i(r)q(r, t)dr \quad (5.70)$$

where $m_i = \int_0^R m(r)\mu_i^2(r)dr$ and is known as the generalised mass, $c_i = \int_0^R \hat{c}(r)\mu_i^2(r)dr$ and $\int_0^R \mu_i(r)q(r, t)dr = Q_i(t)$ is termed the generalised fluctuating load with respect to the i th

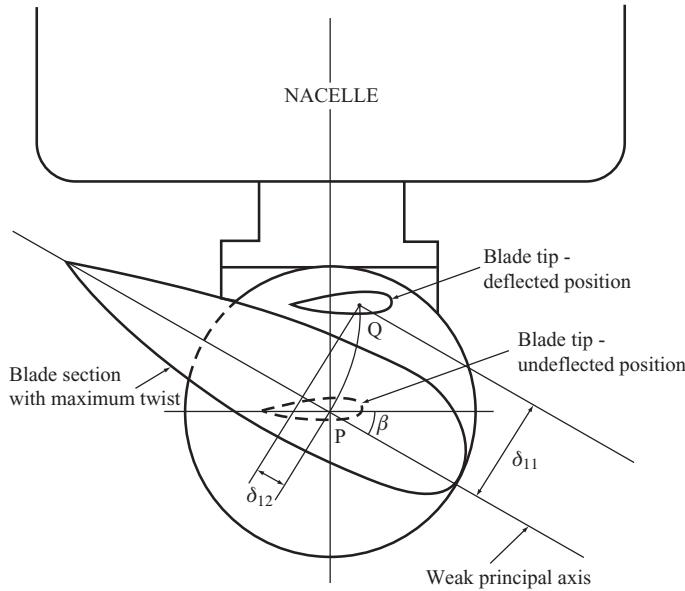


Figure 5.23 Deflection of tip due to flapwise bending of twisted blade (viewed along blade axis)

mode. Equation 5.70 is the fundamental equation governing modal response to time varying loading.

Blade flexural vibrations occur in both the flapwise and edgewise directions (i.e. about the weak and strong principal axes respectively). Blades are typically twisted some 15° , so the weak principal axis does not, in general, lie in the plane of rotation as assumed above. Consequently blade flexure about one principal axis inevitably results in some blade movement perpendicular to the other. This is illustrated in Figure 5.23, in which the maximum blade twist near the root has been exaggerated for clarity. Point P represents the undeflected position of the blade tip, point Q represents the deflected position as a result of flexure about the weak principal axis, and the line between them is built up of the contributions to the tip deflection made by flexure of each element along the blade, $M(R - r)\Delta r/EI$.

The interaction between flexure about the two principal axes can be explored with the help of some simplifying assumptions. If M varies as $(R - r)$ for the first mode and I varies as $(R - r)^2$, then each of the tip deflection contributions referred to above are equal, so that, for a linear twist distribution, the line PQ is the arc of a circle. If the twist varies between zero at the tip and a maximum value of β towards the root then the tip deflection, δ_{12} , in the direction of the weak principal axis, at the blade section with maximum twist, is $\beta/2$ times the tip deflection, δ_{11} , perpendicular to this axis. Hence in the case of $\beta = 15^\circ$, the ratio δ_{12}/δ_{11} approximates to 0.13, with the result that blade first mode flapwise oscillations will result in some relatively small simultaneous edgewise inertia loadings. These will not excite significant edgewise oscillations, because the edgewise first mode natural frequency is typically about double the flapwise one.

It can be seen from the above that the effects of interaction between flapwise and edgewise oscillations are generally minor, so they will not be considered further.

Blades will also be subject to torsional vibrations. In the past, it has generally been possible to ignore these, both because the exciting loads were small and because the high torsional stiffness of a typical hollow blade placed the torsional natural frequencies well above the exciting frequencies. However, with the development of larger, more flexible blades, this is no longer always the case. Blade torsion, which can accompany flapwise bending due to offset of the cross-section centre of mass from the shear centre, for example, may significantly alter the angle of attack in the tip region.

Finally, in the case of a blade hinged at the root, the whole blade will experience oscillations involving rigid body rotation about the hinge. This phenomenon is considered in the section headed ‘Teeter motion’ below.

5.8.2 Mode shapes and frequencies

The mode shape and frequency of the first mode can be derived by an iterative technique called the Stodola method after its originator. Briefly, this consists of assuming a plausible mode shape, calculating the inertia loads associated with it for an arbitrary frequency of one radian per second, and then computing the beam deflected profile resulting from these inertia loads. This profile is then normalised, typically by dividing the deflections by the tip deflection, to obtain the input mode shape for the second iteration. The process is repeated until the mode shape converges. Then, in view of the fact that the inertia loads calculated for the input tip deflection combined with a frequency of 1 radian per second must be the same as those calculated for the output tip deflection combined with the actual frequency, the first mode natural frequency can be obtained from the formula:

$$\omega_1 = \sqrt{\frac{\text{Tip deflection input to last iteration}}{\text{Tip deflection output from last iteration}}}$$

If the mode shapes are orthogonal, advantage can be taken of this property to simplify the derivation of the mode shapes and frequencies of the higher modes, provided this is carried out in ascending order. A trial mode shape is assumed as before, but before using it to calculate the inertia loadings, it is ‘purified’ so that it does not contain any lower mode content. For example, ‘purification’ of a second mode trial mode shape, $\mu_{2T}(r)$, of first mode content is achieved by subtracting

$$\mu_{2C}(r) = \mu_1(r) \frac{\int_0^R \mu_1(r) \mu_{2T}(r) m(r) dr}{\int_0^R \mu_1^2(r) m(r) dr} = \mu_1(r) \frac{\int_0^R \mu_1(r) \mu_{2T}(r) m(r) dr}{m_1} \quad (5.71)$$

from it. The modified second mode trial mode shape, $\mu_{2M}(r) = \mu_{2T}(r) - \mu_{2C}(r)$, then satisfies the orthogonality condition

$$\int_0^R \mu_1(r) \mu_{2M}(r) dr = 0$$

After ‘purification’ of the trial mode shape, the Stodola method can be applied exactly as before. Further ‘purification’ before succeeding iterations should not be necessary if the lower

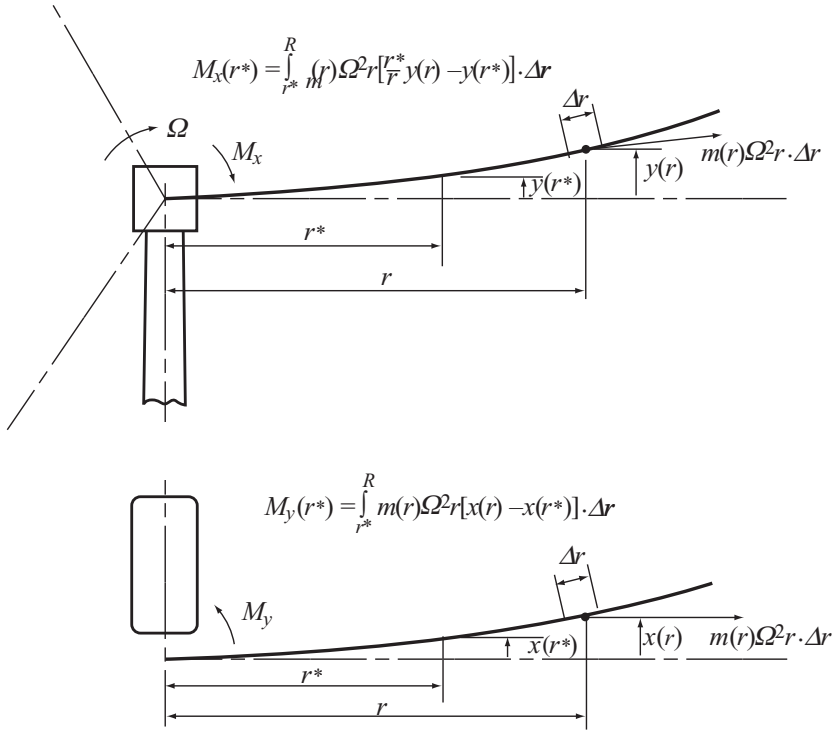


Figure 5.24 Restoring moments due to centrifugal force for in-plane and out-of-plane blade deflections

mode shapes used for the initial ‘purification’ are accurate enough. See Clough and Penzien (1993) for a rigorous treatment of the method.

5.8.3 Centrifugal stiffening

When a rotating blade deflects either in its plane of rotation or perpendicular to it, the centrifugal force on each blade element exerts a restoring force which has the effect of stiffening the blade and thereby increasing the natural frequency compared with the stationary value. The centrifugal forces act radially outwards perpendicular to the axis of rotation, so in the case of an out-of-plane blade deflection, they are parallel to the undeflected blade axis and act at greater lever arms to the inboard part of the blade than they do in the case of in-plane blade deflection. This is illustrated in Figure 5.24.

In order to take account of the effects of centrifugal loads, the equation of motion for a blade element loaded in the out-of-plane direction is modified by the addition of an extra term to become

$$m(r)\ddot{x} + \hat{c}(r)\dot{x} - \frac{\partial}{\partial r} \left[N(r) \frac{\partial x}{\partial r} \right] + \frac{\partial^2}{\partial r^2} \left[EI \frac{\partial^2 x}{\partial r^2} \right] = q(r, t) \quad (5.72)$$

where the centrifugal force at radius r , $N(r)$, is the summation of the forces acting on each blade element outboard of radius r , that is $N(r) = \sum_{r=r}^R m(r)\Omega^2 r \cdot \Delta r$.

The Stodola method for deriving blade mode shapes and frequencies described in the preceding section can be modified to take account of centrifugal effects. In the case of out-of-plane modes, the procedure is as follows:

- 1) Assume plausible trial out-of-plane mode shape, $\mu(r)$.
- 2) 'Purify' trial mode shape of any lower mode content.
- 3) Assume trial value for frequency, ω_j^2 .
- 4) Calculate bending moment distribution due to lateral inertia forces according to:

$$M_{Y.Lat}(r^*) = \int_{r^*}^R m(r)\omega_j^2\mu(r)[r - r^*]dr \quad (5.73)$$

- 5) Calculate bending moment distribution due to centrifugal forces according to:

$$M_{Y.CF}(r^*) = - \int_{r^*}^R m(r)\Omega^2 r[\mu(r) - \mu(r^*)]dr \quad (5.74)$$

- 6) Calculate combined bending moment distribution.
- 7) Calculate new deflected profile resulting from this bending moment distribution.
- 8) Calculate revised estimate of natural frequency from:

$$\omega'_j = \omega_j \sqrt{\frac{\text{Trial tip deflection}}{\text{Tip deflection calculated for new deflected profile}}}$$

- 9) Repeat steps 2–8 with revised mode shape and frequency until calculated mode shape converges.

It is important to note that the lateral loads and deflections of a centrifugally loaded beam do not conform to Bettis's Law, so, as a consequence, the mode shapes are *not* orthogonal. It is for this reason that the 'purification' stage has been included in each cycle of iteration. When convergence of the calculated mode shape has occurred, it will be found that it differs significantly from the 'purified' mode shape input into each iteration, indicating that a true solution has not been obtained. It is then necessary to use a trial and error approach to modify the magnitudes of the 'purifying' corrections applied until the output mode shape and input 'purified' mode shape match. A few further iterations will be required until the natural frequency settles down.

A quick estimate of the first mode frequency of a rotating blade can be derived using the Southwell formula for a uniform rotating beam reported by Putter and Manor (1978) as follows

$$\omega_1 = \sqrt{\omega_{1,0}^2 + \phi_1 \Omega^2} \quad (5.75)$$

in which $\omega_{1,0}$ is the corresponding frequency for the non-rotating blade. The value of ϕ_1 depends on the blade mass and stiffness distribution, and Madsen *et al.* (1984) suggest the value 1.73 for wind turbine blade out-of-plane oscillations. In the case of Blade T40 rotating at 15 rpm, this yields a percentage increase in first mode frequency due to centrifugal stiffening of 7.7% compared to the correct value of 8.1%. Typically, centrifugal stiffening results in an increase of the first mode frequency for out-of-plane oscillations of between 5% and 10%. For higher modes, the magnitude of the centrifugal forces is less in proportion to the lateral inertia forces, so the percentage increase in frequency due to centrifugal stiffening becomes progressively less.

The procedure for deriving the blade first mode shape and frequency in the case of in-plane oscillations is the same as that described above for out-of-plane vibrations, except that the formula for the bending moment distribution due to the centrifugal forces has to be modified to:

$$M_{XCF}(r^*) = \int_{r^*}^R m(r) \Omega^2 r \left[\frac{r^*}{r} \mu(r) - \mu(r^*) \right] dr \quad (5.76)$$

where $\mu(r)$ is now the trial *in-plane* mode shape.

The first mode frequency for in-plane oscillations of Blade T40 in the absence of centrifugal force is 1.56 Hz. This is approximately double the corresponding frequency for out-of-plane oscillations of 0.824 Hz, and so the relative effect of the centrifugal loads is much reduced, even before allowance is made for the smaller lever arms at which they act (Figure 5.24). In fact the increase in the first mode frequency for in-plane oscillations due to centrifugal force is only 0.5% – probably small enough to be ignored.

5.8.4 Aerodynamic and structural damping

Blade motion is generally resisted by two forms of viscous damping, aerodynamic and structural, which are considered in turn.

An approximate expression for the aerodynamic damping per unit length in the flapwise direction can be derived by a method analogous to that used in Section 5.7.5 to derive the linear relation

$$q = \frac{1}{2} \rho \Omega r c(r) \frac{dC_L}{d\alpha} u \quad (5.25)$$

between blade load fluctuations per unit length, q , and fluctuations in the incident wind, u . The wind speed fluctuation, u , is simply replaced by the blade flapwise velocity, $-\dot{x}$, giving

$$\hat{c}_a(r) = \frac{q}{-\dot{x}} = \frac{1}{2} \rho \Omega r c(r) \frac{dC_L}{d\alpha} \quad (5.77)$$

The rate of change of lift coefficient with angle of attack, $dC_L/d\alpha$, is constant and equal to 2π before the blade goes into stall, but can become negative post-stall, leading to the risk of instability – see Section 7.1.9.

It can be seen that the aerodynamic damping per unit length, $\hat{c}_a(r)$, varies spanwise as the product of radius and blade chord, and is therefore not very close to being proportional to the

mass per unit length, as is required to satisfy the orthogonality condition. This will result in some aerodynamic coupling of modes, which is not accounted for in normal modal analysis.

The aerodynamic damping ratio for the i th mode is defined as

$$\xi_{ai} = c_{ai}/2m_i\omega_i = \int_0^R \hat{c}_a(r)\mu_i^2(r)dr/2m_i\omega_i \quad (5.78)$$

Substituting the expression for $\hat{c}_a(r)$ given in Equation 5.77 leads to:

$$\xi_{ai} = \frac{\frac{1}{2}\rho\Omega\frac{dC_L}{d\alpha}\int_0^R rc(r)\mu_i^2(r)dr}{2\omega_i\int_0^R m(r)\mu_i^2(r)dr} \quad (5.79)$$

In the case of fibreglass Blade T40 described in Example 5.1, this yields values of 0.16 and 0.04 for the first and second modes respectively. These high values are a consequence of the lightness of the blade in relation to its width in the vicinity of the tip, an area which dominates the integrals thanks to the mode shape weighting. The corresponding first mode logarithmic decrement ($= 2\pi\xi_a$) is thus 1.0.

At a weight of about 16 tonnes, Blade T40 is very much at the heavy end of the range, with blades for 40 m tip radius of half this weight or less now being typical. The aerodynamic damping ratio is inversely proportional to blade mass, so the first and second mode values would rise to 0.32 and 0.08 respectively for an 8 tonne blade weight.

Structural damping arises from the conversion of mechanical energy to thermal energy during oscillatory motion, as a result of frictional resistance within the flexing material. Denoting the strain energy at peak displacement for the n th cycle as S_n and the loss per cycle as ΔS_n , it can be shown that

$$\frac{\Delta S_n}{S_n} = \frac{2\Delta\sigma_n}{\sigma_n} \quad (5.80)$$

where σ_n is the peak stress for the n th cycle, $\Delta\sigma_n = \sigma_n - \sigma_{n+1}$ and $S_n = \oint (\sigma_n^2/2E)$, with the integral taken over the whole volume of the beam. Hence the logarithmic decrement of damping, defined as the natural logarithm of the ratio of successive peak displacements in the same direction – that is, as $\ln(\sigma_n/\sigma_{n+1})$ for elastic behaviour – approximates to $\Delta\sigma_n/\sigma_n = 0.5(\Delta S_n/S_n)$.

Test results on fibreglass (see, for example, Gibson, 1982) indicate that the energy loss per cycle is unaffected by frequency. However, the percentage energy loss per cycle increases quite rapidly with stress range (Creed, 1993), although it is usual to treat it as independent of stress range for analysis purposes.

Values for the structural damping logarithmic decrement, $\delta_s = 2\pi\xi_s$ at the fundamental natural (i.e. first mode) frequency were given in the 1992 edition of Danish Standard DS 472 ‘Loads and Safety of Wind Turbine Construction’ for several different materials, and these are reproduced in Table 5.5 below, together with corresponding values from EN 1991–1–4 (2005). Equivalent values of the structural damping ratio are also shown. Note that the first mode structural damping ratio for a fibreglass blade is much smaller than the aerodynamic damping ratio for Blade T40 derived above.

Table 5.5 Values of first mode structural damping logarithmic decrements for different materials

Standard	DS 472 (1992)	EN 1991-1-4 (2005)	DS 472	EN 1991-1-4
Material	Logarithmic Decrement, δ_s	Logarithmic Decrement, δ_s	Structural damping ratio, ξ_s	Structural damping ratio, ξ_s
Concrete	0.05	0.03 (towers and chimneys)	0.008	0.005
Steel-welded	0.02	0.012 (chimney)	0.003	0.002
Steel-bolted	0.05	0.03 (high strength bolts)	0.008	0.005
		0.05 (ordinary bolts)		0.008
GRP	0.05	0.04–0.08 (bridges)	0.008	0.006–0.012
Timber	0.05	0.06–0.12 (bridges)	0.008	0.01–0.02

Damping ratios for the first and second flapwise modes of 40 m tip radius blades are presented in Table 5.6.

It is seen that the damping ratio for the second mode is about one third of that for the first.

5.8.5 Response to deterministic loads: step-by-step dynamic analysis

As set out in Section 5.8.1, blade dynamic response to time varying loading is best analysed in terms of the separate excitation of each blade mode of vibration, for which, under the assumptions of unstalled flow and mass-proportional aerodynamic damping, the governing equation is

$$m_i \ddot{f}_i(t) + c_i \dot{f}_i(t) + m_i \omega_i^2 f_i(t) = \int_0^R \mu_i(r) q(r, t) dr \quad (5.70)$$

Table 5.6 Comparison of 40 m radius blade damping ratios for first two flapwise modes

	First mode	Second mode
Natural frequency including centrifugal stiffening	0.89 Hz	2.94 Hz
Structural damping ratio	0.008	0.008
Blade T40 weighing 16 tonne		
Aerodynamic damping ratio	0.16	0.04
Combined damping ratio	0.17	0.05
Lightweight 40 m tip radius		
blade weighing 8 tonne		
Aerodynamic damping ratio	0.32	0.08
Combined damping ratio	0.33	0.09

where $f_i(t)$ and $\mu_i(r)$ are the tip displacement and mode shape for the i th mode respectively. Starting with the initial tip displacement, velocity and acceleration arbitrarily set at zero, this equation can be used to derive values for these quantities at successive time steps over a complete blade revolution by numerical integration. The procedure is then repeated for several more revolutions until the cyclic blade response to the periodic loading becomes sensibly invariant from one revolution to the next.

Linear acceleration method

The precise form of the equations linking the tip displacement, velocity and acceleration at the end of a time step to those at the beginning depends on how the acceleration is assumed to vary over the time step. Newmark has classified alternative assumptions in terms of a parameter β which measures the relative weightings placed on the initial and final accelerations in deriving the final displacement. The simplest assumption is that the acceleration takes a constant value equal to the average of the initial and final values ($\beta = \frac{1}{4}$). Clough and Penzien (1993), however, recommend that the acceleration is assumed to vary linearly between the initial and final values, as this will be a closer approximation to the actual variation. Step-by-step integration with this assumption is known as either the linear acceleration method or the Newmark $\beta = \frac{1}{6}$ method.

Expressions for the tip displacement, velocity and acceleration at the end of the first time step – f_{i1} , $\dot{f}_{i1}$ and $\ddot{f}_{i1}$ respectively – are derived in terms of the initial values – f_{i0} , $\dot{f}_{i0}$ and $\ddot{f}_{i0}$ – as follows. The acceleration at time t during the time step of total duration h is

$$\ddot{f}_i(t) = \ddot{f}_{i0} + \left(\frac{\ddot{f}_{i1} - \ddot{f}_{i0}}{h} \right) t \quad (5.81)$$

This can be integrated to give the velocity at the end of the time step as

$$\dot{f}_{i1} = \dot{f}_{i0} + \ddot{f}_{i0}h + (\ddot{f}_{i1} - \ddot{f}_{i0})h/2 \quad (5.82)$$

Equation 5.81 can be integrated twice to give an expression for the displacement at the end of the time step, which, after rearrangement yields the following expression for the corresponding acceleration:

$$\ddot{f}_{i1} = \frac{6}{h^2} (f_{i1} - f_{i0}) - \frac{6}{h} \dot{f}_{i0} - 2\ddot{f}_{i0} \quad (5.83)$$

Substituting Equation 5.83 into Equation 5.82 yields

$$\dot{f}_{i1} = \frac{3}{h} (f_{i1} - f_{i0}) - 2\dot{f}_{i0} - \ddot{f}_{i0}h/2 \quad (5.84)$$

Equation 5.70 can be written as

$$m_i \ddot{f}_{in} + c_i \dot{f}_{in} + m_i \omega_i^2 f_{in} = \int_0^R \mu_i(r) q_n(r) dr = Q_{in} \quad (5.85)$$

where the suffix n refers to the state at the end of the n th time step. Substituting Equations 5.83 and 5.84 into Equation 5.85 with $n = 1$ and collecting terms yields the displacement at the end of the first time step as follows:

$$f_{i1} = \frac{Q_{i1} + m_i \left(\frac{6}{h^2} f_{i0} + \frac{6}{h} \dot{f}_{i0} + 2\ddot{f}_{i0} \right) + c_i \left(\frac{3}{h} f_{i0} + 2\dot{f}_{i0} + \frac{h}{2} \ddot{f}_{i0} \right)}{m_i \omega_i^2 + \frac{3c_i}{h} + \frac{6m_i}{h^2}} \quad (5.86)$$

The velocity and acceleration at the end of the first time step are then obtained by substituting f_{i1} in Equations 5.84 and 5.83 respectively.

The full procedure for obtaining the blade dynamic response to a periodic loading using the Newmark $\beta = \frac{1}{6}$ method (which is just one of many available) may be summarised as follows:

- 1) Calculate the blade mode shapes, $\mu_i(r)$.
- 2) Select the number of time steps, N , per complete revolution. Then the time step, $h = 2\pi/N\Omega$.
- 3) Calculate the blade element loads, $q(r, \psi_n) = q_n(r)$, at blade azimuth positions corresponding to each time step (i.e. at $2\pi/N$ intervals) using momentum theory. (Here, the suffix n denotes the number of the time step).
- 4) Calculate the generalised load with respect to each mode, $Q_{in} = \int_0^R \mu_i(r) q_n(r) dr$, for each time step.
- 5) Assume initial values of blade tip displacement, velocity and acceleration.
- 6) Calculate first mode blade tip displacement, velocity and acceleration at end of first time step, using Equations 5.86, 5.84 and 5.83 respectively (with $i = 1$).
- 7) Repeat stage 6 for each successive time step over several revolutions until convergence is achieved.
- 8) Calculate cyclic blade moment variation at radii of interest by multiplying the cyclic tip displacement variation by appropriate factors derived from the modal analysis.
- 9) Repeat stages 6–8 for higher modes.
- 10) Combine the responses from different modes to obtain the total response.

Figure 5.25 shows some results of the application of the above procedure to the derivation of the out-of-plane root bending moment response of Blade T40 to tower shadow loading. The case chosen is for a mean wind speed of 12 m/s, uniform across the rotor disc, and an x/D ratio of 1 (where x is the distance between the blade and the tower centreline, and D is the tower diameter), giving a maximum reduction in the blade root bending moment for a rigid blade of 600 KNm (see Figure 5.26). Centrifugal stiffening is included in the derivation of the mode shapes and frequencies. It is evident from Figure 5.25 that the tower shadow gives the blade a sharp ‘kick’ away from the tower, but the duration is too short in relation to the duration of the first mode half cycle for Blade T40 to ‘feel’ the root bending moment reduction that would be experienced by a completely rigid blade. The blade oscillations have largely died away after a complete revolution because of the relatively high levels of damping.

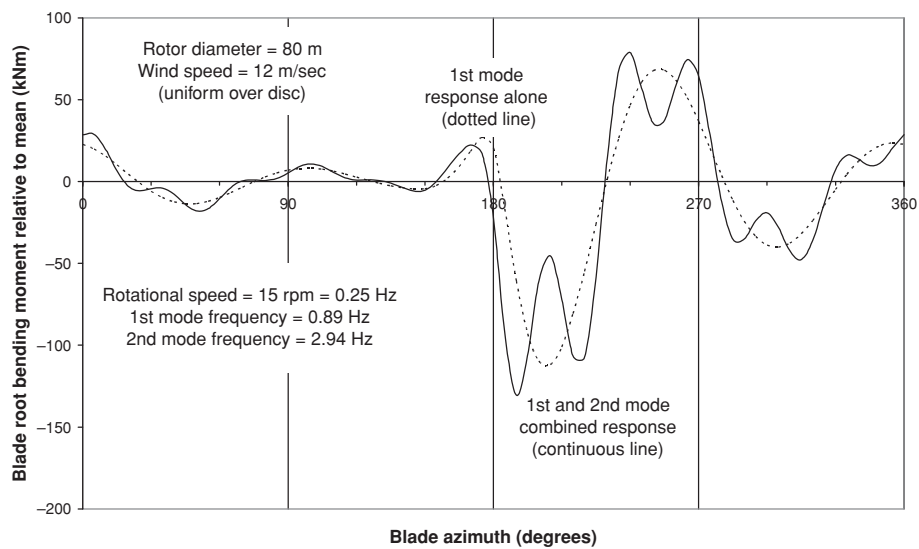


Figure 5.25 Blade T40 out-of-plane root bending moment dynamic response to tower shadow

The response of the Blade T40 out-of-plane root bending moment to tower shadow combined with wind shear is shown in Figure 5.26 for a hub height wind speed of 12 m/s. Also plotted is the corresponding bending moment for a completely rigid blade.

The wind shear loading is approximately sinusoidal (see Figure 5.11) and, consequently, the response is also. However, it is worth noting that the amplitude of the dominating first mode response to wind shear is the result of two effects working against each other – in other

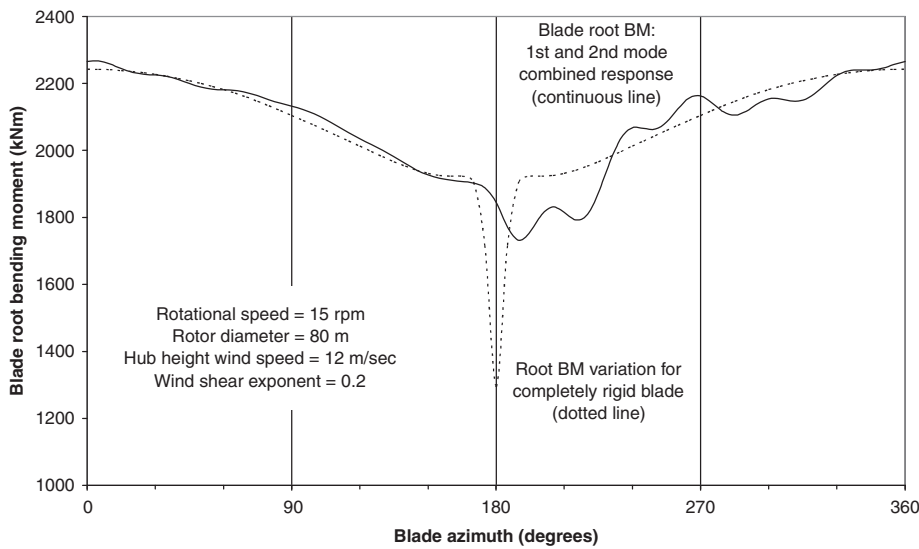


Figure 5.26 Blade T40 out-of-plane root bending moment dynamic response to tower shadow and wind shear

words the increase due to the dynamic magnification factor of about 9% is largely cancelled out by the reduction due to centrifugal stiffening.

Avoidance of resonance: the Campbell diagram

In the course of blade design, it is important to avoid the occurrence of a resonant condition, in which a blade natural frequency equates to the rotational frequency or a harmonic with a significant forcing load. This is often done with the aid of a Campbell diagram, in which the blade natural frequencies are plotted out against rotational frequency together with rays from the origin representing integer multiples of the rotational frequency. Then any intersections of the rays with a blade natural frequency over the turbine rotational speed operating range represent possible resonances. An example of a Campbell diagram is shown in Figure 5.27.

Clearly blade periodic loading is dominated by the loading at rotational frequency from wind shear, yawed flow and shaft tilt (Section 5.7.2), gravity (Section 5.7.3) and gust slicing (Section 5.7.5). However, the short lived load relief resulting from tower shadow will be dominated by higher harmonics.

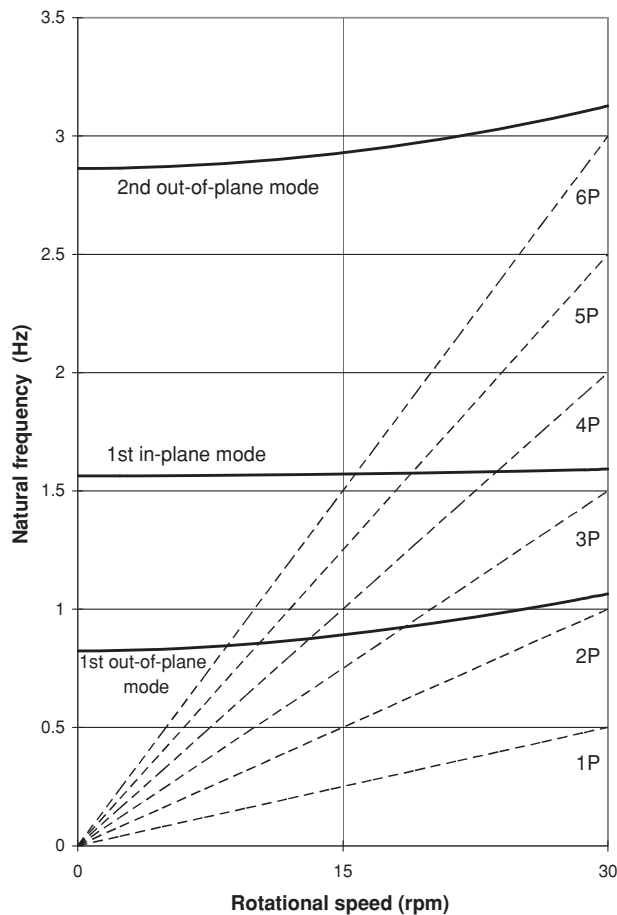


Figure 5.27 Campbell Diagram for Blade T40

5.8.6 Response to stochastic loads

The analysis of stochastic loads in the frequency domain has already been described in Section 5.7.5 for a rigid blade, and this will now be extended to cover the dynamic response of the different vibration modes of a flexible blade using the governing equation, Equation 5.70. Note that the restriction to an unstalled blade operating at a relatively high tip speed ratio still applies.

Power spectrum of generalised blade loading

The generalised fluctuating load with respect to the i th mode is $Q_i = \int_0^R \mu_i(r) q(r) dr$, where

$$\Delta L = L - \bar{L} = u \frac{dL}{du} = \frac{1}{2} \rho (\Omega r)^2 c \frac{dC_L}{d\alpha} \frac{u}{\Omega r} = \frac{1}{2} \rho \Omega r c \frac{dC_L}{d\alpha} u \quad (5.25)$$

Hence

$$Q_i = \frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \int_0^R \mu_i(r) u(r, t) c(r) r dr \quad (5.87)$$

An expression for the standard deviation of Q_i , σ_{Q_i} can be derived by a method analogous to that given in Section A5.4 of Appendix A for a non-rotating blade, yielding

$$\sigma_{Q_i}^2 = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_0^R \int_0^R \left[\int_0^\infty S_u^o(r_1, r_2, n) dn \right] \mu_i(r_1) \mu_i(r_2) c(r_1) c(r_2) r_1 r_2 dr_1 dr_2 \quad (5.88)$$

Here $S_u^o(r_1, r_2, n)$ is the rotationally sampled cross spectrum for a pair of points on the rotating blade at radii r_1 and r_2 . Equation 5.88 is parallel to Equation A5.16 in the Appendix with r and r' replaced by r_1 and r_2 and $(\rho \bar{U} C_F)^2$ replaced by $(\frac{1}{2} \rho \Omega r (dC_L/d\alpha))^2$. From this it can be deduced that the power spectrum of the generalised load with respect to the i th mode is

$$S_{Q_i}(n) = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_0^R \int_0^R S_u^o(r_1, r_2, n) \mu_i(r_1) \mu_i(r_2) c(r_1) c(r_2) r_1 r_2 dr_1 dr_2 \quad (5.89)$$

In practice, this expression is evaluated using summations to approximate to the integrals.

Power spectrum of tip deflection

The expression for the amplitude of the i th mode blade tip response in response to excitation by a harmonically varying generalised load is given by Equation A5.4 in the Appendix. Hence the power spectrum of the tip displacement is related to the power spectrum of the generalised load by

$$S_{\xi_i}(n) = \frac{S_{Q_i}(n)}{k_i^2} \frac{1}{\left[(1 - n^2/n_i^2)^2 + 4\xi_i^2 n^2/n_i^2 \right]} \quad (5.90)$$

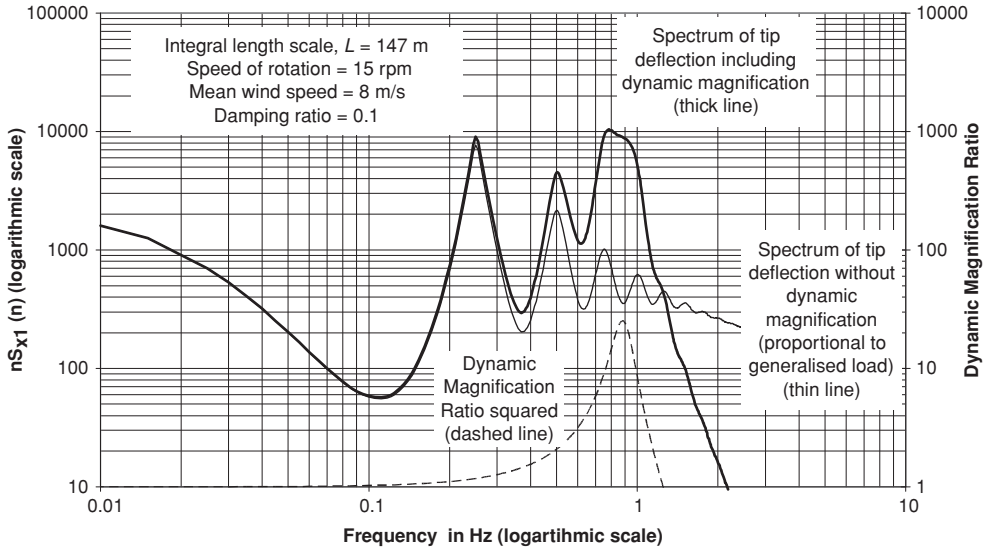


Figure 5.28 Power Spectrum of Blade T40 first out-of-plane mode tip deflection

This can be written $S_{x_i}(n) = (S_{Q_i}(n)/k_i^2)[DMR]^2$ where DMR stands for the dynamic magnification ratio. n_i is the i th mode natural frequency in Hz.

Figure 5.28 shows the power spectrum of first mode tip deflection, $S_{x_1}(n)$, for Blade T40 operating at 15 rpm in a mean wind of 8 m/sec. A lower damping ratio of 0.1 has been selected so that the effect of dynamic magnification is emphasised and the turbulence intensity has been arbitrarily set at 12.5%, so that $\sigma_u = 1$ m/s. Also shown is the first mode tip deflection spectrum ignoring dynamic magnification, $S_{Q_1}(n)/k_i^2$, which, when multiplied by the square of the dynamic magnification ratio (also plotted), yields the $S_{x_1}(n)$ curve. The standard deviation of first mode tip deflection, $\sigma_{x_1} = \int_0^\infty S_{x_1}(n)dn$, comes to 103 mm, a 23% increase compared with the value without dynamic magnification. The former is at a minimum, because the blade T40 first mode natural frequency of 0.89 Hz is approximately midway between the third and fourth harmonics of the rotational frequency. However, it is found that, even if the first mode natural frequency coincided with the third harmonic of the rotational frequency, σ_{x_1} would only increase by 5%. This increase is small, because the peak of $S_{x_1}(n)$ at the third harmonic is not very pronounced, and because the peak in the dynamic magnification ratio is relatively broad.

Power spectrum of blade root bending moment

If the amplitude of tip deflection due to excitation of the blade resonant frequency is defined as $x_R(n_1)$, the amplitude of the corresponding blade root bending moment, $M_Y(n_1)$ is given by

$$M_Y(n_1) = \omega_1^2 x_R(n_1) \int_0^R m(r) \mu_1(r) r dr \quad (5.91a)$$

Noting that $\omega_1^2 = k_1/m_1$, this becomes

$$\frac{M_Y(n_1)}{x_R(n_1)} = k_1 R \frac{\int_0^R m(r) \mu_1(r) (r/R) dr}{m_1} = k_1 R \chi_{M1} \quad (5.91b)$$

This relationship applies at all exciting frequencies, because the right hand side is essentially a function of mode shape. Hence the power spectrum of blade root bending moment due to excitation of the first mode is given by

$$S_{My1}(n) = (k_1 R \chi_{M1})^2 \cdot S_{x1}(n) = (R \chi_{M1})^2 \cdot S_{Q1}(n) \cdot \frac{1}{\left[(1 - n^2/n_1^2)^2 + 4\xi_1^2 n^2/n_1^2 \right]} \quad (5.91c)$$

For Blade T40, the ratio χ_{M1} takes a value of 1.40.

5.8.7 Response to simulated loads

The blade dynamic response to time varying loading derived from wind simulation (Section 5.7.6) can be obtained by a step-by-step dynamic analysis such as that described for use with deterministic loads in Section 5.8.5. The procedure is essentially the same, except that it is more important to select realistic values for the initial blade tip displacement, velocity and acceleration, unless the results from the first few rotation cycles are to be discarded.

5.8.8 Teeter motion

When the rotor is rigidly mounted on the shaft, out-of-plane aerodynamic loads on the blades result in fluctuating bending moments in the low speed shaft additional to those due to gravity. In the case of two-bladed machines, the transfer of blade out-of-plane aerodynamic moments to the shaft can be eliminated and blade root bending moments reduced by mounting the rotor on a hinge with its axis perpendicular to both the low speed shaft and the axis of the rotor. This allows the rotor to teeter to and fro in response to differential aerodynamic loads on each blade.

The restoring moment acting on a rotor rotating at a constant teeter angle is generated by the lateral components of the centrifugal force acting on each blade element (see Figure 5.29). For small teeter angles – up to, say 5 degrees – it may be approximated by

$$M_R = \int_0^R r \cdot m(r) \Omega^2 r \cdot \zeta \cdot dr = I \Omega^2 \zeta \quad (5.92)$$

where ζ is the teeter angle, Ω is the rotational speed and I is the rotor moment of inertia about its centre. The equation of motion for small free teeter oscillations thus becomes, $I\ddot{\zeta} + I\Omega^2\zeta = 0$ (omitting the aerodynamic damping term for the moment), indicating that the natural frequency of the teeter motion with the teeter hinge perpendicular to the rotor axis is equal to the rotational frequency. Since both the deterministic and stochastic components of the exciting moment are dominated by this frequency, it is clear that the system operates at resonance, with aerodynamic damping alone controlling the magnitude of the teeter excursion. In the absence of stochastic wind loads, a teetering rotor can be thought of as rotating in a

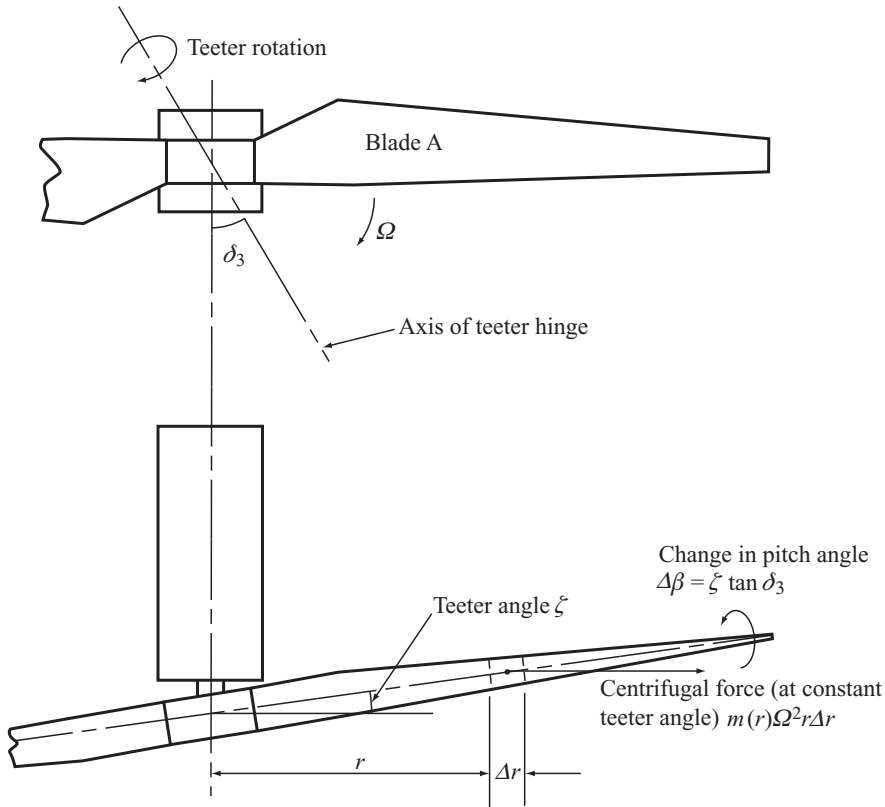


Figure 5.29 Teeter geometry

fixed plane at an angle ζ_0 to the plane perpendicular to the shaft axis (with ζ_0 equal to the maximum teeter angle), because the teetering frequency is equal to the rotational frequency.

The magnitude of teeter excursions would clearly be reduced if the teeter natural frequency were moved away from the rotational frequency. This can be done by rotating the teeter hinge axis relative to the rotor in the plane of rotation, as illustrated in Figure 5.29, so that teeter motion results in a change of blade pitch – positive in one blade and negative in the other – known as Delta 3 coupling. Consider the case of blade A slicing through a gust. The increased thrust on the blade will cause it to move in the downwind direction, by rotating about the teeter hinge. If the teeter angle, defined as the rotation of the blade in its own radial plane, is ζ , then the increase in blade A's pitch angle will be $\zeta \tan \delta_3$, where δ_3 is as defined in Figure 5.29. The increase in the pitch angle of blade A will reduce the angle of attack, α , and thereby reduce the thrust loading on it. The net result of this and a simultaneous increase in the thrust loading on blade B is to introduce an additional restoring moment which will further help to reduce the teeter motion.

The first stage for the exploration of teeter response to different loadings is the derivation of the complete equation of motion. It is assumed that the blades are unstalled and are operating at a relatively high tip speed, so that the linear relations adopted in the derivation of Equation 5.25 in Section 5.7.5 can be retained. The various contributions to the

change in the aerodynamic force on a blade element relative to the steady state situation are therefore:

$$\frac{1}{2}\rho\Omega r c \frac{dC_L}{d\alpha}(u - \dot{\zeta}r) - \frac{1}{2}\rho(\Omega r)^2 c \frac{dC_L}{d\alpha} \cdot \Delta\theta \quad (5.93)$$

where the three terms result from the fluctuation of the incident wind, teeter motion and Delta 3 coupling respectively. Multiplication of these terms by radius, integration over the length of the blade and addition of the centrifugal and inertia hub moment terms yields the following equation of motion for the teeter response:

$$\begin{aligned} I\ddot{\zeta} + \frac{1}{2}\rho\Omega \frac{dC_L}{d\alpha} \left[\int_{-R}^R r^3 c(r) dr \right] \cdot \dot{\zeta} + \frac{1}{2}\rho\Omega \frac{dC_L}{d\alpha} \left[\int_{-R}^R r^3 c(r) dr \right] \Omega \tan \delta_3 \zeta + I\Omega^2 \zeta \\ = \frac{1}{2}\rho\Omega \frac{dC_L}{d\alpha} \int_{-R}^R u(r, t) c(r) \cdot r \cdot |r| dr \end{aligned} \quad (5.94)$$

assuming a frozen wake. By dividing through by the moment of inertia and writing

$$\eta = \frac{1}{2} \frac{\rho}{I} \frac{dC_L}{d\alpha} \left[\int_{-R}^R r^3 c(r) dr \right] \quad (5.95)$$

this can be simplified to

$$\ddot{\zeta} + \eta\Omega\dot{\zeta} + (1 + \eta \tan \delta_3) \Omega^2 \zeta = \frac{1}{2}\rho \frac{\Omega}{I} \frac{dC_L}{d\alpha} \int_{-R}^R u(r, t) c(r) \cdot r \cdot |r| dr \quad (5.96)$$

η is a measure of the ratio of aerodynamic to inertial forces acting on the blade, and is one eighth of the Lock number.

Delta 3 coupling thus raises the natural frequency, ω_n , of the teeter motion from Ω to $\Omega\sqrt{1 + \eta \tan \delta_3}$. For an 80 m diameter rotor consisting of two T40 blades mounted on a teeter hinge set at a δ_3 angle of 30° , $\eta = 0.89$ and $\tan \delta_3 = 0.577$, so the increase in natural frequency due to the δ_3 angle is 23%. The corresponding damping ratio, given by $\xi = \eta/(2\sqrt{1 + \eta \tan \delta_3})$ is quite high at 0.36.

Teeter response to deterministic loads

The teeter response to deterministic loads can be found using the same step-by-step integration procedure set out in Section 5.8.5. However, as the loadings due to wind shear and yaw are both approximately sinusoidal, an estimate of the maximum teeter angle for these cases may be obtained by using the standard solution for forced oscillations. For a harmonically varying teeter moment, $M_T = M_{To} \cos \Omega t$ due to wind shear, the teeter angle is given by

$$\zeta = \frac{M_{To}}{I\omega_n^2} \frac{\cos(\Omega t - \vartheta)}{\sqrt{(1 - (\Omega/\omega_n)^2)^2 + (2\xi\Omega/\omega_n)^2}} \quad (5.97)$$

where $\vartheta = \tan^{-1}((2\xi\Omega/\omega_n)/(1 - (\Omega/\omega_n)^2)) = 90^\circ - \delta_3$ is the phase lag with respect to the excitation.

For the two-bladed turbine described above, rotating at 15 rpm in a wind with a hub-height mean of 10 m/s and a shear exponent of 0.2, the teeter moment amplitude, M_o , is approximately 360 KNm (see Figure 5.11, which gives the blade root bending moment variation with azimuth for a fixed hub machine, based on momentum theory). Taking the rotor moment of inertia as 9,822,000 kg m² for T40 blades, and $\omega_n = 1.23\pi/2$ rad/sec, the maximum teeter angle comes to 0.83° for $\delta_3 = 30^\circ$. This increases by about 15% to 0.96° if the δ_3 angle is reduced to zero.

If the wind speed variation due to wind shear experienced by a vertical blade is assumed to be linear with height, that is, $u = \bar{U}(kr/R)$, and the teeter moment is calculated from the expression on the right hand side of Equation 5.94, which assumes a frozen wake instead of the equilibrium wake resulting from momentum theory, a very simple expression for the teeter angle results in the case of zero δ_3 angle. The teeter moment becomes

$$M_T = M_{TO} \cdot \cos \Omega t = \frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \frac{\bar{U}k}{R} \int_{-R}^R c(r) r^3 dr \cdot \cos \Omega t \quad (5.98)$$

Substitution of Equation 5.98 in Equation 5.97, with ω_n set equal to Ω for the case of a zero δ_3 angle, results in the following expression for the teeter angle

$$\zeta = \frac{\bar{U}k}{\Omega R} \cos(\Omega t - \pi/2) \quad (5.99)$$

Thus, the magnitude of the teeter excursion is simply equal to the velocity gradient divided by the rotational speed. For a hub height of 70 m, the equivalent uniform velocity gradient over the rotor disc for the case above is $0.125\bar{U}/R = 0.03125$ m/sec per m, giving a teeter excursion of 0.020 radians or 1.14°. This differs from the earlier value of 0.96° because of the frozen wake assumption.

Teeter response to stochastic loads

As usual, it is convenient to analyse the response to the stochastic loads in the frequency domain. The teeter moment providing excitation is given by the right hand side of Equation 5.94. By following a similar method to that used for the generalised load in Section 5.8.6, the following expression for the power spectrum of the teeter moment can be derived:

$$S_{MT}(n) = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_{-R}^R \int_{-R}^R S_u^o(r_1, r_2, n) c(r_1) c(r_2) \cdot r_1 r_2 |r_1| |r_2| dr_1 dr_2 \quad (5.100)$$

where $S_u^o(r_1, r_2, n)$ is the rotationally sampled cross spectrum. In practice, $S_u^o(r_1, r_2, n)$ is evaluated for a few discrete radius values, and the integrals replaced by summations.

The power spectrum of the teeter angle response is related to the teeter moment power spectrum by a formula analogous to Equation 5.90, as follows:

$$S_\zeta(n) = \frac{S_{MT}(n)}{(I\omega_n^2)^2} \frac{1}{\left[(1 - (2\pi n/\omega_n)^2)^2 + (2\xi \cdot 2\pi n/\omega_n)^2 \right]} \quad (5.101)$$

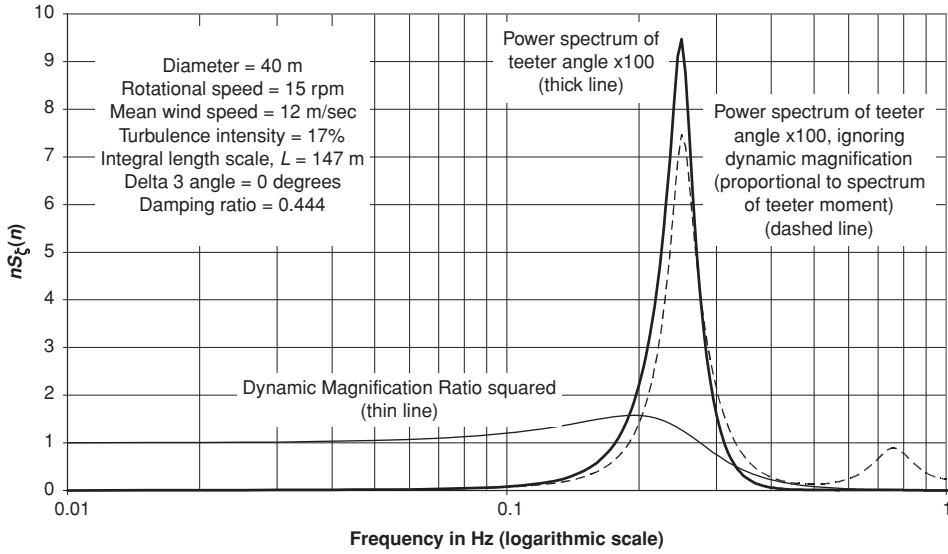


Figure 5.30 Teeter angle power spectrum for two-bladed rotor with T40 blades

This can be written $S_{\xi}(n) = (S_{MT}(n)/(I\omega_n^2)^2) [DMR]^2$ where *DMR* stands for the dynamic magnification ratio.

Figure 5.30 shows the teeter angle power spectrum, $S_{\xi}(n)$, for a two-bladed rotor with T40 blades and zero δ_3 angle operating at 15 rpm in a mean wind of 12 m/sec. The turbulence intensity is taken as 17%, for a Class B site, and the damping ratio, $\xi = \eta/2$, is 0.444, calculated from Equation 5.95. Also shown in the figure is the teeter angle power spectrum ignoring dynamic magnification, $S_{MT}(n)/(I\omega_n^2)^2$, which, when multiplied by the square of the dynamic magnification ratio (also plotted), yields the $S_{\xi}(n)$ curve. The resulting teeter angle standard deviation, obtained by taking the square root of the area under the power spectrum, is 0.93° .

Having calculated the teeter angle standard deviation, the extreme value over any desired exposure period can be predicted from Equation 5.59. As is evident from Figure 5.30, the teeter angle power spectrum is all concentrated about the rotational frequency, Ω , so the zero upcrossing frequency, ν , can be set equal to it. Thus, for a machine operating at 15 rpm, a one hour exposure period gives, $\nu T = 900$ and $\zeta_{\max}/\sigma_{\xi} = 3.84$. The predicted maximum teeter angle due to stochastic loading over a one hour period for the case above is, thus, $3.84 \times 0.93^\circ = 3.6^\circ$. This reduces to 3.0° if a δ_3 angle of 30° is introduced.

As already mentioned, teetering relieves blade root bending moments as well as those in the low speed shaft. The reduction of the stochastic component of root bending moment can be derived in terms of the standard deviations of blade root bending moment and hub teeter moment for a rigid hub two blade machine. Integration of Equation 5.100 yields the following expression for the latter:

$$\sigma_{MT}^2 = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_{-R}^R \int_{-R}^R \kappa_u^o(r_1, r_2, 0) c(r_1) c(r_2) \cdot r_1 r_2 |r_1| |r_2| dr_1 dr_2 \quad (5.102)$$

where, for convenience of notation, r_1 and r_2 , take *negative* values on the second blade. $\kappa_u^o(r_1, r_2, 0)$ is the cross correlation function between the longitudinal wind fluctuations between points at radii r_1 and r_2 on the rotating rotor and is given by the right hand side of Equation 5.51, with $\Omega\tau$ set equal to zero when r_1 and r_2 define points on the same blade, and replaced by π when r_1 and r_2 define points on different blades. Defining $\rho_u^o(r_1, r_2, 0)$ as the normalised cross correlation function, $\kappa_u^o(r_1, r_2, 0)/\sigma_u^2$, Equation 5.102a can be rewritten as:

$$\sigma_{MT}^2 = \sigma_u^2 \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_{-R}^R \int_{-R}^R \rho_u^o(r_1, r_2, 0) c(r_1) c(r_2) \cdot r_1 r_2 |r_1| |r_2| dr_1 dr_2 \quad (5.102a)$$

The corresponding expression for the standard deviation of the mean of the two blade root bending moments is:

$$\sigma_M^2 = \frac{1}{4} \sigma_u^2 \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_{-R}^R \int_{-R}^R \rho_u^o(r_1, r_2, 0) c(r_1) c(r_2) \cdot r_1^2 r_2^2 dr_1 dr_2 \quad (5.103)$$

By inspection of the integrals, it is easily shown that:

$$\frac{1}{4} \sigma_{MT}^2 + \sigma_M^2 = \sigma_u^2 \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_0^R \int_0^R \rho_u^o(r_1, r_2, 0) c(r_1) c(r_2) \cdot r_1^2 r_2^2 dr_1 dr_2 = \sigma_M^2 \quad (5.104)$$

where σ_M is the standard deviation of root bending moment for a rigidly mounted blade. Thus, if the rotor is allowed to teeter, the standard deviation of the blade root bending moment will drop from σ_M to $\sigma_{\overline{M}}$ where σ_M is given by the equation above. The extent of the reduction is driven primarily by the ratio of rotor diameter to the integral length scale of the wind turbulence. For a two-bladed rotor with T40 blades and an integral length scale of 147 m, the reduction is 11%.

5.8.9 Tower coupling

In the preceding sections, consideration of the dynamic behaviour of the blade has been based on the assumption that the nacelle is fixed in space – that is, that the tower is rigid. In practice, of course, no tower is completely rigid, so fluctuating loads on the rotor will result in fore-aft flexure of the tower, which, in turn, will affect the blade dynamics. This section explores the effect the coupling of the blade and tower motions has on blade response.

The application of standard modal analysis techniques to the dynamic behaviour of the system comprising the tower and rotating rotor treated as a single entity is complicated by the system's continually changing geometry, which means that the mode shapes and frequencies of the structure taken as a whole would have to be re-evaluated at each succeeding rotor azimuth position.

An alternative approach is to base the analysis on the mode shapes and frequencies of the different elements of the structure considered separately, with the displacements arising from each set of modes superposed. Thus, the tower modes are calculated on the basis of a completely rigid rotor, and the blade modes are calculated as if the blades were cantilevered from a rigidly mounted shaft – that is, in the same way as before. The blade modes are

not orthogonal to the tower modes, so the equations of motion for the different modes are no longer independent of each other, but contain coupled terms. Furthermore, the blade deflections arising from excitation of the tower modes vary with blade azimuth, so a step-by-step solution is required. The treatment which follows is limited to the fundamental blade and tower modes, but could be extended to encompass higher modes.

The equation of motion of the blade is given by Equation 5.62. The blade deflection for blade J may be written

$$x(r, t) = \mu(r) \cdot f_J(t) + \mu_{TJ}(r) \cdot f_T(t) \quad (5.105)$$

where $\mu(r)$ is the first blade mode shape for a rigid tower and, $\mu_{TJ}(r)$ is the normalised rigid body deflection of blade J resulting from excitation of the tower first mode. Assuming the normalisation is carried out with respect to hub deflection,

$$\mu_{TJ}(r) = 1 + \frac{r}{L} \cos \psi_J \quad (5.106)$$

where L is the depth below the hub of the intercept between the tangent to the top of the deflected tower and the undeflected tower axis, as illustrated in Figure 5.31.

Substitution of Equation 5.105 into Equation 5.62 yields, with the aid of Equation 5.65:

$$\begin{aligned} m(r)\mu(r)\ddot{f}_J(t) + \hat{c}(r)\mu(r)\dot{f}_J(t) + m(r)\omega^2\mu(r)f_J(t) &= q(r, t) - m(r)\mu_{TJ}(r)\ddot{f}_T(t) \\ &\quad - \hat{c}(r)\mu_{TJ}(r)\dot{f}_T(t) \end{aligned} \quad (5.107)$$

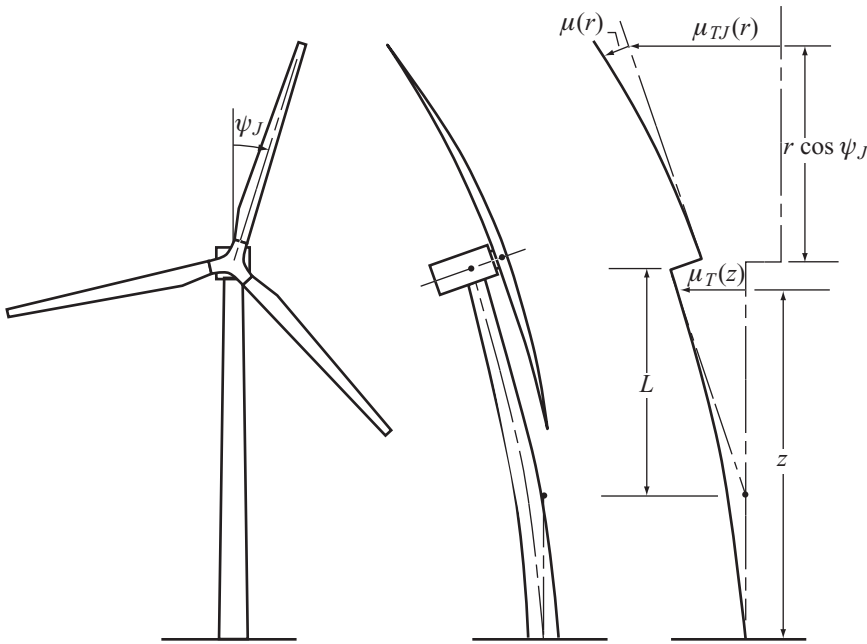


Figure 5.31 Fundamental mode shapes of blade and tower

where the coupled terms have been transferred to the right hand side. Multiplying through by $\mu(r)$ and integrating over the length of the blade gives:

$$\begin{aligned} m_1 \ddot{f}_J(t) + c_1 \dot{f}_J(t) + m_1 \omega^2 f_J(t) = & \int_0^R \mu(r) q(r, t) dr - \int_0^R m(r) \mu(r) \mu_{TJ}(r) dr \cdot \ddot{f}_T(t) \\ & - \int_0^R \hat{c}(r) \mu(r) \mu_{TJ}(r) dr \cdot \dot{f}_T(t) \end{aligned} \quad (5.108)$$

By analogy with Equation (5.70), the equation of motion of the tower is

$$m_{T1} \ddot{f}_T(t) + c_{T1} \dot{f}_T(t) + m_{T1} \omega_T^2 f_T(t) = \int_0^H \mu_T(z) q(z, t) dz \quad (5.109)$$

Here μ_T is the tower first mode shape and m_{T1} is the generalised mass of the tower, nacelle and rotor (including the contribution of rotor inertia), with respect to the first mode, given by

$$m_{T1} = \int_0^H m_T(z) \mu_T^2(z) dz + m_N + m_R + I_R/L^2 \quad (5.110)$$

Here $m_T(z)$ is the mass per unit height of the tower, m_N and m_R are the nacelle and rotor masses, and I_R is the inertia of the rotor about the horizontal axis in its plane, which is constant over time for a three-bladed rotor. For a two-bladed, fixed hub rotor it varies with rotor azimuth, and for a teetering rotor it is omitted altogether.

The major component of the loading on the tower, $q(z, t)$, is the load fed in at hub height, H , from the blades. The inertia forces on the blades due to rigid body motion associated with the tower first mode have been accounted for by including rotor mass and inertia in m_{T1} , and the corresponding damping forces can be accounted for in the calculation of the damping coefficient, c_{T1} . However, the aerodynamic loads on the blades and the inertia and damping forces associated with blade flexure – all of which are transmitted to the tower top – have to be included in the right hand side of Equation 5.109 as

$$\mu_T(H) \cdot F + \left(\frac{d\mu_T}{dz} \right)_H \cdot M = F + M/L \quad (5.111)$$

where

$$F = \sum_N \int_0^R q_J(r, t) dr - \sum_N \int_0^R m_1(r) \mu(r) dr \cdot \ddot{f}_J(t) - \sum_N \int_0^R \hat{c}(r) \mu(r) dr \cdot \dot{f}_J(t) \quad (5.112)$$

and

$$\begin{aligned} M = & \sum_N \int_0^R r \cos \psi_J q_J(r, t) dr - \sum_N \int_0^R r \cos \psi_J m_1(r) \mu(r) dr \cdot \ddot{f}_J(t) \\ & - \sum_N \int_0^R r \cos \psi_J \hat{c}(r) \mu(r) dr \cdot \dot{f}_J(t) \end{aligned} \quad (5.113)$$

The suffix J refers to the J th blade, and N in the summations is the total number of blades.

Hence

$$\begin{aligned} F + M/L = & \sum_N \int_0^R \mu_{TJ} q_J(r, t) dr - \sum_N \int_0^R m_1(r) \mu(r) \mu_{TJ}(r) dr \cdot \ddot{f}_J(t) \\ & - \sum_N \int_0^R \hat{c}(r) \mu(r) \mu_{TJ}(r) dr \cdot \dot{f}_J(t) \end{aligned}$$

and Equation 5.109 becomes

$$\begin{aligned} m_{T1} \ddot{f}_T(t) + c_{T1} \dot{f}_T(t) + m_{T1} \omega_T^2 f_T(t) = & \sum_N \int_0^R \mu_{TJ} q_J(r, t) dr \\ & - \sum_N \int_0^R m_1(r) \mu(r) \mu_{TJ}(r) dr \cdot \ddot{f}_J(t) - \sum_N \int_0^R \hat{c}(r) \mu(r) \mu_{TJ}(r) dr \cdot \dot{f}_J(t) \quad (5.114) \end{aligned}$$

omitting the term for loading on the tower itself.

Equations 5.108 and 5.114 provide $(N + 1)$ simultaneous equations of motion with periodic coefficients μ_{TJ} corresponding to the $(N + 1)$ degrees of freedom assumed. The procedure for the step-by-step dynamic analysis which is based on these equations may be summarised as follows:

1. Substitute the displacements, velocities and aerodynamic loads at the beginning of the first time step into Equations 5.108 and 5.114, and solve for the initial accelerations.
2. Formulate the *incremental* equations of motion for the time step, based on Equations 5.108 and 5.114, retaining the coupled terms on the right hand side – i.e. as pseudo forces.
3. Assume initially that the coupled terms are constant over the duration of the time step, so that they disappear from the incremental equations of motion altogether, rendering them uncoupled.
4. Solve the uncoupled incremental equations of motion to obtain the increments of displacement and velocity over the time step. Adopting the linear acceleration method (Section 5.8.5), the expressions for the displacement and velocity increments at the tip of blade J are as follows:

$$\Delta f_J = \frac{\Delta Q_J + m_1 \left(\frac{6}{h} \dot{f}_{J0} + 3 \ddot{f}_{J0} \right) + c_1 \left(3 \dot{f}_{J0} + \frac{h}{2} \ddot{f}_{J0} \right)}{m_1 \omega^2 + \frac{3c_1}{h} + \frac{6m_1}{h^2}} \quad (5.115)$$

$$\Delta \dot{f}_J = \frac{3}{h} \Delta f_J - 3 \dot{f}_{J0} - \frac{h}{2} \ddot{f}_{J0} \quad (5.116)$$

The derivation of these expressions parallels that for the absolute values of displacement and velocity at the end of the time step, given in Section 5.8.5. Similar expressions obtain for the displacement and velocity increments at the hub due to tower flexure.

5. Solve Equations 5.108 and 5.114 for the accelerations at the *end* of the time step.
6. Solve the incremental equations of motion again – this time including the changes in the coupled terms on the right hand side over the time step – to obtain revised increments of displacement and velocity over the time step.
7. Repeat step 5 and step 6 until the increments of displacement and velocity converge.
8. Repeat steps 1–7 for the second and subsequent time steps.

If the analysis is being carried out to obtain the response to deterministic loads, advantage may be taken of the fact that the behaviour of each blade mirrors that of its neighbours with an appropriate phase difference. This means that the number of equations of motion can be reduced to two, and the analysis iterated over a number of revolutions until a steady state response is achieved. For example, in the case of a machine with three blades, A, B and C, the values of blade B and blade C tip velocities and accelerations, which are required on the right hand side of Equation 5.114, would be equated to the corresponding values for blade A occurring $T/3$ and $2T/3$ earlier (T being the period of blade rotation).

Figure 5.32 shows the results from the application of the above procedure to the derivation of blade tip and hub displacements in response to tower shadow loading, considering only the blade and tower fundamental modes. The machine is three-bladed and the parameters chosen are, as far as the rotor is concerned, generally the same as for the rigid tower example in Section 5.8.5 illustrated in Figure 5.25. The tower natural frequency is 0.58 Hz, and the

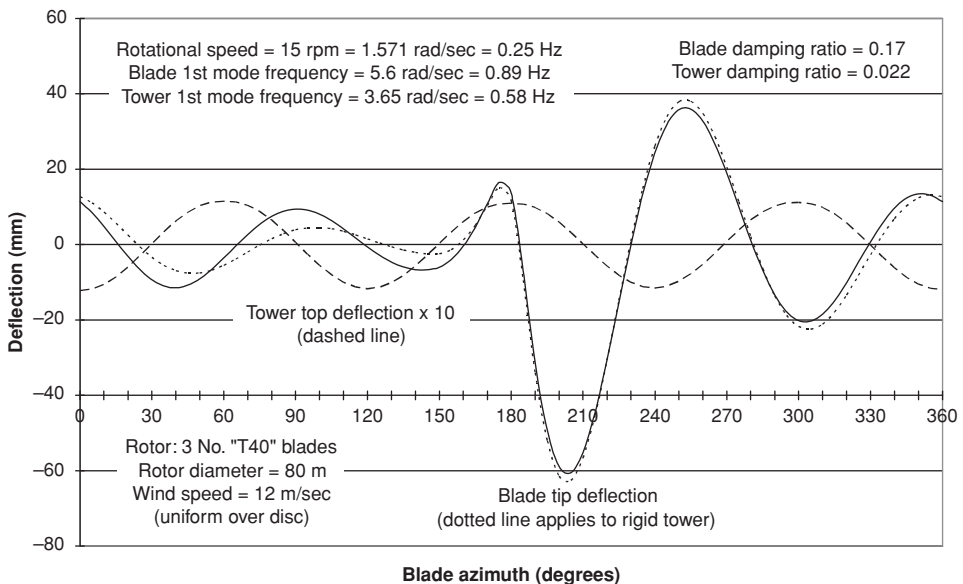


Figure 5.32 Tower top and blade tip deflections resulting from tower shadow, considering fundamental mode responses only

tower damping ratio (which is dominated by the aerodynamic damping of the blades) is taken as 0.022.

It can be seen that the tower response is sinusoidal at blade passing frequency, which is the forcing frequency. The amplitude is only about one fiftieth of the maximum blade tip displacement of about 60 mm, reflecting the large generalised mass associated with the tower mode relative to that associated with the blade mode. The tower shadow effect causes the blade to accelerate rapidly upwind as it passes the tower, with the maximum deflection occurring at an azimuth of about 205° . Also plotted on Figure 5.32 is the deflection that would occur if the nacelle were fixed, and it is seen from the comparison that one effect of tower flexibility is to slightly reduce the peak deflection. However, a more significant effect of the tower motion is the maintenance of the amplitude of the subsequent blade oscillations at a higher level prior to the next tower passing.

The modal analysis method outlined above forms the basis for a number of codes for wind turbine dynamic analysis, such as the Garrad Hassan BLADED code (Garrad Hassan, 2010). Typically these codes allow at least the first few blade modes including flapwise, edgewise and torsional degrees of freedom and several tower modes (fore-aft, side-to-side and torsional) to be represented together with drive train dynamics. See Section 5.13.

Rather than use modal analysis, the dynamic behaviour of coupled rotor/tower systems can also be investigated using finite elements. Standard finite element dynamics packages are, however, inappropriate to the task, because they are only designed to model the displacements of structures with fixed geometry. Lobitz (1984) has pioneered the application of the finite element method to the dynamic analysis of wind turbines with two-bladed, teetering rotors, and Garrad (1987) has extended it to three-bladed, fixed hub machines. In both cases, equations of motion are developed in matrix form for the blade and tower displacement vectors and then amalgamated using a connecting matrix which is a function of blade azimuth and satisfies the compatibility and equilibrium requirements at the tower/rotor interface. Solution of the equations is carried out by a step-by-step procedure. The finite element method is more demanding of computing power, so the modal analysis method is generally preferred.

5.8.10 Aeroelastic stability

Aeroelastic instability can arise when the change in aerodynamic loads resulting from a blade displacement is such as to exacerbate the displacement rather than diminish it, as is normally the case. A theoretical example would be a teetering rotor operating in stalled flow, where the rate of change of lift coefficient with angle of attack is negative, so that the aerodynamic damping is negative likewise. In such circumstances, teeter excursions would be expected to grow until the limits of the negative damping band or of the teeter stops were reached. In practice, this phenomenon can be avoided if the blade is designed so that the blade root flapwise bending moment increases monotonically with wind speed over the full wind speed operational range (Armstrong and Hancock, 1991).

A real instance of incipient aeroelastic instability was the development of an edgewise blade resonance under stalled conditions on some larger three-bladed machines. A negative rate of change of lift coefficient with angle of attack is believed to have been the prime cause – see Section 7.1.9.

Another potential instance of aeroelastic instability is classical flutter, encountered in the design of helicopter rotors, in which the blade structure is such that out-of-plane flexure in the downwind direction results in blade twisting, causing an increase in the angle of attack.

During the development of some of the early large machines, the dangers of aeroelastic instability were considered to be a real concern, and much analysis work was directed to demonstrating that individual turbine designs would not be susceptible to it. However, partly no doubt because of the high torsional rigidity of the closed cell hollow structure adopted for most wind turbine blades, aeroelastic instability was not found to be critical in practice.

With the recent development of some very flexible blade designs for large machines, stability analyses are once again becoming important in the design process.

5.9 Blade fatigue stresses

5.9.1 Methodology for blade fatigue design

The verification of the adequacy of a blade design in fatigue requires knowledge of the fatigue loading cycles expected over the lifetime of the machine at different radii, derivation of the resultant stress cycles and calculation of the corresponding fatigue damage number in relation to known fatigue properties of the material. The procedure is less or more complicated, depending upon whether blade loading in one or two planes is taken into account. If bending about only the weaker principal axis is taken into account, considering only aerodynamic lift forces, the steps involved are as follows:

1. Derive the individual fatigue load spectra for each mean wind speed and for each radius. This is a non-trivial task, because, unless wind simulation is used, the information on the periodic and stochastic load components is available in different forms, that is, as a time history and a power spectrum respectively. Sections 5.9.2 and 5.9.3 consider methods of addressing this difficulty.
2. Synthesise the complete fatigue load spectrum at each radius from the separate load spectra for each mean wind speed, including start-ups and shutdowns (see Section 5.5.1).
3. Convert the fatigue load cycles (expressed as bending moments) to fatigue stresses by dividing by the appropriate section modulus. (The section modulus with respect to a particular principal axis is defined as Second Moment of Area of the cross-section about that axis divided by the distance of the point under consideration from the axis).
4. Sum the fatigue damage numbers, n_i/N_i , according to Miner's rule, for each moment range 'bin' in the fatigue load spectrum, according to the appropriate $S-N$ curve for the material. $S-N$ curves for different blade materials are considered in Sections 7.1.6 and 7.1.7, together with the allowance necessary for mean stress.

Sections 5.9.2 and 5.9.3 are concerned with the first step of the sequence above. For a given mean wind speed, the periodic component of blade loading will be invariant over time, and the stochastic component will be stationary. As noted in Section 5.7.5, the stochastic component can be analysed either in the frequency domain (provided that a linear relationship between incident wind speed and blade loadings can be assumed) or in the time domain – that is, by using wind simulation. Section 5.9.2 considers how the deterministic and stochastic components may be combined if the latter have been analysed in the frequency domain, while Section 5.9.3 looks in detail at the option of assessing fatigue damage completely in the frequency domain.

If the fatigue damage resulting from both in-plane and out-of-plane loading is to be computed, it is necessary to revise the ordering of the steps above, in order to derive the periodic and stochastic components of the stress variation for each point under consideration and for each mean wind speed. For a chosen point, the procedure becomes:

- A1. For a given mean wind speed, calculate the time histories of the bending moments about the principal axes resulting from the periodic load components over one blade rotation. The derivation of aerodynamic moments from blade element loads is illustrated in Figure 5.33.

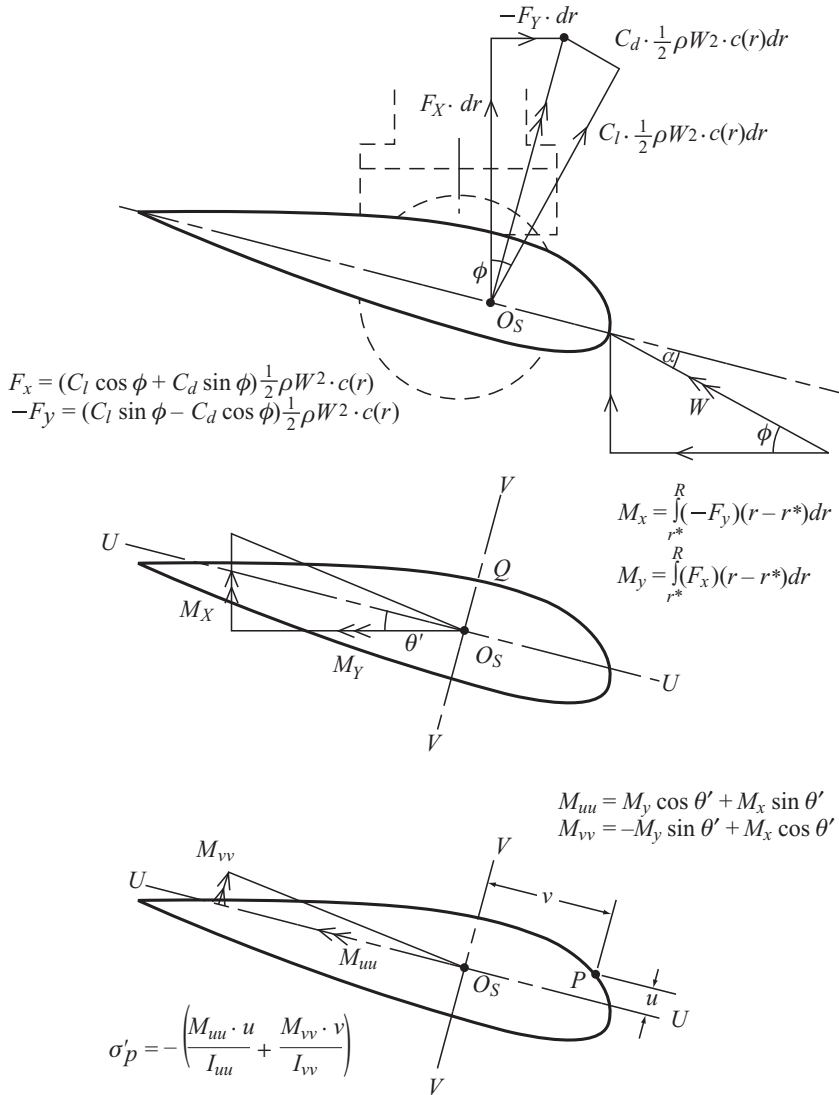


Figure 5.33 Derivation of blade bending stresses at radius r^* due to aerodynamic loads

- A2. Convert these bending moment time histories to stress time histories by dividing by the appropriate section modulus, and adding them together.
- B. For the same mean wind speed, convert the power spectrum of the stochastic bending moment component (which, because of the linearity assumption, arises from fluctuating lift only) to a power spectrum of stress at the chosen point.
- C. Calculate the fatigue damage resulting from the combined periodic and stochastic stress components, using the methods of Sections 5.9.2 and 5.9.3.
- D. Repeat the above steps for the other mean wind speeds.
- E. Add together the fatigue damages arising at each mean wind speed to obtain the total fatigue damage during normal running.

5.9.2 Combination of deterministic and stochastic components

Previous sections have shown how the deterministic (i.e. periodic) and stochastic components of blade bending moments can be characterised in terms of time histories and power spectra respectively. Unfortunately the spectral description of the stochastic loading is not in a suitable form to be combined with the time history of the periodic loading, but this difficulty can be resolved by one of two methods, as follows:

1. The power spectrum of the stochastic component can be transformed into a time history by inverse Fourier transform, which can then be added directly to the time history of the periodic component. Applications of this method have been reported by Garrad and Hassan (1986) and Warren *et al.* (1988). With the subsequent development of wind simulation techniques, this method is no longer commonly used, because the use of transformations to generate time-histories of *wind speed* rather than of wind loading avoids the need to assume that wind speed and wind loading are linearly related when deriving the power spectrum of the stochastic load component.
2. A probability density function for the load cycle ranges can be derived empirically, based on the spectral properties of the power spectrum of the stochastic and periodic components of loading combined.

The second approach is considered in the next section.

5.9.3 Fatigue prediction in the frequency domain

The probability density function (pdf) of peaks of a narrow band, Gaussian process are given by the well-known Rayleigh distribution. As each peak is associated with a trough of similar magnitude, the pdf of cycle ranges is Rayleigh likewise.

Wind turbine blade loading cannot be considered as narrow band, despite the concentration of energy at the rotational frequency by ‘gust slicing’ (Section 5.7.5), and neither can it be considered as Gaussian because of the presence of periodic components. Dirlik (1985) produced an empirical pdf of cycle ranges applicable to both wide and narrow band Gaussian processes, in terms of basic spectral properties determined from the power spectrum. This was done by considering 70 power spectra of various shapes, computing their rainflow cycle

range distributions and fitting a general expression for the cycle range pdf in terms of the first, second and fourth spectral moments. Dirlik's expression for the cycle range pdf is:

$$p(S) = \frac{\frac{D_1}{Q} e^{-Z/Q} + \frac{D_2 Z}{R^2} e^{-(Z^2/2R^2)} + D_3 Z e^{-(Z^2/2)}}{2\sqrt{m_0}} \quad (5.117)$$

where

$$\begin{aligned} Z &= \frac{S}{2\sqrt{m_0}}, \quad D_1 = \frac{2(x_m - \gamma^2)}{1 + \gamma^2}, \quad D_2 = \frac{(1 - \gamma - D_1 + D_1^2)}{1 - R}, \quad D_3 = 1 - D_1 - D_2 \\ Q &= \frac{1.25(\gamma - D_3 - D_2 R)}{D_1}, \quad R = \frac{\gamma - x_m - D_1^2}{(1 - \gamma - D_1 + D_1^2)}, \quad x_m = \frac{m_1}{m_0} \sqrt{\frac{m_2}{m_4}}, \quad \gamma = \frac{m_2}{\sqrt{m_0 m_4}} \\ m_i &= \int_0^\infty n^i S_\sigma(n) dn \end{aligned}$$

$S_\sigma(n)$ is the power spectrum of stress and S is the cycle stress range.

Although the Dirlik cycle range pdf was not intended to apply to signals containing periodic components, several investigations (Hoskin *et al.*, 1989; Morgan and Tindal; 1990; Bishop *et al.*, 1991) have been carried out to determine its validity for wind turbine fatigue damage calculations, using monitored data for flapwise bending from the MS1 wind turbine on Orkney. Cycle range pdfs were calculated from power spectra of monitored strains using the Dirlik formula and fatigue damage rates derived from these pdfs compared with damage rates derived directly from the monitored signal by rainflow cycle counting. The ratio of damage calculated by the Dirlik method to damage calculated by the rainflow method ranged from 0.84 to 1.46, from 1.01 to 2.48 and from 0.73 to 2.34 in the three investigations listed above, using an S/N curve exponent of 5 in each case, as the blade structure was of steel. In view of the fact that the calculated damage rates vary as the fifth power of the stress ranges, these results indicate that the Dirlik method is capable of giving quite accurate results, despite the presence of the periodic components.

There are two main drawbacks to the application of the Dirlik formula to power spectra containing periodic components. Firstly the presence of large spikes in the spectra due to the periodic components renders them very different from the smooth distributions Dirlik originally considered, and secondly information about the relative phases of the periodic components is lost when they are transformed to the frequency domain. Morgan and Tindal (1990) illustrate the effect of varying phase angles by a comparison of plots of $(\cos \omega t + 0.5 \cos 3\omega t)$ and $(\cos \omega t - 0.5 \cos 3\omega t)$ which is reproduced in Figure 5.34. For a material with an S/N curve exponent of 5, stresses conforming to the first time history would result in 5.25 times as much fatigue damage as stresses conforming to the second.

Bishop, Wang and Lack (1995) developed a modified form of the Dirlik formula to include a single periodic component, using a neural network approach to determine the different parameters in the formula from computer simulations.

Madsen *et al.* (1984) adopted a different approach to the problem of determining fatigue damage resulting from combined stochastic and periodic loading, involving the derivation of a single equivalent sinusoidal loading that would produce the same fatigue damage as the actual loading. The method applies a reduction factor, g , which is dependent on bandwidth, to account for the reduced cycle ranges implicit in a wide band as opposed to a narrow

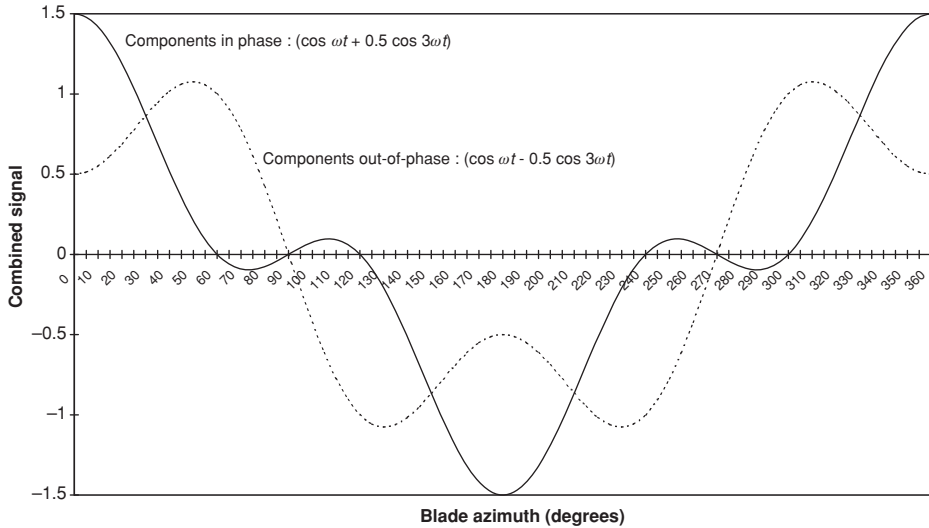


Figure 5.34 Effect of variation of phase angle between harmonics on combined signal

band process, and utilises Rice's pdf for the peak value of a single sinusoid combined with a narrow band stochastic process, substituting half the maximum range of the periodic signal, including harmonics, for the amplitude of the sinusoid. A fuller summary is given in Hoskin *et al.* (1989). They concluded, along with Morgan and Tindal (1990), that the Madsen method yielded slightly less accurate fatigue damage values than the Dirlik method for the MS1 monitored data for flapwise bending referred to above.

Ragan and Manuel (2007) used about 2500 datasets of blade in-plane and out-of-plane bending moments from a 1.5 MW turbine in Colorado to compare fatigue damage equivalent loads calculated in the frequency domain by the Dirlik method with corresponding values calculated in the time domain. They concluded that the Dirlik method performed reasonably well for out-of-plane moments but very poorly for blade in-plane bending moments, which have a large periodic component.

5.9.4 Wind simulation

Wind simulation, which was introduced in Section 5.7.6, has two significant advantages over the methods described above for fatigue damage evaluation. Firstly it can handle non-linear relationships between wind speed fluctuations and blade loadings in the calculation of stochastic loads, and secondly it avoids the difficulty of deriving the fatigue stress ranges arising from combined periodic and stochastic load components. It is therefore currently the favoured method for detailed fatigue design. The procedure is essentially as follows:

- 1) Generate a 3D 'run of wind' for the chosen mean wind speed, with the desired shear profile and tower shadow correction.
- 2) Perform a step-by-step dynamic analysis on the turbine operating in this wind field, to obtain in-plane and out-of-plane bending moment time histories at different radii.

- 3) Convert these bending moment time histories to time histories of bending moments about the principal axes.
- 4) Compute stress time histories at chosen points on each cross-section.
- 5) Derive the number of cycles in each stress range 'bin' by Rainflow Cycle Counting (see Section 5.9.5 below).
- 6) Scale up the cycle numbers in line with the predicted number of hours of operation at the chosen mean wind speed.
- 7) Calculate corresponding fatigue damage numbers based on the applicable S-N curve.
- 8) Repeat above steps for different mean wind speeds, and total the resulting fatigue damages at each point.

A computationally simpler alternative is to generate a 1D 'run of wind' (in which only the longitudinal component of turbulence is modelled), and run a number of simulations at different, fixed yaw angles.

The duration of wind simulations is limited by available computing power, with a time history length of 600 seconds being frequently chosen. A consequence of this is that a single simulation will not provide an accurate picture of the infrequent high stress range fatigue cycles, which can have a disproportionate effect on fatigue damage for materials with high m value, such as those used for blades. However, this inaccuracy can be reduced (and quantified) by running several simulations with different random number seeds at each wind speed – see Thomsen (1998).

5.9.5 Fatigue cycle counting

As noted in Section 5.9.4, the dynamic analysis of turbine behaviour in a simulated wind field yields time histories of loads or stresses which then need to be processed to abstract details of the fatigue cycles. There are two established methods of fatigue cycle counting: the reservoir method and the rainflow method, both of which yield the same result.

In the reservoir method, the load or stress history (with time axis horizontal) is imagined as the cross-section of a reservoir, which is successively drained from each low point, starting at the lowest and working up. Each draining operation then yields a load or stress cycle. See BS 5400 (1980) for a full description.

The rainflow method was first proposed by Matsuishi and Endo in 1968, and its title derives from the concept of water flowing down the 'rooves' formed when the time history is rotated so that the time axis is vertical. However, the following description not involving the rainflow analogy may be easier to understand.

The first step is to reduce the time history to a series of peaks and troughs, which are then termed extremes. Then, each group of four successive extremes is examined in turn to determine whether the values of the two intermediate extremes lie between the values of the initial and final extremes. If so, the two intermediate extremes are counted as defining a stress cycle, which is then included in the cycle count, and the two intermediate extremes are deleted from the time history. The process is continued until the complete series of extremes forming the time history has been processed in this way. Then the sequence remaining will consist simply of a diverging and a converging part from which the final group of stress ranges can be extracted. See 'Fatigue Characteristics' in the IEA series of Recommended Practices for

Wind Turbine Testing and Evaluation for a full description of the method and for details of algorithms that can be used for automating the process.

Although, in principle, the fatigue cycles obtained from, say, a 600 second time history could be listed individually, it is normal to reduce the volume of data by allocating individual cycles to a series of equal load or stress ranges known as 'bins' – for example, 0–2, 2–4, 4–6 N/mm² etc. The fatigue spectrum is then presented in terms of the number of cycles falling into each 'bin'.

5.10 Hub and low speed shaft loading

5.10.1 Introduction

The loadings on the hub consist of the aerodynamic, gravity and inertia loadings on the blades and the equal and opposite (discounting hub self-weight) reaction from the shaft. For fixed hub machines, the loading on the shaft will include a significant moment arising from blade aerodynamic loads, but in the case of teetered two-bladed rotors this moment will be virtually eliminated. In either case, however, the cantilevered low-speed shaft will experience large fluctuating moments due to rotor weight as it rotates. Figure 5.35 shows a low-speed shaft and front bearing in a factory prior to assembly.

The shaft moments due to out-of-plane loads on the blades can be expressed as moments about a pair of rotating axes, one perpendicular to blade and the other parallel to it. In the case



Figure 5.35 Low-speed shaft and front bearing before assembly. The hub mounting flange at the right-hand end is bolted to a temporary support to allow the bearing to be threaded on the shaft. Reproduced by permission of NEG Micon

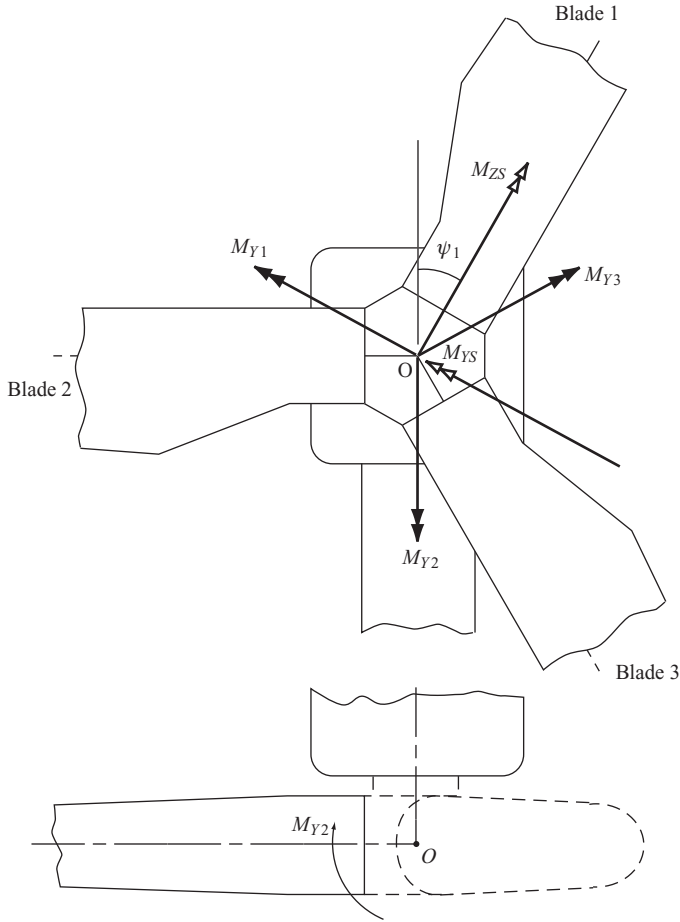


Figure 5.36 Shaft bending moments with rotating axis system referred to blade 1

of a three bladed rotor, these moments are respectively as follows:

$$M_{YS} = \Delta M_{Y1} - \frac{1}{2} (\Delta M_{Y2} + \Delta M_{Y3}) \quad M_{ZS} = \frac{\sqrt{3}}{2} (\Delta M_{Y3} - \Delta M_{Y2}) \quad (5.118)$$

Here ΔM_{Y1} , ΔM_{Y2} and ΔM_{Y3} are the fluctuations of the blade out-of-plane moments about the hub centre (M_{Y1} , M_{Y2} and M_{Y3}) about the mean value. See Figure 5.36.

5.10.2 Deterministic aerodynamic loads

The deterministic aerodynamic loads on the rotor may be split up into a steady component, equal for each blade, and a periodic component, also equal for each blade, but with differing phase angles. The blade root out-of-plane bending moments due to the first component will be in equilibrium, and will apply a 'dishing' moment to the hub which will result in tensile

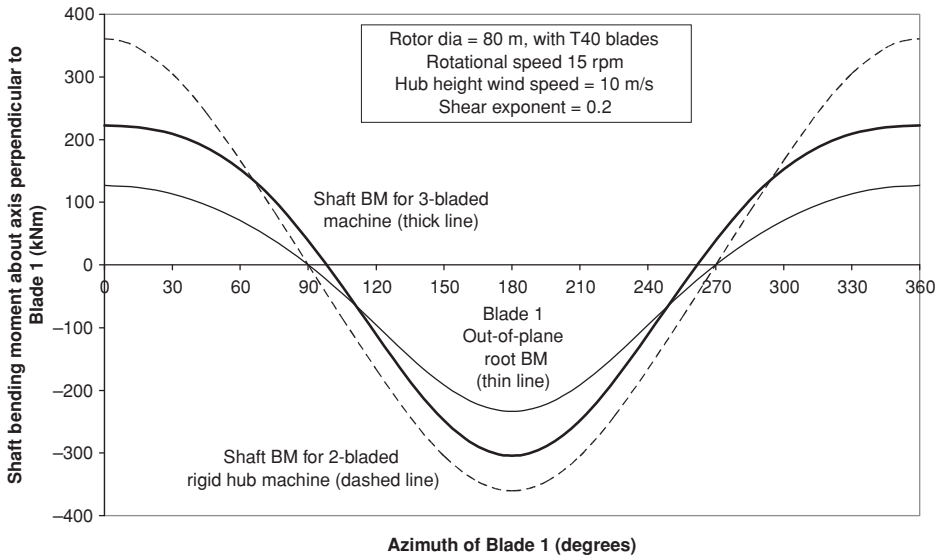


Figure 5.37 Shaft bending moment fluctuations due to wind shear

stresses in the front and compression stresses in the rear. These stresses will be uniaxial for a two-bladed rotor, and biaxial for a three-bladed rotor.

The fluctuations in out-of-plane blade root bending moment due to wind shear, shaft tilt and yaw misalignment will often be approximately sinusoidal, with a frequency equal to the rotational frequency. Using Equations 5.118, it is easily shown that, for a sinusoidally varying blade root bending moment with range ΔM , the range of the resulting shaft bending moment is $1.5\Delta M$ for a three-bladed machine and $2\Delta M$ for a rigid hub two-bladed machine.

In the case of wind shear conforming to a power law, the loading on a horizontal blade is always greater than the average of the loadings on blades pointing vertically upwards and downwards, so the loading departs significantly from sinusoidal. The shaft bending moment fluctuations due to wind shear with a 0.2 exponent are compared in Figure 5.37 for two- and three-bladed rigid hub machines operating at 15 rpm in a hub-height wind speed of 10 m/s. The ratio of moment ranges is still close to 2:1.5.

5.10.3 Stochastic aerodynamic loads

The out-of-plane blade root bending moments arising from stochastic loads on the rotor will result in both a fluctuating hub ‘dishing’ moment (see above) and fluctuating shaft bending moments. For a two-bladed, rigid hub rotor, the shaft moment is equal to the difference between the two out-of-plane blade root bending moments, or teeter moment, the standard deviation of which is given by Equation 5.102a. Similarly, the standard deviation of the mean of these two moments (i.e. the ‘dishing’ moment) is given by Equation 5.103.

The derivation of the standard deviation of the shaft moment for a three-bladed machine is at first sight more complicated, as the integration has to be carried out over three blades instead of two. However, if the shaft moment about an axis parallel to one of the blades,

M_{ZS} (Figure 5.36), is chosen, the contribution of loading on that blade disappears, and the expression for the shaft moment standard deviation becomes

$$\sigma_{M_{ZS}}^2 = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_{-R}^R \int_{-R}^R \kappa_u^o(r_1, r_2, 0) c(r_1) c(r_2) \frac{\sqrt{3}}{2} r_1 \frac{\sqrt{3}}{2} r_2 |r_1| |r_2| dr_1 dr_2 \quad (5.119)$$

where the limits of the integrations refer to the other two blades. $\kappa_u^o(r_1, r_2, 0)$ is given by Equation 5.51, with $\Omega \tau$ set equal to zero when r_1 and r_2 are radii to points on the same blade, and replaced by $2\pi/3$ when r_1 and r_2 are radii to points on different blades. Note that, compared with the two bladed case, the cross correlation function, $\kappa_u^o(r_1, r_2, 0)$ will be increased when r_1 and r_2 relate to *different* blades, because of the reduced separation between the two blade elements resulting from the 120° angle between the blades. Equation 5.119a can be rewritten in terms of the normalised cross correlation function, $\rho_u^o(r_1, r_2, 0) = \kappa_u^o(r_1, r_2, 0)/\sigma_u^2$, as follows:

$$\sigma_{M_{ZS}}^2 = \sigma_u^2 \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_{-R}^R \int_{-R}^R \rho_u^o(r_1, r_2, 0) c(r_1) c(r_2) \frac{\sqrt{3}}{2} r_1 \frac{\sqrt{3}}{2} r_2 |r_1| |r_2| dr_1 dr_2 \quad (5.119a)$$

In the case of 80 m diameter turbines with T40 blades operating in wind with a turbulence length scale of 147 m, the standard deviation of shaft moment due to stochastic loading for a three-bladed machine is 82% of that for a two-bladed, fixed hub machine rotating at the same speed. This ratio would rise to $\sqrt{3}/2$ if the effect on the cross correlation function of the 120° blade spacing were ignored.

It is worth noting that, for a three-bladed machine, the standard deviation of the shaft moment M_{YS} due to stochastic loading is the same as that of M_{ZS} .

By analogy with the derivation of the shaft moment above, the standard deviation of the hub ‘dishing’ moment for a three bladed machine due to stochastic loading is given by:

$$\sigma_{M_h}^2 = \frac{1}{4} \sigma_u^2 \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \int_{-R}^R \int_{-R}^R \rho_u^o(r_1, r_2, 0) c(r_1) c(r_2) \frac{\sqrt{3}}{2} r_1^2 \frac{\sqrt{3}}{2} r_2^2 dr_1 dr_2 \quad (5.120)$$

where the integrations are carried out over two blades only, and the cross correlation function is modified as before to account for the 120° angle between the blades.

It can be shown that

$$\frac{1}{4} \sigma_{M_{ZS}}^2 + \sigma_{M_h}^2 = \frac{3}{4} \sigma_{M_{Y1}}^2 \quad (5.121)$$

5.10.4 Gravity loading

An important component of shaft loading is the cyclic cantilever bending moment due to rotor weight, which usually has a dominant effect on shaft fatigue design. As an illustration, a rotor consisting of three 8 tonne blades, and a 24 tonne hub cantilevered 1.7 m beyond the shaft main bearing, will produce a maximum shaft gravity moment of about 800 KNm. This compares with a shaft moment range due to wind shear of 360 KNm for a hub-height wind of 10 m/s and a shear exponent of 0.2, and a shaft moment standard deviation of 350 KNm due to turbulence, taking a turbulence intensity of 21% and the same hub-height mean wind speed. Note that the shaft moment due to wind shear relieves that due to gravity, so it would be wise to adopt a smaller wind shear exponent for shaft fatigue calculations.

5.11 Nacelle loading

5.11.1 Loadings from rotor

The previous section considered the moments applied to the shaft by the rotor hub using an axis system rotating with the shaft. In addition to these moments, the shaft also experiences an axial load due to rotor thrust, and radial forces arising from differential blade edgewise loadings and any out-of-balance centrifugal force.

In order to calculate loadings on the elements of the nacelle structure, it is first necessary to transform the shaft loads (or the constituent blade loads) defined in terms of the rotating axis system into nacelle loads expressed in terms of a fixed axis system. Here the conventional system, in which the x -axis is downwind, the y -axis is horizontal to starboard and the z -axis is vertically upwards, will be adopted. Thus, the moments acting on the nacelle about the y - and z -axes as a result of the blade root out-of-plane bending moments are as follows for a three-bladed machine with shaft tilt η :

$$M_{YN} = M_{Y1} \cos \psi + M_{Y2} \cos(\psi - 120^\circ) + M_{Y3} \cos(\psi - 240^\circ) \quad (5.122)$$

$$M_{ZN} = (M_{Y1} \sin \psi + M_{Y2} \sin(\psi - 120^\circ) + M_{Y3} \sin(\psi - 240^\circ)) \cos \eta \quad (5.123)$$

where ψ is the azimuth of blade 1. See Figure 5.38.

It is instructive to compare the moments acting on the nacelle due to deterministic loading for three-bladed and two-bladed machines. The fluctuations of out-of-plane root

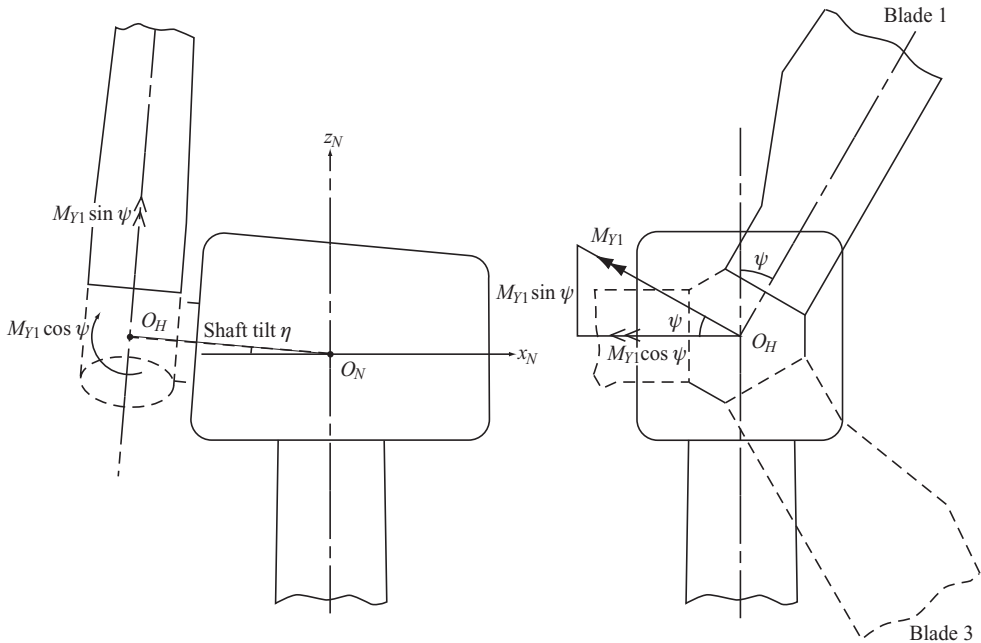


Figure 5.38 Components of blade 1 out-of-plane root bending moment about fixed set of axes

Table 5.7 Comparison between nacelle moments due to deterministic loads for two- and three-bladed machines

	Nacelle moments resulting from out-of-plane blade root bending moment fluctuations due to wind shear and yaw misalignment approximated by: $M_{Y1} = M_0 \cos \psi, M_{Y2} = M_0 \cos(\psi - 2\pi/N)$ etc.		Nacelle moments resulting from out-of-plane blade root bending moment fluctuations due to shaft tilt approximated by: $M_{Y1} = M_0 \sin \psi, M_{Y2} = M_0 \sin(\psi - 2\pi/N)$ etc.	
	Nacelle nodding moment, M_{YN}	Nacelle yaw moment, M_{ZN}	Nacelle nodding moment, M_{YN}	Nacelle yaw moment, M_{ZN}
Three-bladed machine	$1.5 M_0$	Zero	Zero	$1.5 M_0 \cos \eta$
Two-bladed, rigid hub machine	$M_0(1 + \cos 2\psi)$	$M_0 \sin 2\psi \cos \eta$	$M_0 \sin 2\psi$	$M_0(1 - \cos 2\psi) \cos \eta$

bending moment due to wind shear and yaw misalignment are approximately proportional to the cosine of blade azimuth for an unstalled blade. Substituting $M_{Y1} = M_0 \cos \psi$, $M_{Y2} = M_0 \cos(\psi - 2\pi/N)$ etc. into Equations 5.122 and 5.123 yields $M_{YN} = 1.5 M_0$ and $M_{ZN} = 0$ for a three-bladed machine, whereas the corresponding results for a rigid hub two-bladed machine are $M_{YN} = M_0(1 + \cos 2\psi)$ and $M_{ZN} = M_0 \sin 2\psi \cos \eta$. Thus, the moments on the nacelle are constant for a three-bladed machine, but continually fluctuating with amplitude M_0 for a rigid hub two-bladed machine. Parallel results are obtained for $M_{Y1} = M_0 \sin \psi$ which approximates to the out-of-plane root bending moment due to shaft tilt – again for an unstalled blade. The full comparison is given in Table 5.7.

It is clear that the moments acting on the nacelle due to deterministic loading are much more benign for a three-bladed rotor than for a two-bladed rotor with rigid hub.

In the case of three-bladed machines, the standard deviation of shaft bending moment due to stochastic rotor loading is independent of the rotating axis chosen (Section 5.10.3), so the standard deviation of the resulting moment on the nacelle will take the same value about both the nacelle y- and z-axes.

5.11.2 Cladding loads

Except in the case of sideways wind loading, cladding loads are not usually of great significance. They may be calculated according to the rules given in standard wind loading codes. For sideways wind loading, a drag factor of 1.2 will generally be found to be conservative.

5.12 Tower loading

5.12.1 Extreme loads

It is customary to base the calculation of extreme loads on a non-operational turbine on the 50 year return three second gust. Several loading configurations may need to be considered,

and the critical load case for the tower base will generally differ from that for the tower top. In addition, it is necessary to investigate the extreme operational load cases, as these can sometimes govern instead if the tip speed is high in relation to the design gust speed.

In the case of non-operational, stall-regulated machines, the critical case for the tower base occurs when the wind is blowing from the front and inducing maximum drag loading on the blades. By contrast, sideways wind loading to produce maximum lift on a blade pointing vertically upwards or rear wind loading on the rotor with one blade shielded by the tower will produce the maximum tower top bending moment.

One of the benefits of pitch-regulation of three-bladed machines is that blade feathering at shut down considerably reduces non-operational rotor loading. The critical configuration as far as tower base bending moment is concerned is sideways wind loading, with two of the blades inclined at 30° to the vertical. The horizontal component of the loading on these blades is $\cos^3 30^\circ$ of the loading on a vertical blade, so that the total rotor loading is only 43.3% ($= 100 \cdot \sqrt{3}/4\%$) of the maximum experienced by a stall regulated machine.

The cases of sideways wind loading on a wind turbine referred to above can only arise if the yaw drive is disabled by grid loss for sufficient time for a 90° wind direction change to take place, so IEC 61400-1 Edition 3 treats this case (DLC 6.2) as an abnormal load case with a reduced load factor of 1.1 (in place of 1.35 for normal load cases). As a result of this, the load case causing extreme tower base overturning moment on pitch-regulated machines is not always clear cut. If the rotor is braked with one blade vertically upwards and the yaw angle is small (corresponding to IEC 61400-1 Edition 3 DLC 6.1), the loading on the top blade could approach maximum lift. Loads on the other blades would be smaller and their horizontal components would probably act in the opposite direction. The maximum loading on the top blade would be $(0.5\rho V^2)1.5A_B$, ignoring wind shear, where A_B is the blade area. On the other hand, DLC 6.2 gives a drag loading of $2 \cos^3 30^\circ (0.5\rho V^2)1.3A_B = (0.5\rho V^2)1.69A_B$, again ignoring wind shear, which becomes $(0.5\rho V^2)1.86A_B$ after inclusion of the load factor. This is somewhat less than the factored load on a vertical blade experiencing maximum lift of $2.025(0.5\rho V^2)A_B$. However, the drag loading on the blades in DLC 6.2 acts in the same direction as the drag loading on the tower, whereas the lift load on the vertical blade in DLC 6.1 acts at right angles to the drag loading on the tower.

Information on the drag factors appropriate for cylindrical and lattice towers is to be found in – EN 1991-1-4: 2005 ‘Eurocode 1: Actions on structures – Part 1-4: Wind actions’, and in national codes such as BS 8100 (1986) or DS 410 (1983). The drag factor for a cylindrical tower is typically 0.6–0.7. Rotor loading is generally the dominating component of tower base moment for stall-regulated machines, but with pitch-regulated machines the contributions of tower loading and rotor loading are often of similar magnitude.

5.12.2 Dynamic response to extreme loads

Just as in the case of the single, stationary cantilevered blade considered in Section 5.6.3, the quasistatic bending moments in the tower calculated for the extreme gust speed will be augmented by inertial moments resulting from the excitation of resonant tower oscillations by turbulence. As before, it is convenient to express this augmentation in terms of a dynamic factor, Q_D , defined as the ratio of the peak moment over a ten minute period, including resonant excitation of the tower, to the peak quasistatic moment over the same period. Thus,

$$M_{Max} = \frac{1}{2} \rho U_{e50}^2 H \oint C_f \left(\frac{z}{H} \right)^{1+2\alpha} dA \cdot Q_D \quad (5.124)$$

where

U_{e50} is the 50 year return gust speed at hub-height

z is height above ground

H is the hub height

C_f is the force factor (lift or drag) for the element under consideration

α is the shear exponent, taken as 0.11 in IEC 61400-1

and

$$Q_D = \frac{1 + g \left(2 \frac{\sigma_u}{U} \right) \sqrt{K_{SMB} + \frac{\pi^2}{2\delta} R_u(n_1) K_{Sx}(n_1) \cdot \lambda_{M1}^2}}{1 + g_0 \left(2 \frac{\sigma_u}{U} \right) \sqrt{K_{SMB}}} \quad (5.17)$$

The integral sign $\oint$ signifies that the integral is to be undertaken over each blade, the nacelle and the tower. The derivation of Equation 5.17 is explained in Section 5.6.3 and the Appendix in relation to a cantilevered blade.

The essentially similar procedure for a tower supporting a braked rotor and nacelle is as follows:

- 1) Calculate the resonant size reduction factor, $K_{Sx}(n_1)$, which reflects the effect of the lack of correlation of the wind fluctuations at the tower natural frequency along the blades and tower. Adopting an exponential expression for the normalised co-spectrum as before, Equation A5.25 becomes:

$$K_{Sx}(n_1) = \frac{\oint \oint \exp[-Csn_1/\bar{U}] C_f^2 c(r) c(r') \mu_1(r) \mu_1(r') dr dr'}{\left(\oint C_f c(r) \mu_1(r) dr \right)^2} \quad (5.125)$$

where

- the integral sign $\oint$ denotes integration over the blades and the tower;
- r and r' denote radius in the case of the blades and depth below the hub in the case of the tower;
- s denotes the separation between the elements dr and dr' ;
- C_f is the relevant force coefficient;
- $c(r)$ denotes chord in the case of the blades and diameter in the case of the tower;
- $\mu_1(r)$ denotes the tower first mode shape.

This expression can be considerably simplified by setting $\mu_1(r)$ to unity for the rotor and ignoring the tower loading contribution entirely. This is not unreasonable, as only loading near the top of the tower is of significance, and this does not add much to the spatial extent of the loaded area.

- 2) Calculate the damping logarithmic decrement, δ , for the tower first mode. The aerodynamic component is given by

$$\delta_a = 2\pi\xi_a = 2\pi \frac{c_{a1}}{2m_{T1}\omega_1} = \frac{\oint \hat{c}_a(r)\mu_1^2(r) dr}{2m_{T1}n_1} \quad (5.126)$$

where m_{T1} is the generalised mass of the tower, nacelle and rotor (including the contribution of rotor inertia) with respect to the first mode given by (Equation 5.110), and n_1 is the tower natural frequency in Hz. For a stall-regulated machine facing the wind, the rotor contribution to aerodynamic damping is simply $\rho\bar{U}C_D A_R/2m_{T1}n_1$ where A_R is the rotor area.

- 3) Calculate the standard deviation of resonant nacelle displacement according to

$$\frac{\sigma_{x1}}{\bar{x}_1} = 2 \frac{\sigma_u}{\bar{U}} \frac{\pi}{\sqrt{2\delta}} \sqrt{R_u(n_1)} \sqrt{K_{Sx}(n_1)} \quad (5.7)$$

- 4) Calculate the ratio λ_{M1} , which relates the ratio of the standard deviation of resonant tower base moment to the mean value to the corresponding ratio for nacelle displacement as follows:

$$\frac{\sigma_{M1}}{\bar{M}} = \lambda_{M1} \frac{\sigma_{x1}}{\bar{x}_1} \quad (5.8a)$$

If $\mu_1(r)$ is set to unity for the rotor, λ_{M1} is given by:

$$\lambda_{M1} = \frac{\int_0^H m(z)\mu_1(z) \cdot z dz \left\{ C_D A_R + \int_0^H C_f \left[\frac{U(z)}{\bar{U}} \right]^2 d(z)\mu_1(z) dz \right\}}{m_{T1} H \left\{ C_D A_R + \int_0^H C_f \left[\frac{U(z)}{\bar{U}} \right]^2 d(z) \frac{z}{H} dz \right\}} \quad (5.127)$$

where z is the height up the tower measured from the base, $d(z)$ is the tower diameter at height z and H is the hub-height. If the loading on the tower is relatively small, this approximates to

$$\lambda_{M1} = \frac{\int_0^H m(z)\mu_1(z) \cdot z dz}{m_{T1} \cdot H} \quad (5.127a)$$

which is close to unity because the tower head mass dominates the integral.

- 5) Calculate the size reduction factor for the root bending moment quasistatic or background response, K_{SMB} , which reflects the lack of correlation of the wind fluctuations along the blades and tower. K_{SMB} may be derived from a similar expression to that for the resonant size reduction factor given in Equation 5.125, but with the exponential function modified to $\exp[-s/0.3L_u^x]$.

- 6) Calculate the peak factors for the combined (i.e. resonant plus quasistatic) and quasistatic responses in terms of the respective zero up-crossing frequencies. In estimating the zero up-crossing frequency of the quasistatic response, the blade area should be replaced by the rotor area in Equation A5.57.
- 7) Substitute the parameter values derived in steps 1–6 into Equation 5.17 to obtain the dynamic factor, Q_D .

In the case of a rotor which is allowed to idle when shutdown, turbine geometry is continually changing as the rotor rotates to and fro in response to wind gusts, so the calculation of resonant tower excitation, should it occur, becomes a complex undertaking.

5.12.3 Operational loads due to steady wind (deterministic component)

Tower fore and aft bending moments result from rotor thrust loading and rotor moments. The moments acting on the nacelle due to deterministic rotor loads have already been described in Section 5.11.1. Although the thrust loads on individual blades vary considerably with azimuth as a result of yaw misalignment, shaft tilt or wind shear, the fluctuations on different blades balance each other, so that the total rotor thrust shows negligible azimuthal variation as a result of these effects. For example, on two-bladed machines, a wind shear exponent of 0.2 results in a rotor thrust variation of about $\pm 1\%$. Tower shadow loading results in a sinusoidal tower top displacement at blade passing frequency – see Figure 5.32.

Figure 5.39 illustrates the variation of rotor thrust with wind speed for stall and pitch regulated 80 m diameter three-bladed machines.

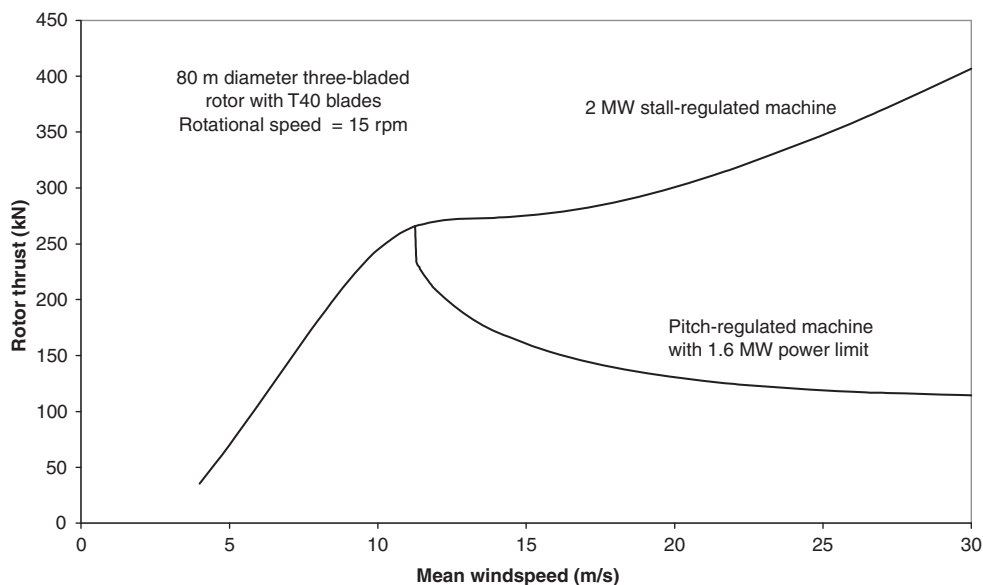


Figure 5.39 Rotor thrust during operation in steady, uniform wind: variation with wind speed for similar stall-regulated and pitch-regulated machines

5.12.4 Operational loads due to turbulence (stochastic component)

Analysis in the frequency domain

Except near the top of the tower, the dominant source of fore-aft stochastic tower bending moments is rotor thrust. The standard deviation of rotor thrust can be expressed in terms of the turbulence intensity and the cross correlation function between wind fluctuations at different points on the rotor, following the method used for deriving the standard deviation of stochastic blade root bending moment in Section 5.7.5. As before, a linear relation between the wind fluctuations and the resultant load fluctuations is assumed, so that the perturbation of loading per unit length of blade, q , at radius r is given by

$$q = \frac{1}{2} \rho \Omega r c(r) \frac{dC_L}{d\alpha} u \quad (5.25)$$

and the perturbation of rotor thrust by

$$\Delta T = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right) \oint u c(r) r dr \quad (5.128)$$

where the integral sign $\oint$ signifies that the integration is carried out over the whole rotor. Hence the following expression for the variance of the rotor thrust is obtained:

$$\sigma_T^2 = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \sigma_u^2 \oint \oint \rho_u^o(r_1, r_2, 0) c(r_1) c(r_2) r_1 r_2 dr_1 dr_2 \quad (5.129)$$

where $\rho_u^o(r_1, r_2, 0)$ is the normalised cross correlation function, $\kappa_u^o(r_1, r_2, 0) / \sigma_u^2$ for points at radii r_1 and r_2 on the same or on different blades. $\kappa_u^o(r_1, r_2, 0)$ is given by Equation 5.51, with $\Omega \tau$ replaced by the phase angle between the two blades on which r_1 and r_2 are measured. For a three-bladed, 80 m diameter rotor and an integral length scale of 147 m, the reduction in the standard deviation of the stochastic rotor thrust fluctuations is about 20% due to the lack of correlation of the wind speed variations over the rotor. If the machine is rotating at 15 rpm in an 8 m/s wind and the turbulence intensity is 20%, the rotor thrust standard deviation will be about 38 kN – that is, 22% of the steady value.

The derivation of the expression for the power spectrum of rotor thrust parallels that for the power spectrum of blade root bending moment (Section 5.7.5), yielding:

$$S_T(n) = \left(\frac{1}{2} \rho \Omega \frac{dC_L}{d\alpha} \right)^2 \oint \oint S_{uJ,K}^o(r_1, r_2, n) c(r_1) c(r_2) r_1 r_2 dr_1 dr_2 \quad (5.130)$$

where $S_{uJ,K}^o(r_1, r_2, n)$ is the rotationally sampled cross spectrum for points at radii r_1 and r_2 on blades J and K respectively. Note that on a machine with three blades, A , B and C , $S_{uJ,K}^o(r_1, r_2, n)$ is complex when J and K are different, but $S_{uA,B}^o(r_1, r_2, n)$ and $S_{uA,C}^o(r_1, r_2, n)$ are complex conjugates, so the double integral in Equation 5.130 is still real. An example power spectrum of rotor thrust for an 80 m diameter three-bladed machine is shown in Figure 5.40. It can be seen that there is some concentration of energy at the blade passing frequency of 0.75 Hz due to gust slicing, but that the effect is not large. The concentration effect is significantly greater for two-blade machines – see Figure 5.41. This shows the power

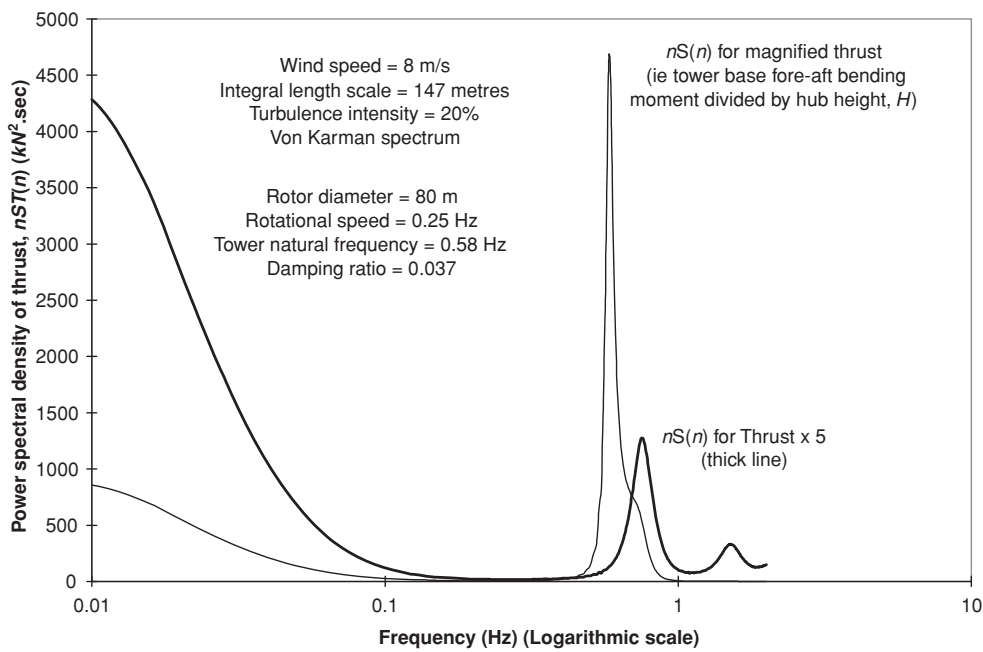


Figure 5.40 Power spectra of rotor thrust and resultant tower base fore-aft bending moment for three-bladed, 80 m diameter turbine

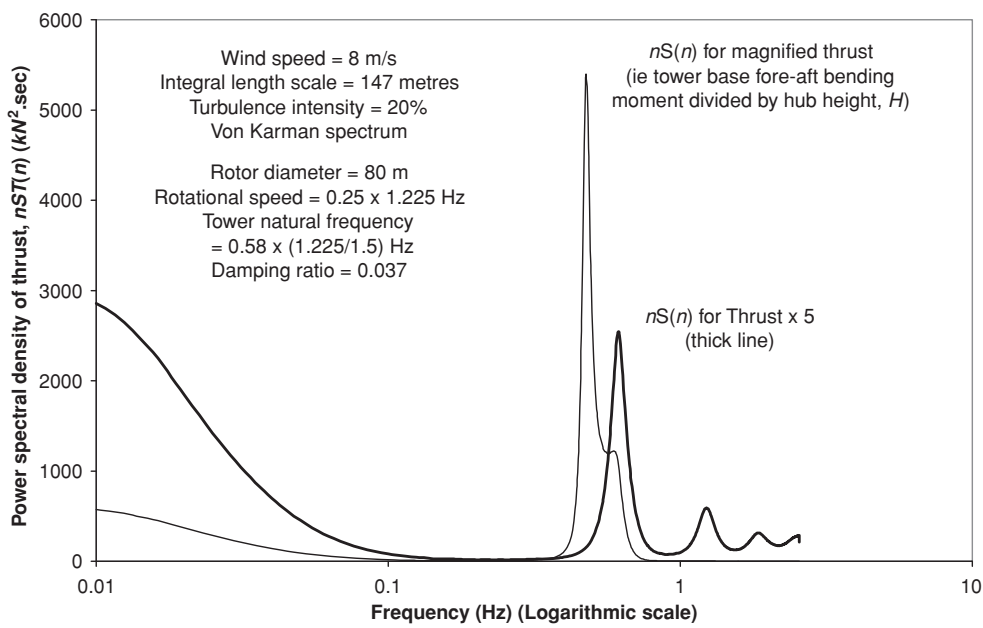


Figure 5.41 Power spectra of rotor thrust and resultant tower base fore-aft bending moment for two-bladed, 80 m diameter turbine

spectrum of rotor thrust for a two-bladed machine with the same blade platform, but rotating 22.5% faster to give comparable performance.

In addition to thrust fluctuations, longitudinal turbulence will also cause rotor torque fluctuations and in-plane rotor loads due to differential loads on different blades, both of which will result in tower sideways bending moments. The expression for the in-plane component of aerodynamic lift per unit length, $-F_Y(r) = \frac{1}{2}\rho W^2 C_L c(r) \sin \phi$, can be differentiated with respect to the wind fluctuation as follows:

$$\begin{aligned} -\frac{dF_Y}{du} &= \frac{1}{2}\rho c(r) \frac{d}{du} [W^2 \sin \phi \cdot C_L] = \frac{1}{2}\rho c(r) \frac{d}{du} [W\{U_\infty(1-a) + u\}C_L] \\ &\cong \frac{1}{2}\rho c(r) W \left[C_L + \sin \phi \frac{dC_L}{d\alpha} \right] \end{aligned}$$

so, approximately,

$$-\frac{dF_Y}{du} = \left(\frac{1}{2}\rho\Omega \frac{dC_L}{d\alpha} \right) c(r) \cdot r \left[\frac{C_L}{dC_L/d\alpha} + \sin \phi \right] \quad (5.131a)$$

Thus, the standard deviation of rotor torque is approximately given by

$$\sigma_Q = \left(\frac{1}{2}\rho\Omega \frac{dC_L}{d\alpha} \right) \sigma_u \left\{ \oint r^2 c(r) \left[\frac{C_L}{dC_L/d\alpha} + \sin \phi \right] dr \right\} \quad (5.131b)$$

(which parallels Equation 5.26) provided the relationship between blade loading and wind speed fluctuation remains linear and the turbulence length scale is large compared with rotor diameter. Equation 5.131b can be used to derive an expression for the variance of the rotor torque in the same way as for rotor thrust above. At the top of the tower the stochastic M_X (i.e. side-to-side) moment due to rotor torque fluctuations is typically of the same order of magnitude as the stochastic M_Y (i.e. fore-aft) moment due to differential out-of-plane loads on the rotor, but at the tower base the dominant effect of rotor thrust loading means that the stochastic side-to-side moments are usually significantly less than the stochastic fore-aft moments before the excitation of tower resonance is taken into account.

Analysis in the time domain

As noted in Section 5.7.5, there are situations, such as operation in stalled flow, when the linear relationship between blade loading and wind speed fluctuations required for analysis in the frequency domain does not apply. In these cases, recourse must be made to analysis in the time domain using wind simulation techniques such as described in Section 5.7.6.

5.12.5 Dynamic response to operational loads

The power spectrum of rotor thrust will usually contain some energy at the tower natural frequency, leading to dynamic magnification of deflections and, hence, of tower bending

moments. The power spectrum of hub deflection, $S_{x1}(n)$, resulting from the excitation of the tower first fore-aft flexural mode, is related to the power spectrum of rotor thrust by

$$S_{x1}(n) = \frac{S_T(n)}{k_1^2} \frac{1}{[(1 - n^2/n_1^2) + 4\xi_1^2 n^2/n_1^2]} \quad (5.132)$$

This relation is analogous to Equation 5.90, and derived in the same way.

The amplitude of tower base fore-aft moment at resonance in the first mode, M_{Y1} , can be derived from the corresponding amplitude of hub deflection, x_{H1} , as follows

$$M_{Y1} = \omega_1^2 x_{H1} \int_0^H m(z) \mu(z) z dz = \omega_1^2 x_{H1} m_{T1} H \frac{\int_0^H m(z) \mu(z) z dz}{H \int_0^H m(z) \mu^2(z) dz} \quad (5.133)$$

The quotient on the right hand side is close to unity because of the dominance of the tower head mass, so, substituting k_1 for $\omega_1^2 m_{T1}$, the equation reduces to $M_{Y1} = x_{H1} k_1 H$, which applies at any exciting frequency. Hence the power spectrum for the tower base fore-aft bending moment due to rotor thrust loading is given by

$$S_{My1}(n) = S_T(n) \cdot H^2 \frac{1}{[(1 - n^2/n_1^2) + 4\xi_1^2 n^2/n_1^2]} \quad (5.134)$$

The aerodynamic damping is almost entirely provided by the rotor, the damping ratio for the first tower mode being approximately

$$\xi_{a1} = N \frac{\frac{1}{2} \rho \Omega \int_0^R \frac{dC_l}{d\alpha} r c(r) dr}{2m_{T1}\omega_1} \quad (5.135)$$

where N is the number of blades (see Section 5.8.4). The overall damping ratio is obtained by adding this to the structural damping ratio for the tower (see Table 5.5), and is generally low compared to the blade first mode damping because of the large tower head mass. The effect of a low damping ratio is illustrated by the power spectrum of fore-aft tower bending moment shown in Figure 5.40, which has a very high peak at the tower natural frequency of 0.58 Hz, despite this frequency being somewhat removed from the blade passing frequency of 0.75 Hz. The damping ratio is calculated as 0.037, consisting of 0.035 due to aerodynamic damping (based on a tower head mass of 120 tonnes) and 0.002 due to structural damping (for a welded steel tower).

In the example shown in Figure 5.40, the tower dynamic response increases the standard deviation of the tower base fore-aft bending moment by 9%. However, the effect of tower dynamic response results in a larger increase of 15% in the case of the two-bladed machine featured in Figure 5.41, despite the reduction in tower natural frequency to maintain the same tower natural frequency to blade passing frequency ratio.

It is important to note that the rotor provides negligible aerodynamic damping in the side-to-side direction, so that effectively the only damping present is the structural damping.

This means that, even though the side-to-side loadings are small in relation to the fore-aft loads, the side-to-side tower moment fluctuations can sometimes approach the fore-aft ones in magnitude.

5.12.6 Fatigue loads and stresses

The tower moments at height z are related to the hub height loads as follows, omitting the tower inertial loads:

$$\begin{aligned} M_Y(z, t) &= F_X(H, t) \cdot (H - z) + M_Y(H, t) \\ M_X(z, t) &= -F_Y(H, t) \cdot (H - z) + M_X(H, t) \\ M_Z(z, t) &= M_Z(H, t) \end{aligned} \quad (5.136)$$

For three-bladed machines, the five hub-height fatigue loads are almost entirely stochastic, because the deterministic load component is either constant (for a given mean wind speed) or negligible, and it is instructive to consider how they relate to one another. Recognising that the centre of any gust lying off the rotor centre will be located at a random azimuth, then it is clear that the rotor out-of-plane loads – that is, the moment about the horizontal axis, $M_Y(H, t)$, the hub moment about the vertical axis, $M_Z(H, t)$, and the rotor thrust, $F_X(H, t)$ – will all be statistically independent of each other. The same will apply to the rotor in-plane loads – the rotor torque, $M_X(H, t)$, and the sideways load, $F_Y(H, t)$. However, as the out-of-plane and in-plane loads on a blade element are both assumed to be proportional to the local wind speed fluctuation, u , it follows that the rotor torque fluctuations will be in phase with the rotor thrust fluctuations, and the rotor sideways load fluctuations will be in phase with the fluctuations of the hub moment about the horizontal axis, $M_Y(H, t)$.

The above relationships have implications for the combination of fatigue loads. Clearly the power spectrum of the fore-aft tower moment at height z , $S_{M_Y}(z, n)$, can be obtained by simply adding the power spectrum of the hub moment about the horizontal axis to $(H - z)^2$ times the power spectrum of the rotor thrust. Similarly the power spectrum of the side-to-side tower moment at height z , $S_{M_X}(z, n)$, can be obtained by adding the power spectrum of the rotor torque to $(H - z)^2$ times the power spectrum of the rotor sideways load.

Having obtained power spectra for the M_X , M_Y and M_Z moments at height z , the corresponding fatigue load spectra can be derived with reasonable accuracy by means of the Dirlik method described in Section 5.9.3. As the tower stress ranges will be enhanced by tower resonance, the input power spectra should incorporate dynamic magnification, as outlined in Section 5.12.5. Ragan and Manuel (2007) compared fatigue loads calculated in the frequency and time domains (see Section 5.9.3) and concluded that the Dirlik method performed very well in estimating tower fatigue bending moments for the case investigated.

Fatigue stress ranges due to bending about the two axes can easily be calculated separately from the $M_X(z)$ and $M_Y(z)$ fatigue spectra, but the stress ranges due to the two fatigue spectra combined cannot be calculated precisely because of lack of information about phase relationships. However, as noted above, the $M_X(H)$ component of the $M_X(z)$ fluctuations is in phase with the $F_X(H)$ component of the $M_Y(z)$ fluctuations, and the $F_Y(H)$ component of the $M_X(z)$ fluctuations is in phase with the $M_Y(H)$ component of the $M_Y(z)$ fluctuations so the stress ranges due to the $M_X(z)$ and $M_Y(z)$ fatigue spectra combined can be conservatively calculated as if they were in phase too. Theoretically this means pairing the largest $M_X(z)$ and $M_Y(z)$ loading cycles, the second largest, the third largest and so on, right through the

Table 5.8 Example M_X and M_Y fatigue spectra

ΔM_Y (KNm)	No. of ΔM_Y cycles	ΔM_X (KNm)	No. of ΔM_X cycles
200–300	5	100–150	10
100–200	15	50–100	40
0–100	80	0–50	50

fatigue spectra, and calculating the stress range resulting from each pairing. In practice, of course, the $M_X(z)$ and $M_Y(z)$ load cycles are distributed between two sets of equal size ‘bins’, so they have to be reallocated to bins in a two-dimensional matrix of descending load ranges, as shown in the grossly simplified example given in Tables 5.8 and 5.9 below.

For a circular tower, the stress ranges would have to be computed at several points around the circumference in order to identify the location (with respect to the nacelle axis) where the fatigue damage was maximum.

A simpler but potentially cruder approach to the combination of the two fatigue spectra is to use the ‘Damage Equivalent Load’ method. This involves the calculation of constant amplitude fatigue loadings, $M_{X.Del}$ and $M_{Y.Del}$, of, say 10^7 cycles each, that would respectively produce the same fatigue damages as the M_X and M_Y spectra, using the S - N curve appropriate to the fatigue detail under consideration. If the M_X and M_Y fluctuations are treated as being in-phase as before, the combined ‘Damage Equivalent Load’ moment is, $\sqrt{M_{X.Del}^2 + M_{Y.Del}^2}$.

5.13 Wind turbine dynamic analysis codes

A large modern turbine is a complex structure. Relatively sophisticated methods are required in order to predict the detailed performance and loading of a wind turbine. These methods should take into account:

- the aerodynamics of the rotating blade, including induced flows (i.e. the modification of the flow field caused by the turbine itself), three-dimensional flow effects and dynamic stall effects when appropriate;
- structural analysis of the blades, drive train and tower, allowing their vibrational dynamics to be modelled;

Table 5.9 Joint M_X and M_Y cycle distribution

ΔM_X (KNm)	ΔM_Y (KNm)			Total No. of M_X cycles
	200–300	100–200	0–100	
100–150	5	5		10
50–100		10	30	40
0–50			50	50
Total No. of M_Y cycles	5	15	80	

- aeroelastic feedback, i.e. the modification of the aerodynamic forces due to the vibrational velocities of the structure;
- dynamic response of subsystems such as the generator, yaw system and blade pitch control system;
- control algorithms used during normal operation, start-up and shut-down of the turbine; and
- temporal and spatial variations of the wind field impinging on the turbine, including the three-dimensional structure of the turbulence itself.

For offshore wind turbines, this should be extended to include:

- hydrodynamic forces on the submerged structure; and
- hydroelastic feedback, i.e. the modification of the hydrodynamic forces due to the vibrational velocities of the structure.

Starting from a wind turbulence spectrum, it is possible to develop techniques in the frequency domain which account for many of these aspects, including rotational sampling of the turbulence by the blades, the response of the structure, and the control system. These techniques are set out in Sections 5.7.5, 5.8.6, 5.12.4 and elsewhere. However, although frequency domain methods are elegant and computationally efficient, they can only be applied to linear time-invariant systems and, therefore, cannot deal with some important aspects of wind turbine behaviour, such as:

- stall aerodynamics and hysteresis;
- non-linearities in subsystems such as bearing friction and pitch rate limits;
- nonlinear aspects of control algorithms;
- variable speed operation; and
- start-up and shut-down.

As a result, time-domain methods almost exclusively are now used for wind turbine design calculations. The ready availability of computing power means that the greater computational efficiency of frequency domain methods is no longer such an important consideration.

A number of codes are available commercially for the calculation of wind turbine performance and loads using time-domain simulations. These simulations use numerical techniques to integrate the equations of motion over time, by subdividing the time into short timesteps, as described in Section 5.8.5. In this way, all the non-linearities and non-stationary aspects of the system, such as those listed above, can be dealt with to any desired level of accuracy. A useful early comparative survey of such codes was given by Molenaar and Dijkstra (1999).

As explained in Section 5.8.5, there are a number of different algorithms or solvers for integrating the equations of motion. Some use a fixed timestep h (which has to be short enough to account for all modal frequencies which are considered important), while others use a variable timestep which is continually adjusted during the simulation, keeping it as long as possible to maximise simulation speed while still keeping all the integrated states within a certain error tolerance.

The use of variable timestep methods also allows accurate modelling of discontinuities, because close to a discontinuity the timestep can be adjusted to find the exact moment when the characteristics of the system change. Discontinuities can occur for many reasons: for example stick-slip friction (of pitch and yaw bearings, shaft brake, slipping clutch etc.), grid loss, faults, controller or safety system actions etc. Note that the equations of motion and structural resonant frequencies change at the moment when a friction element like a brake changes from slipping to sticking.

On the other hand these codes can provide a valuable way to test turbine controllers, linking the real controller to the simulation model which acts as a 'virtual turbine', and in this case a fixed timestep may be more appropriate, to allow regular communication with the controller running in real time, and to ensure that the calculations for each timestep are completed within that real time interval. Necessarily this may mean a loss of accuracy in predicting the effect of higher frequency modes and discontinuities. As an example, the *Bladed* code mentioned below normally uses a variable timestep, but also provides a fixed step option for real-time applications such as controller testing.

Two principal approaches to the modelling of structural dynamics are embodied in time-domain simulation packages. Some use a full finite-element representation of the structure, which is broken down into small elements. The equations of motion are solved for each element, with boundary conditions matched at the interfaces between elements. An example of such a code is Adams-WT (Hansen, 1998), which consists of a general purpose finite-element code (Adams) interfaced to an aerodynamic module.

The other main approach is the modal analysis method as described in Section 5.8.1, in which simple finite-element methods are used to predict just the first few modes of vibration of the main components, such as the rotor blades and the tower. These are typically modelled as beam elements, but it is important to include geometric stiffening effects so that, for example, the centrifugal stiffening of the blades is taken into account – that is, the increase in apparent stiffness with rotational speed due to the effect of centrifugal force on the element mass. Additional degrees of freedom are added as required, for example for the drive train rotation and torsion, pitch and yaw motion etc. The equations of motion are then derived for the entire coupled system. Traditionally this can be done by constructing the Lagrangian for the system including all degrees of freedom. With full rotation of the yaw bearing on top of a flexible tower, and then full rotation of the rotor (also flexible) about the shaft axis, the co-ordinate transformations involved mean that the equations rapidly become very large, usually requiring some form of symbolic processing to derive them in an automated way. More recently, methods based on the approach of Multi-body Dynamics (see for example Shabana, 1998) have been used, which provides a generalised way to link together the separate equations of motion of each rigid or flexible component by means of defined linking elements, which can include rigid links, revolute hinges, sliding joints etc. This provides a very powerful technique which is readily extended to structures of arbitrary complexity. Using a technique originally proposed by Craig and Bampton (1968), the mode shapes of each modal component can be defined in a way which is independent of any other component to which it is attached.

One example of a widely-used commercial code based on the component mode approach is *Bladed* (Garrad Hassan, 2010). Originally built using a Lagrangian approach, this code has recently been converted to use a multi-body approach. Beam element models for the blades and tower are combined with elements representing other components of the transmission system, the yaw and pitch actuators etc. The control system, which has a major influence on the performance as well as the loads, can be modelled in full detail. The code can also model the

electrical generator and power converter, allowing detailed calculations of turbine response to network faults, generator short-circuit faults etc. in a fully integrated way. Interfaces are provided to link in more detailed models of subcomponents such as gearboxes. For most calculations this level of detail is not required, but where certain load cases are critical for the particular component it may be useful to be able to run a more detailed model.

By using a limited number of modes, the modal approach results in rapid calculations, so that a complete set of design or certification load cases, typically amounting to several hundred load cases each consisting of a ten-minute simulation, can be run in a few hours on a standard desktop computer. A small number of modes is generally adequate for predicting the loads: the higher frequency modes generally have little effect. However, to model the deflections accurately it would be necessary to model more modes, because the modelled deflection is a linear combination of the mode shapes used, and a small number of mode shapes may not be sufficient to model the actual deflected shape. Rather than using more modes, the static improvement technique can be used (Barltrop and Adams, 1991): effectively the calculated loading is used as if it were a static load, and combined with the stiffness matrix to recalculate the deflections.

The modal approach generally assumes that all deflections of the flexible bodies remain small. More complex non-linear beam-element models are being developed to help increase accuracy in case of larger deflections, such as may be found with very flexible blades. Alternatively, the non-linearity can be captured by modelling the blade as a number of shorter linear beam elements joined end to end.

For the aerodynamics, all these codes generally use BEM (blade element momentum theory) as described in Chapter 3, as this is currently the only way to achieve rapid enough simulations for the standard sets of calculations which are normally needed. More advanced aerodynamic methods such as vortex wake and panel methods are starting to be used to examine specific cases where BEM is not sufficiently accurate. This might include highly yawed flow, tip vanes, ducted rotors or the need to understand the aerodynamic interaction between the blades and the nacelle, for instance. Ultimately CFD (computerised fluid dynamics) methods based on direct solution of the Navier-Stokes equation could be used, and some general commercial CFD codes are now available, but these methods are still far too slow and cumbersome to be useful except perhaps to examine very special cases in detail.

It is important to be able to simulate three-dimensional turbulent wind fields because the dynamic wind speed variations across the rotor are of major importance in determining the loads. The Veers method (Veers, 1988) as described in Section 5.7.6 is a convenient way to do this: a random number sequence is filtered using a representation of the spectrum and spatial coherence of the turbulence to generate a three-dimensional wind field which is consistent with the chosen spectral model. This method is used by the codes mentioned above. *Bladed* also incorporates a different technique, due to Mann (1998), which generates a turbulence field by means of a three-dimensional inverse fast Fourier transform of the three-dimensional wavenumber spectrum (see Chapter 2). For offshore turbines, *Bladed* uses a related method to generate stochastic wave time histories impacting on the submerged part of the structure, in addition to the loading from water currents. As with the effect of aerodynamics, the effect of the vibrational velocities of the structure on the hydrodynamic forces is significant. This leads to considerable interactions between the wind and wave loading. Jamieson *et al.* (2000) have demonstrated that if wind and wave loading are treated in isolation from each other, an over-conservative design is likely to result.

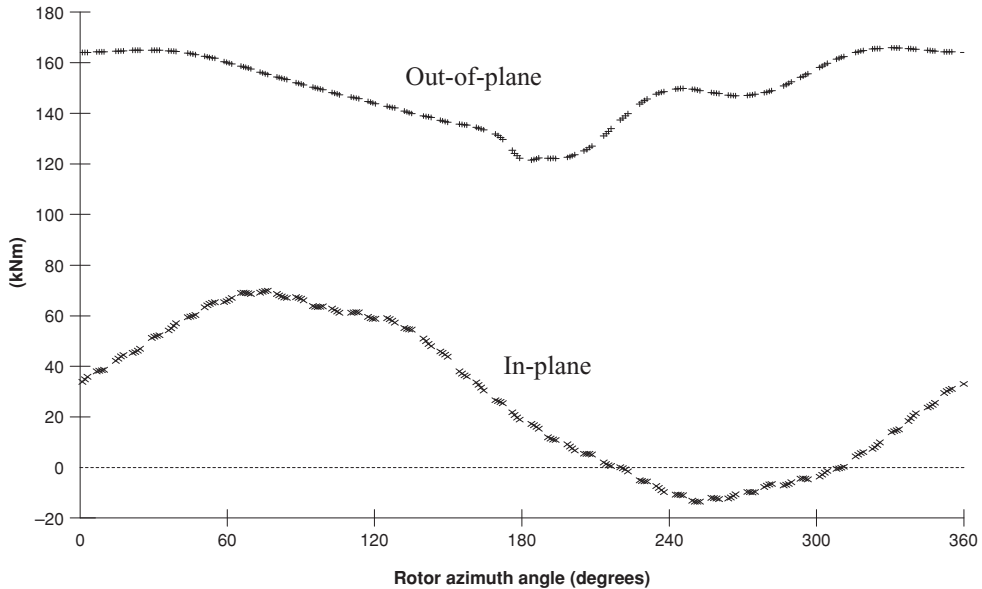


Figure 5.42 Blade root bending moment in steady wind

The use of sophisticated calculation methods such as those described above are now mandatory for the certification of wind turbines, particularly at the larger sizes. A few illustrative examples of results obtained with *Bladed* are described below.

Figure 5.42 shows a *Bladed* simulation of the in- and out-of-plane bending moments at the root of one of the blades, during operation in steady, sheared wind. The in-plane moment is almost a sinusoidal function of azimuth, being dominated by the gravity loading due to the self-weight of the blade which, relative to the blade, changes direction once per revolution. The mean is offset from zero because of the mean positive aerodynamic torque developed by the blade. There is a slight distortion of the sinusoid, partly because of the variation of aerodynamic torque due to wind shear and the effect of tower shadow, and partly because of the effect of structural vibrations.

The out-of-plane moment is always positive, the mean value being dominated by the aerodynamic thrust on the blade. There is a systematic variation with azimuth resulting from the wind shear, giving a lower load at 180° azimuth (bottom dead centre) than at 0°. A sharp dip at 180° is also visible, and this is the effect of the tower shadow (the reduction in wind speed in the vicinity of the tower). The blade out-of-plane vibrational dynamics contribute a significant higher-frequency variation.

In turbulent wind, the loads take on a much more random appearance, as shown in Figure 5.43. The out-of-plane load in particular is varying with wind speed and, as this is a pitch-controlled machine, with pitch angle. The in-plane load is more regular, as it is always dominated by the reversing gravity load.

Spectral analysis provides a useful means of understanding these variations. Figure 5.44 shows auto-spectra of the blade root out-of-plane bending moment and the hub thrust force. The out-of-plane bending moment is dominated by peaks at all multiples of the rotational frequency of 0.8 Hz. These are caused mainly by the rotational sampling of turbulence by

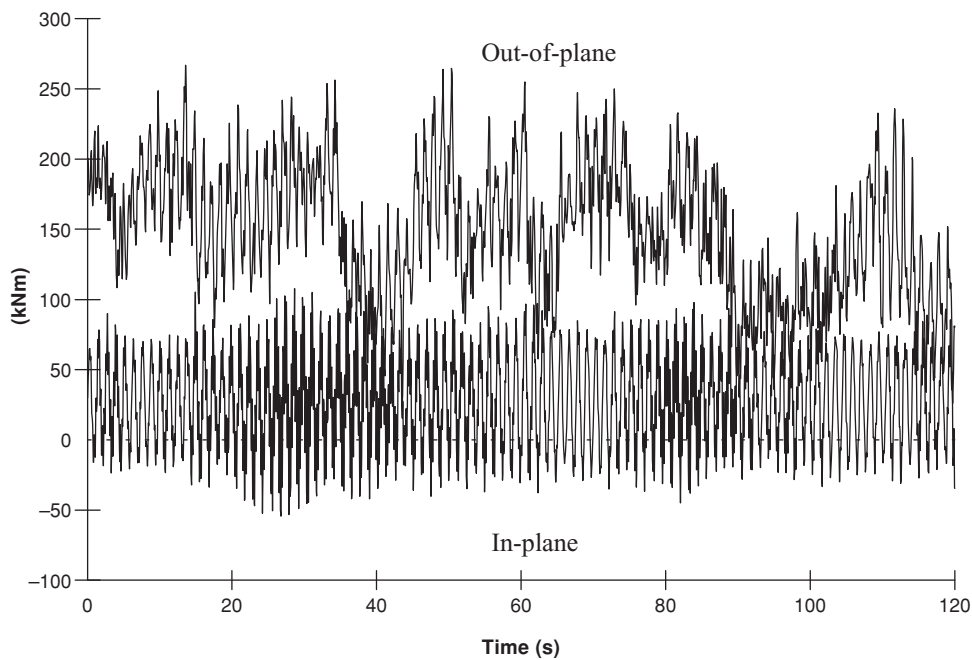


Figure 5.43 Blade root bending moment in turbulent wind

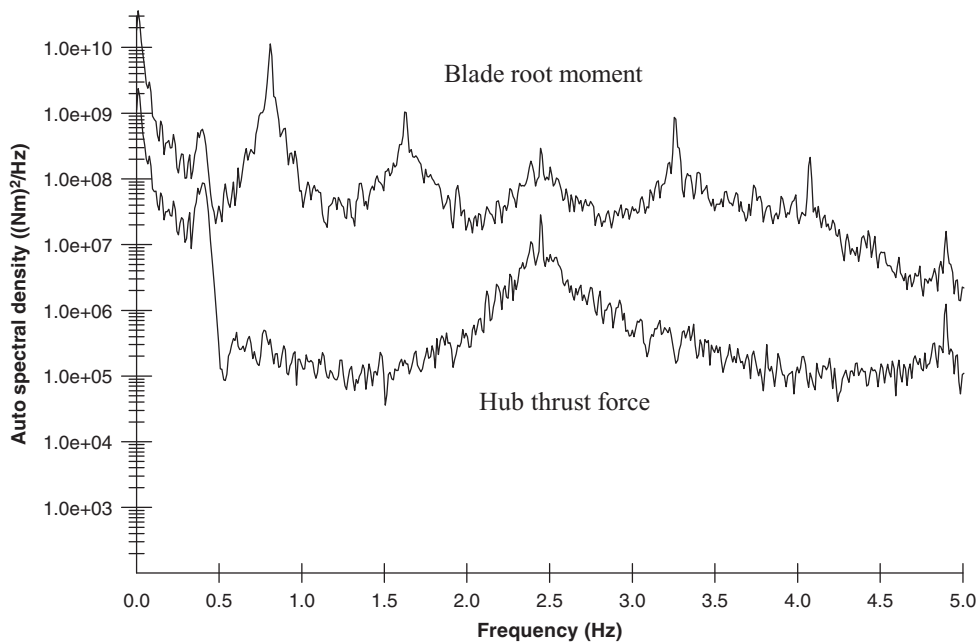


Figure 5.44 Spectra of out-of-plane loads in turbulent wind

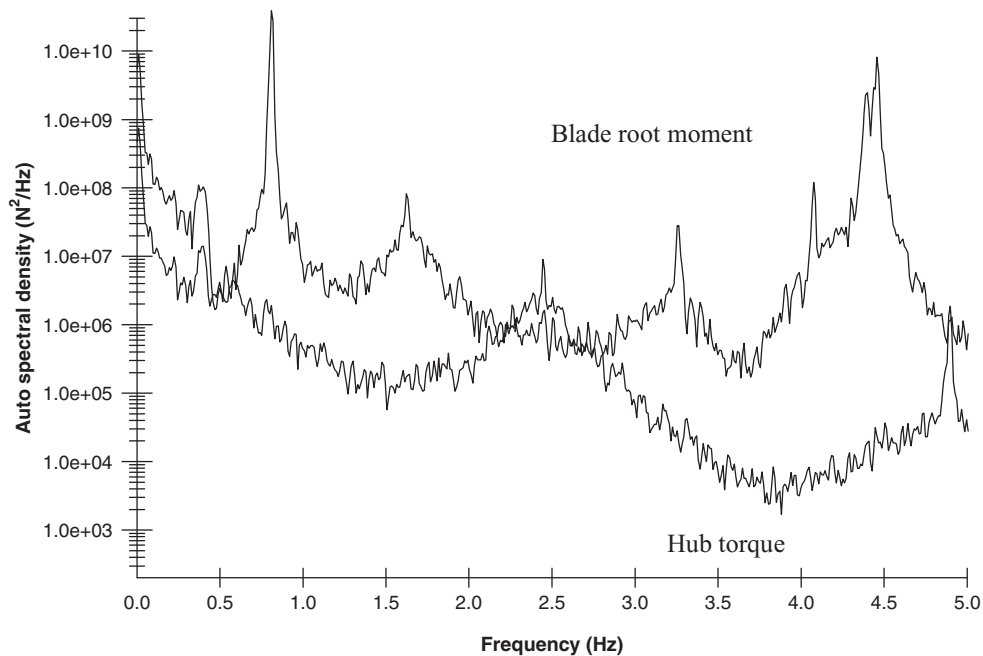


Figure 5.45 Spectra of in-plane loads in turbulent wind

the blade as it sweeps around, repeatedly passing through turbulent eddies. Wind shear and tower shadow also contribute to these peaks. A small peak due to the first out-of-plane mode of vibration at about 3.7 Hz is just visible. There is also a significant effect of the first tower fore-aft mode of vibration at about 0.4 Hz.

This tower effect is also visible in the spectrum of the hub thrust force. However, this force is the sum of the shear forces at the roots of the three blades. These forces are 120° out of phase with each other, with the result that the peak at the rotational frequency (1P) is eliminated; as are the peaks at multiples of this frequency such as 2P, 4P etc. Only the peaks at multiples of 3P remain, since at these frequencies the three blades act in phase with each other.

This effect is even more significant in the in-plane load spectra (Figure 5.45). Of the blade load peaks at multiples of 1P, only the relatively small peaks at 3P and 6P come through to the hub torque. The 1P peak in the blade load, which is dominated by gravity, is particularly large, but it is completely eliminated from the hub torque. The tower peak at 0.4 Hz is visible in both loads. A large blade load peak at the first in-plane blade vibrational mode at 4.4 Hz is also seen, but this is a mode which does not include any rotation at the hub, and consequently is not seen in the hub torque. Some higher frequency blade modes (not shown) will be coupled with hub rotation.

5.14 Extrapolation of extreme loads from simulations

In the case of Load Case 1.1 (normal operation in a turbulent wind) IEC 61400–1 Edition 3 requires the characteristic blade root out-of-plane and in-plane bending moments and the

characteristic tip deflection to be determined by statistical extrapolation of the extreme values of the load time series output from the simulations. For simulations of ten minutes duration, the characteristic value is defined as that with a 3.8×10^{-7} probability of exceedance – that is, the load with a return period of 50 years. This section considers ways in which the load exceedance probability distribution can be derived from the simulations and extrapolated to the 3.8×10^{-7} value. As it is more convenient to work with load *non*-exceedance probability distributions – otherwise known as cumulative distribution functions – the discussion below is in these terms, using the notation $P(X \leq x) = F(x)$.

There are two sequences that can be followed in order to assemble a single load probability distribution from simulation data from different wind speed bins:

- Derivation of load non-exceedance probability distributions for each wind speed bin followed by combination of these distributions in proportion to the operating time in each bin ('Fitting before aggregation').
- Aggregation of results from all wind speed bins, with the number of simulations per bin proportional to the hours of operation in each bin, followed by the derivation of a single load non-exceedance probability distribution ('Aggregation before fitting').

It is also necessary to decide how many extreme values from each simulation are to be utilised. In the 'global extremes' method, only the largest extreme value in each ten-minute simulation – that is, the global extreme – is used in the construction of the load non-exceedance probability distribution, but in the 'local extremes' method all extreme values that can be considered independent are utilised. The 'local extremes' method would appear to be more attractive, because it uses much more of the available data, but it throws up the problem of establishing a criterion for independence.

The various stages of the 'Fitting before aggregation' sequence using the 'global extremes' method are considered first.

5.14.1 Derivation of empirical cumulative distribution function of global extremes

Each ten-minute time series will yield a maximum value of the load under investigation and, for n ten-minute simulations at a particular wind speed, there will be n such global extremes, which can be ranked $1, 2, \dots, i, \dots, n_k$ from smallest to largest. An empirical non-exceedance probability distribution for the ten-minute extreme load, x_k , at wind speed U_k can then be constructed as

$$F(x_{ki} | U_k) = \frac{i}{n_k + 1}, \quad i = 1, 2, \dots, n_k \quad (5.137)$$

Harris (1996) has shown that, if $F(x)$ is a known function of x , the mean of the L non-exceedance probabilities $F(x_i)$ for the i th extremes from L sets of n ten-minute simulations, derived from the x_i simulation results, tends to the value $i/(n + 1)$ for L large.

5.14.2 Fitting an extreme value distribution to the empirical distribution

There are several extreme value distributions which can be fitted to the empirical probability distributions obtained from the simulations. These include the Gumbel distribution (also known as the Fisher-Tippett I distribution), which was introduced in the context of extreme wind speeds in Section 2.8, the log-normal distribution, the three parameter Weibull distribution and the generalised extreme value distribution, which are described in turn below.

a) Gumbel distribution

The probability that the variable X will not exceed the value x is given by

$$P(X \leq x) = F(x) = \exp \left[-\exp \left(-\frac{x - x_o}{c} \right) \right] \quad (5.138)$$

where x_o is the most likely extreme value or the mode of distribution, c is the dispersion and $y = (x - x_o)/c$ is termed the reduced variate.

This relationship is a straight line if $y = -\ln[-\ln(F(x))]$ is plotted against x , so a Gumbel distribution can be fitted to the empirical distribution by the method of least squares and the parameters x_o and c thereby determined. However, Harris (1996) has pointed out two defects of the classical least squares method.

First of all, the mean of the function which is plotted, $-\ln[-\ln(F(x_i))]$, is not the same as the double natural logarithm of the mean of $F(x_i)$ itself, given by Equation 5.137. Harris provides a formula by which the mean of $-\ln[-\ln(F(x_i))]$, that is, $\bar{y}_i$, may be evaluated when the data should conform to a Gumbel distribution – as follows:

$$\bar{y}_v = \frac{N!}{(v-1)!(N-v)!} \int_0^1 -\ln[-\ln(z)] z^{N-v} (1-z)^{v-1} dz \quad (5.139)$$

In this formula, N is the number of data points ($= n_k$), v is the rank of data points with the *largest* first, so that $v = (N+1) - i$ and $z = F(x_y)$.

Secondly, the classical least squares method assumes that the variability of each plotted ordinate is of similar magnitude, whereas, for extreme value data, the variability of the reduced variate, y , is much greater for the largest values than for the others. Accordingly, Harris proposes weighting the data points in inverse proportion to the variance of the y values, before the least squares fitting is carried out.

Values of $\bar{y}_v$ and its standard deviation are given in Table 5.10 for the case of $N = 15$, consistent with the IEC 61400–1 Edition 3 requirement for at least 15 ten-minute simulations for wind speeds above rated. Values of y calculated by the standard Gumbel method are also included for comparison and it is seen that the differences are significant for the largest of the extremes. The Harris weighting factor is included in the table in the last column. It is seen that, in general, the Harris method will result in a steeper straight line which is less influenced by the largest extremes.

An alternative method of fitting a straight line to the $-\ln[-\ln(F(x))]$ plot is the *method of statistical moments* (Moriarty *et al.*, 2004), in which the first two statistical moments of the data – defined as the mean and the variance respectively – are equated to analytical expressions

Table 5.10 Table of mean values of the reduced variate, $y_y = -\ln[-\ln(F(x_y))]$, its standard deviation and the Harris weighting factor, w_y

Rank, ν (Largest first)	Rank, i (Smallest first)	$y_y =$ $-\ln[-\ln((N+1-\nu)/(N+1))]$ $= -\ln[-\ln(i/(N+1))]$ (Gumbel method)	$\bar{y}_\nu$ - Equation 5.139 (Harris method)	σ_y - Standard Deviation of $\bar{y}_\nu$	Harris Weighting factor, w_y
1	15	2.7405	3.2853	1.2825	0.0064
2	14	2.0134	2.2504	0.8031	0.0164
3	13	1.572	1.7133	0.6291	0.0266
4	12	1.2459	1.3404	0.5341	0.037
5	11	0.9816	1.0478	0.4726	0.0472
6	10	0.755	0.8019	0.4291	0.0573
7	9	0.5528	0.5852	0.3965	0.0671
8	8	0.3665	0.3873	0.3714	0.0765
9	7	0.1903	0.201	0.3518	0.0852
10	6	0.0194	0.0206	0.3366	0.0931
11	5	-0.1511	-0.1595	0.3254	0.0996
12	4	-0.3266	-0.3458	0.3184	0.104
13	3	-0.5152	-0.5485	0.317	0.1049
14	2	-0.7321	-0.7884	0.3257	0.0994
15	1	-1.0198	-1.1326	0.3648	0.0793

for these moments. Thus, for the standard Gumbel distribution, $F(y) = \exp[-\exp(-y)]$, the probability density function is $f(y) = dF(y)/dy = \exp[-y - \exp(-y)]$ and the mean is

$$\mu_y = \int_{-\infty}^{\infty} y \cdot f(y) dy = \int_{-\infty}^{\infty} y \cdot \exp[-y - \exp(-y)] dy = \gamma = 0.5772 \quad (5.140)$$

Similarly the variance is

$$\sigma_y^2 = \int_{-\infty}^{\infty} (y - \mu_y)^2 \cdot f(y) dy = \int_{-\infty}^{\infty} (y - \mu_y)^2 \cdot \exp[-y - \exp(-y)] dy = \frac{\pi^2}{6} \quad (5.141)$$

Denoting the mean and the standard deviation of the dataset of extreme values as μ_x and σ_x respectively, and noting that $y = (x - x_o)/c$, we obtain $\mu_y = (\mu_x - x_o)/c = 0.5772$ and $\sigma_y = \sigma_x/c = 1.2825$.

Hence

$$c = \sigma_x/1.2825 \quad \text{and} \quad x_o = \mu_x - c\mu_y = \mu_x - 0.5772 \frac{\sigma_x}{1.2825} = \mu_x - 0.450\sigma_x.$$

Figure 5.46 compares the three methods described above for fitting a straight line to empirical data on a Gumbel plot, for a dataset consisting of 15 global extremes of

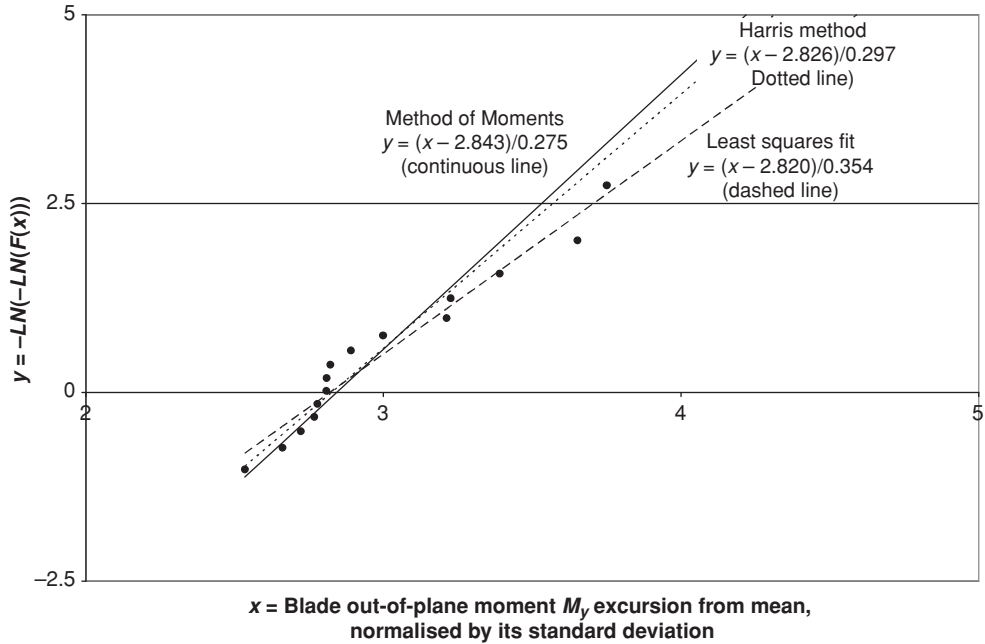


Figure 5.46 Comparison of techniques for fitting a straight line to empirical data on a Gumbel plot

out-of-plane blade root bending moment, taken from 15 ten-minute simulations of operation in a 12 m/s wind speed. Note that the straight lines derived using the Harris method and the method of moments appear less influenced by the largest values than that based on the least squares method.

b) Log-normal distribution

In the log-normal distribution, the *logarithm* of the variable is normally distributed, so the probability distribution function of the variable x is

$$f(x) = \frac{1}{\sqrt{2\pi}\sigma_z x} \exp \left[-\frac{1}{2} \left(\frac{\ln(x) - \mu_z}{\sigma_z} \right)^2 \right] \quad (5.142)$$

where μ_z and σ_z are the mean and standard deviation respectively of $z = \ln(x)$. The mean and standard deviation of the variable x itself, μ_x and σ_x , are given in terms of μ_z and σ_z as follows:

$$\mu_x = \exp \left[\mu_z + \sigma_z^2/2 \right] \quad \sigma_x = \exp \left[\mu_z + \sigma_z^2/2 \right] \sqrt{\exp(\sigma_z^2) - 1} = \mu_x \sqrt{\exp(\sigma_z^2) - 1} \quad (5.143)$$

The log-normal distribution parameters, μ_z and σ_z , can again be fitted to the extreme value data by the method of statistical moments, using

$$\sigma_z = \sqrt{\ln(1 + (\sigma_x/\mu_x)^2)} \quad \text{and} \quad \mu_z = \ln(\mu_x) - \sigma_z^2/2 \quad (5.144)$$

derived from Equation 5.143.

c) Three parameter Weibull distribution

The three parameter Weibull distribution is defined as $P(Y \leq y) = F(y) = 1 - \exp[-y^\alpha]$, where $y = (x - x_o)/c$ as before, and $F(y)$ only applies to positive values of y . Hence the probability density function, $f(y)$, is given by $\alpha y^{\alpha-1} \exp[-y^\alpha]$. The three parameters, x_o , c and α can be fitted to the data by the method of statistical moments described above, but in this case using the first three moments instead of the first two.

Using the shorthand $G_m(\alpha) = (m/\alpha)\Gamma(m/\alpha)$ where Γ is the Gamma function, the first three statistical moments are as follows:

$$\begin{aligned} \mu_y &= G_1(\alpha), \quad \sigma_y^2 = G_2(\alpha) - G_1^2(\alpha), \\ \eta_y \sigma_y^3 &= \int_{-\infty}^{\infty} (y - \mu_y)^3 f(y) dy = G_3(\alpha) - 3G_2(\alpha)G_1(\alpha) + 2G_1^3(\alpha) \end{aligned} \quad (5.145)$$

Note that η_y , the skewness parameter, is the third statistical moment normalised by the cube of the standard deviation.

d) Generalised extreme value (GEV) distribution

The generalised extreme value distribution was introduced by Jenkinson in 1955 and is more versatile than the Gumbel distribution, as it allows the skewness of extreme value distributions to be modelled as well as their spread. It is defined as $P(Y \leq y) = F(y) = \exp[-\{1 - ky\}^{1/k}]$, where $y = (x - x_o)/c$ as before, and k is the shape parameter, which determines the curvature of the distribution when it is plotted as $-\ln[-\ln(F(x))]$ against x . If k is positive, the curve is concave upwards and has an upper bound of $x_o + c/k$, whereas if k is negative, the curve is concave downwards and has a lower bound of $x_o + c/k$.

Hosking, Wallis and Wood (1985) advocate the use of the method of probability weighted moments (PWMs) for fitting a GEV distribution to empirical data. For a probability distribution $F = F(x)$, with an inverse distribution function $x(F)$, probability weighted moments β_0 , β_1 and β_2 are defined by $\beta_r = \int_0^1 x(F) F^r dF$, and the parameters x_o , c and k of the GEV are determined by equating the PWMs of the empirical data to the PWM's of the GEV.

The PWM's of the GEV are given by the formula

$$\beta_r = \frac{1}{r+1} \left[x_o + \frac{c}{k} \left\{ 1 - \frac{\Gamma(1+k)}{(r+1)^k} \right\} \right] \quad (5.152)$$

for $k > -1$, where Γ is the Gamma function. This gives:

$$\begin{aligned} \beta_0 &= x_o + \frac{c}{k} \{1 - \Gamma(1+k)\}, \quad 2\beta_0 - \beta_1 = \frac{c}{k} \Gamma(1+k) (1 - 2^{-k}) \quad \text{and} \\ \frac{3\beta_2 - \beta_0}{2\beta_1 - \beta_0} &= \frac{1 - 3^{-k}}{1 - 2^{-k}} \end{aligned} \quad (5.146)$$

Solution of the last equation requires iterative methods, but Hosking *et al.* (1985) have shown that a good approximation for k is $7.8590C + 2.9554C^2$, where $C = (2\beta_1 - \beta_0)/(3\beta_2 - \beta_0) - (\log 2/\log 3)$. This value of k can then be substituted in the first two Equations 5.146 to obtain x_0 and c .

The PWM's of the empirical data can be estimated from the formula:

$$\beta_r[p_{i,n}] = \frac{1}{n} \sum_{i=1}^n p_{i,n}^r x_i \quad (5.147)$$

where $p_{i,n}$ is the probability assigned to the i th global extreme, with ranking from smallest to largest. Hosking *et al.* propose two alternative expressions for use in estimating $p_{i,n}$ as follows:

$$p_{i,n} = (i - a)/n, \quad 0 < a < 0 \quad p_{i,n} = (i - a)/(n + 1 - 2a), \quad -0.5 < a < 0.5$$

Another approach is to estimate the PWM's directly from the unbiased estimators b_r given by:

$$b_r = \frac{1}{n} \sum_{i=1}^n \frac{(i-1)(i-2)\cdots(i-r)}{(n-1)(n-2)\cdots(n-r)} x_i \quad (5.148)$$

Figure 5.47 compares two GEV distributions fitted to the dataset of 15 global extremes described above and extrapolated to a 50-year return period. In one case unbiased estimators of the PWM's are used for the fitting and, in the other, the formula $p_{i,n} = (i - a)/n$ with

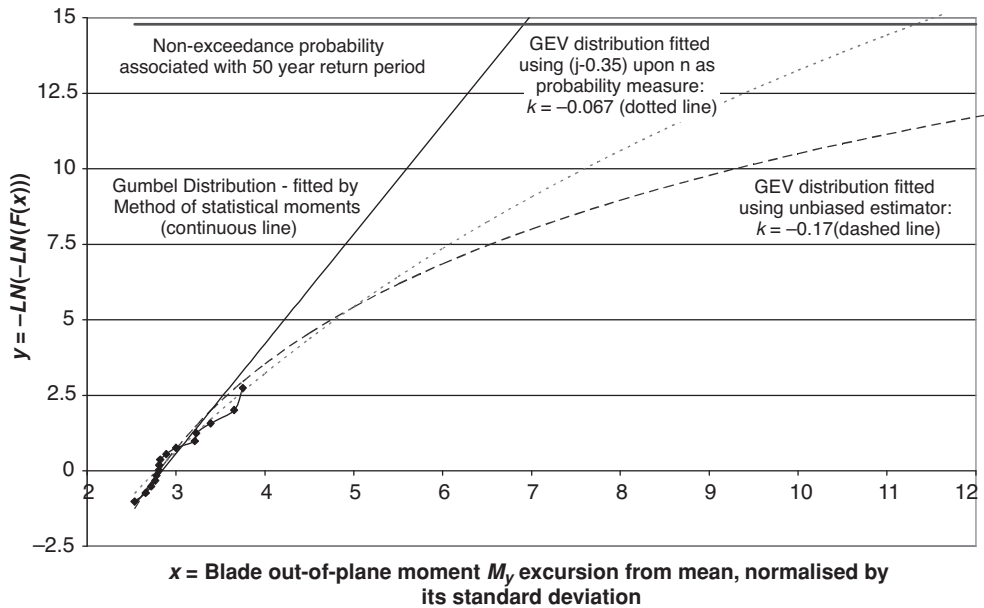


Figure 5.47 Comparison of GEV distributions fitted to empirical data on a Gumbel plot

$a = 0.35$ was employed for the probabilities, as this was found to give the best overall results in a computer simulation by Hosking *et al.* Also shown for comparison is the Gumbel distribution fitted to the dataset using the method of statistical moments.

It is seen that there is poor agreement between the two GEV distributions. This is likely to be due – at least in part – to the small size of the dataset, which results in coarse estimates of the PWMs. It should also be noted that the use of different datasets for the same load case results in considerable variation in the shape parameter, suggesting that a dataset of 15 extreme values is not large enough to yield a meaningful result.

5.14.3 Comparison of extreme value distributions

Gumbel, 3 parameter Weibull and log-normal distributions have been fitted to the dataset of 15 global extremes of out-of-plane blade root bending moment introduced above, using the method of statistical moments in each case, and compared on a Gumbel plot of $-\ln[-\ln(F(x))]$ against the normalised bending moment excursion, x , in Figure 5.48.

Figure 5.48 shows the three extreme value distribution extrapolated to the 50-year return period exceedance probability of 3.8×10^{-7} (for which, $-\ln[-\ln(F(x))]$ is 14.78) to illustrate their behaviour. It is seen that there is a wide variation in the predicted value of the 50-year return load.

Although the loadings on wind turbine components are generally not narrow-banded, it is instructive to investigate how well the above three extreme value distributions and the GEV distribution can be fitted to the distribution of global extremes arising from a narrow banded process. Consider a blade rotating at 15 rpm, which experiences 150 loading cycles in ten

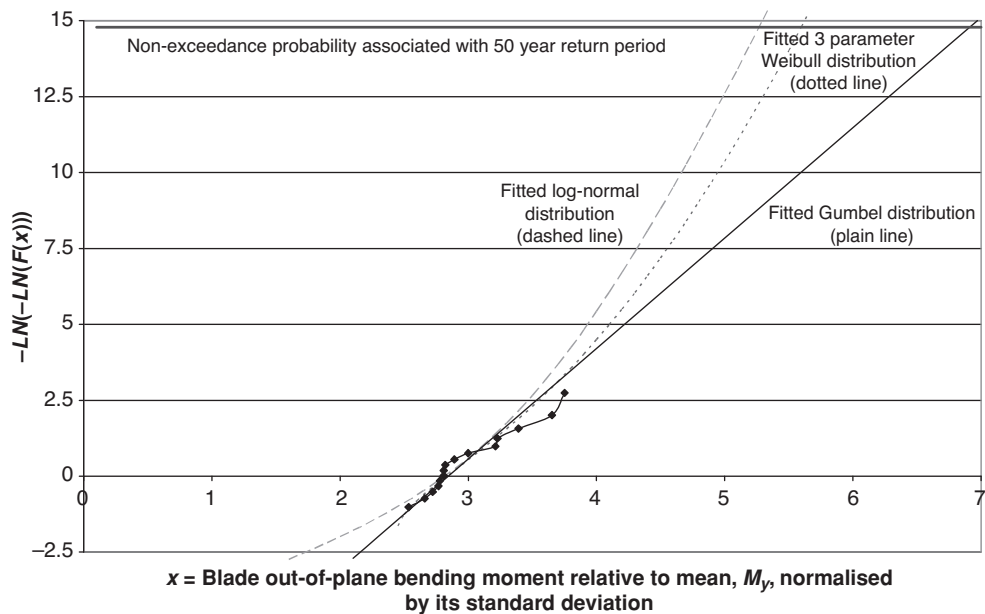


Figure 5.48 Gumbel plot comparison of three extreme value distributions fitted to empirical data

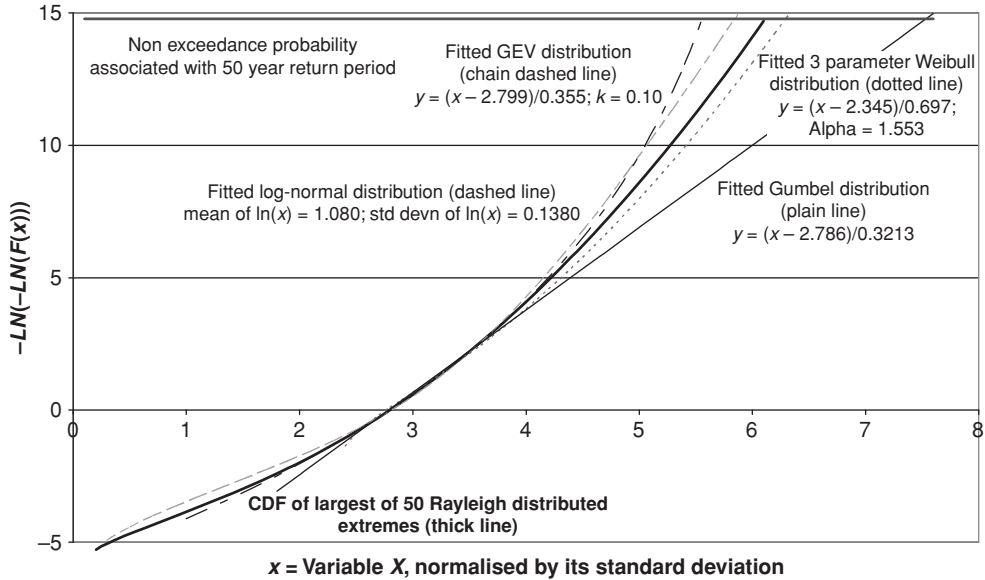


Figure 5.49 Gumbel plot comparison of four extreme value distributions fitted to the distribution of the largest of 50 Rayleigh distributed extremes

minutes. Assuming that the extreme loads in adjacent three-cycle blocks are independent of one another, there are 50 independent maxima in the ten-minute time interval, and the cumulative probability distribution of the global maximum is

$$F(x) = \left\{ 1 - \exp \left[-\frac{x^2}{2} \right] \right\}^{50} \quad (5.149)$$

Gumbel, 3 parameter Weibull, log-normal and GEV distributions have been fitted to this distribution using the method of statistical moments, and Gumbel plots of the fitted distributions are presented on Figure 5.49, together with their defining parameters.

It is seen that the log-normal and the 3 parameter Weibull distributions fit the distribution of global maxima of a narrow banded process better than the other two.

Freudenreich and Argyriadis (2008) compared extrapolations made using the above four extreme value distributions by reference to five one-year simulations for a 5 MW pitch-regulated turbine. Using the extreme values from 30 ten-minute time histories per wind speed bin as input, they concluded that the extrapolations of blade flapwise root bending moment made with the 3 parameter Weibull and log-normal distributions were the most accurate, whereas the Gumbel extrapolation was significantly conservative.

5.14.4 Combination of probability distributions

The procedure outlined above yields a family of load non-exceedance probability distributions conditional upon wind speed for the extreme load in the simulation period, $F(x|U_k)$ – one for each wind speed bin. These are combined to yield a single distribution for all operating wind

speeds by weighting each one according to the number of hours of operation applicable to the wind speed bin and summing the results. Mathematically, the summation is expressed as follows:

$$F_{\text{Long-term}}(x) \approx \sum_{k=1}^M F(x | U_k) p_k \quad (5.150)$$

where p_k is the proportion of operating hours in the wind speed bin with mean wind speed U_k and the operating range of wind speeds is divided into M bins.

5.14.5 Extrapolation

The load with a 50-year return period is found by extrapolating $F_{\text{Long-term}}(x)$ to the requisite non-exceedance probability – that is, $(1 - 3.8 \times 10^{-7})$ for a ten-minute simulation period. As $F_{\text{Long-term}}(x)$ is the summation of several mathematically defined distributions, this is straightforward. Figure 5.48 illustrates how the extrapolation might look if a *single* wind speed bin dominated the extreme loads for the three extreme value distributions considered.

5.14.6 Fitting probability distribution after aggregation

With this sequence, the first step is to aggregate the data points from the simulations from all the wind speed bins. If the number of simulations from each wind speed bin is proportional to the number of hours of operation in that bin and there are m ten-minute simulations in all, then the m global extremes can be ranked $1, 2, \dots, i, \dots, m$ from smallest to largest and the long-term empirical load non-exceedance probability distribution for the ten-minute extreme load, x , can be constructed according to the formula

$$F_{\text{Long-term}}(x_i) = \frac{i}{m + 1}, \quad i = 1, 2, \dots, m$$

In cases where the number of simulations in each wind speed bin is not proportional to the actual likelihood of that bin, then different weights need to be assigned to the global extremes in different bins, and the probability distribution becomes

$$F_{\text{Long-term}}(x) = \sum_{k=1}^b \left(\sum_{i=1}^{n_k} \frac{I[x_{i,k} \leq x]}{m + 1} \right) w_k = \sum_{k=1}^b \left(\sum_{i=1}^{n_k} \frac{I[x_{i,k} \leq x]}{m + 1} \right) \frac{m p_k}{n_k} \quad (5.151)$$

where

$x_{i,k}$ is the i th extreme in the k th wind speed bin

$I[x_{i,k} \leq x]$ 1 if $x_{i,k} \leq x$ and 0 otherwise

n_k is the number of simulations in the k th wind speed bin

b is the number of wind speed bins

w_k is the weighting factor for k th wind speed bin

p_k is the proportion of operating hours in the k th wind speed bin

The second step is the fitting of an extreme value distribution to the long-term empirical load distribution to permit extrapolation to the 50-year return load. Clearly, the ‘aggregation

before fitting’ approach is simpler than the ‘fitting before aggregation’ approach in that only one fitting operation is required, although the inherent complexity of the long-term empirical distribution may make a close fit more difficult to achieve.

5.14.7 Local extremes method

The local extremes method is the same as the global extremes method, except that several independent extreme values in each ten-minute simulation are utilised in constructing the empirical load non-exceedance probability distribution(s) instead of just the single largest value – that is, the ‘global extreme’ – from each simulation.

If there are n independent extreme values in each ten-minute simulation, then the empirical non-exceedance probability distribution for the ten-minute extreme load, x_k , at wind speed U_k can be constructed from the non-exceedance probability distribution of the local extremes using

$$F(x_k | U_k) = [F_{local}(x_k | U_k)]^n$$

To ensure independence of the local extremes, Annex F of IEC 61400–1 Edition 3 suggests that individual extremes should be separated by at least three response cycles. A less rigorous but computationally simpler approach is to divide the ten-minute simulation period into n segments or ‘blocks’ of equal duration and use the maximum value from each. This method of extracting the local extremes is illustrated in Figure 5.50 for a turbine rotating at 15 rpm with a block size of 12 seconds, corresponding to three cycles of rotation.

Fogle *et al.* (2008) have applied statistical tests for independence to block maxima obtained from 200 simulated load time histories for a 5 MW turbine model developed by NREL, and

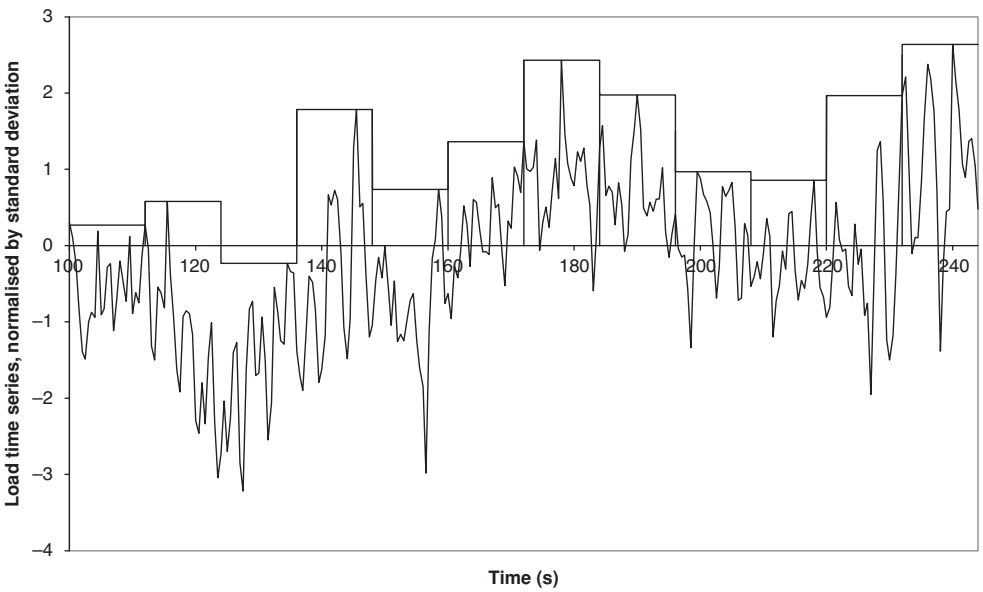


Figure 5.50 Local extremes derived from blocks of 12 seconds duration

concluded that a block duration of 30 seconds is required to ensure independence. However, they also found that the use of block durations as short as five seconds made negligible difference to the tail of the empirical probability distribution.

5.14.8 Convergence requirements

IEC 61400–1 Edition 3 imposes a limit on the uncertainty of extrapolated loads by requiring that sufficient simulations are carried out so that the 90% confidence interval on the 84% quantile load (that is, the extreme load in a ten-minute simulation likely to be exceeded 16% of the time) is less than 15% of that load. This requirement is expressed as

$$\frac{\hat{S}_{0.84,0.05} - \hat{S}_{0.84,0.95}}{\hat{S}_{0.84}} < 0.15 \quad (5.152)$$

where the confidence bounds $\hat{S}_{0.84,0.05}$ and $\hat{S}_{0.84,0.95}$ are the empirical estimates of the 84% quantile load with a 5% and 95% probability of not being exceeded respectively.

One approach to estimating the confidence bounds utilises the binomial expansion to obtain the probability, $C(j; m, 0.84)$, of j or fewer occurrences of the non-exceedance of the 84% quantile load in m simulations. This is given by:

$$C(j; m, 0.84) = \sum_{i=0}^j \frac{m!}{i! (m-i)!} 0.84^i 0.16^{m-i} \quad (5.153)$$

As an example, this probability is plotted against j for the case of $m = 15$ simulations in Figure 5.51. It is seen that the 5% and 95% confidence bounds on the number of trials in

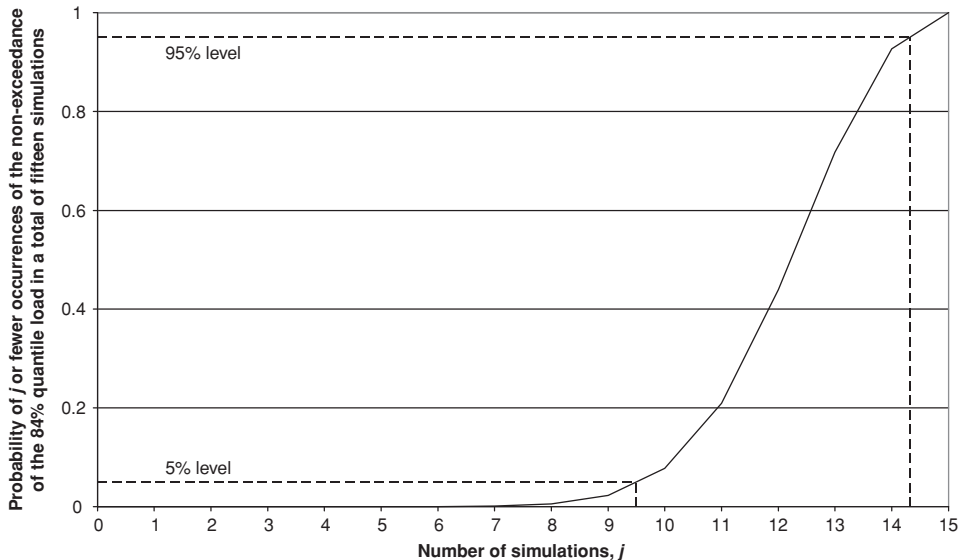


Figure 5.51 Probability of j or fewer occurrences of the non-exceedance of the 84% quantile load in a total of 15 simulations

Table 5.11 Table of confidence bounds on the number of simulations in which the 84% load is not exceeded for different numbers of simulations in total

Total no. of simulations, m	No. of occurrences of the non-exceedance of the 84% quantile load with 5% probability of not being exceeded	No. of occurrences of the non-exceedance of the 84% quantile load with 95% probability of not being exceeded
15	9.5	14.32
20	13.35	18.83
25	17.23	23.39
30	21.18	27.83
35	25.13	32.32

which the 84% quantile load is not exceeded are 9.5 and 14.3 respectively. The confidence bounds on the 84% quantile load, $\hat{S}_{0.84,0.05}$ and $\hat{S}_{0.84,0.95}$, required for the inequality (Equation 5.152) can then be derived by interpolating between the ninth and tenth and fourteenth and fifteenth ranked extremes respectively. The ranking is from smallest to largest as usual.

Annex F of IEC 61400–1 Edition 3 provides a table of 5% and 95% confidence bounds on the number of simulations in which the 84% quantile load is not exceeded for each value of m , the total number of simulations, from 15 to 35. Selected values are given in Table 5.11 above.

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Appendix 5: Dynamic response of stationary blade in turbulent wind

A5.1 Introduction

As described in Chapter 2, the turbulent wind contains wind speed fluctuations over a wide range of frequencies, as described by the power spectrum. Although the bulk of the turbulent energy is normally at frequencies much lower than the blade first mode out-of-plane frequency, which is typically over 1 Hz, the fraction close to the first mode frequency will excite resonant

blade oscillations. This appendix describes the method by which the resonant response may be determined. Working in the frequency domain, expressions for the standard deviations of both the tip displacement and root bending moment responses are derived, and then the method of deriving the peak value in a given period is described. Initially the wind is assumed to be perfectly correlated along the blade, but subsequently the treatment is extended to include the effect of spatial variation.

A5.2 Frequency response function

A5.2.1 Equation of motion

The dynamic response of a cantilever blade to the fluctuating aerodynamic loads upon it is most conveniently investigated by means of modal analysis, in which the excitations of the various different natural modes of vibration are computed separately and the results superposed. Thus, the deflection $x(r, t)$ at radius r is given by:

$$x(r, t) = \sum_{i=1}^{\infty} f_i(t) \mu_i(t)$$

Normally, in the case of a stationary blade, the first mode dominates and higher modes do not need to be considered. The equation of motion for the i th mode, which is derived in Section 5.8.1, is as follows:

$$m_i \ddot{f}_i(t) + c_i \dot{f}_i(t) + m_i \omega_i^2 f_i(t) = \int_0^R \mu_i(r) q(r, t) dr \quad (\text{A5.1})$$

where

$q(r, t)$ is the applied loading

$f_i(t)$ is the tip displacement

$\mu_i(r)$ is the non-dimensional mode shape of the i th mode, normalised to give a tip displacement of unity

ω_i is the natural frequency in radians per second

m_i is the generalised mass, $\int_0^R m(r) \mu_i^2(r) dr$

and c_i is the generalised damping, $\int_0^R \hat{c}(r) \mu_i^2(r) dr$.

A5.2.2 Frequency response function

If $q(r, t)$ varies harmonically, with frequency ω and amplitude $q_0(r)$, then it can be shown that:

$$\begin{aligned} f_i(t) &= \frac{1}{m_i} \frac{\int_0^R \mu_i(r) q_0(r) dr}{\sqrt{(\omega_i^2 - \omega^2)^2 + (c_i/m_i)^2 \omega^2}} \cos(\omega t + \phi_i) \\ &= \frac{1}{m_i \omega_i^2} \frac{\int_0^R \mu_i(r) q_0(r) dr}{\sqrt{(1 - \omega^2/\omega_i^2)^2 + (c_i/m_i \omega_i^2)^2 \omega^2}} \cos(\omega t + \phi_i) \end{aligned} \quad (\text{A5.2})$$

Defining $k_i = m_i \omega_i^2$, and noting that the damping ratio $\xi_i = c_i / 2m_i \omega_i$, this becomes:

$$f_i(t) = \frac{1}{k_i} \frac{\int_0^R \mu_i(r) q_0(r) dr}{\sqrt{(1 - \omega^2 / \omega_i^2)^2 + 4\xi_i^2 \omega^2 / \omega_i^2}} \cos(\omega t + \phi_i) = A_i \cos(\omega t + \phi_i) \quad (\text{A5.3})$$

The numerator $\int_0^R \mu_i(r) q_0(r) dr$ is the amplitude of the equivalent harmonic loading at the tip of the cantilever that would result in the same tip displacement as the loading $q(r, t)$, and is known as the generalised load with respect to the i th mode, $Q_i(t)$. Thus, the ratio between the tip displacement amplitude and the amplitude of the generalised load is

$$\begin{aligned} \frac{A_i}{\int_0^R \mu_i(r) q_0(r) dr} &= \frac{1}{k_i \sqrt{(1 - \omega^2 / \omega_i^2)^2 + 4\xi_i^2 \omega^2 / \omega_i^2}} \\ &= \frac{1}{k_i \sqrt{(1 - n^2 / n_i^2)^2 + 4\xi_i^2 n^2 / n_i^2}} = |H_i(n)| \end{aligned} \quad (\text{A5.4})$$

The ratio $|H_i(n)|$ is the modulus of the complex frequency response function, $H_i(n)$, and its square can be used to transform the power spectrum of the wind incident on the blade into the power spectrum of the i th mode tip displacement. Thus, in the case of the dominant first mode, the tip displacement in response to a harmonic generalised loading, $Q_1(t)$, of frequency n is given by

$$x_1(R, t) = f_1(t) = Q_1(t) |H_1(n)|$$

and the power spectrum of the first mode tip deflection is $S_{1x}(n) = S_{Q1}(n) |H_1(n)|^2$.

In what follows, the simplifying assumption is made initially that the wind is perfectly correlated along the blade.

A5.3 Resonant displacement response ignoring wind variations along the blade

A5.3.1 Linearisation of wind loading

For a fluctuating wind speed $U(t) = \bar{U} + u(t)$, the wind load per unit length on the blade is $\frac{1}{2} C_f \rho U^2(t) c(r) = \frac{1}{2} C_f \rho [\bar{U}^2 + 2\bar{U}u(t) + u^2(t)] c(r)$, where C_f is the lift or drag coefficient, as appropriate, and $c(r)$ is the local blade chord dimension. In order to permit a linear treatment, the third term in the square brackets, which will normally be small compared to the first two, is ignored, so that the fluctuating load $q(r, t)$ becomes $C_f \rho \bar{U} u(t) c(r)$.

A5.3.2 First mode displacement response

Setting $q(r, t) = C_f \rho \bar{U} u(t) c(r)$, the first mode tip displacement response to a sinusoidal wind fluctuation of frequency $n (= \omega/2\pi)$ and amplitude $u_o(n)$ given by Equation A5.3 becomes

$$f_1(t) = \int_0^R \mu_1(r) C_f \rho \bar{U} c(r) dr u_o(n) |H_1(n)| \cos(2\pi n t + \phi_1) = C_f \rho \bar{U} \int_0^R \mu_1(r) c(r) dr u_o(n) |H_1(n)| (\cos 2\pi n t + \phi_1) \quad (\text{A5.5})$$

Hence, the power spectrum of first mode tip displacement is

$$S_{1x}(n) = \left[C_f \rho \bar{U} \int_0^R \mu_1(r) c(r) dr \right]^2 S_u(n) |H_1(n)|^2 \quad (\text{A5.6})$$

where $S_u(n)$ is the power spectrum for the along wind turbulence. Thus, the standard deviation of the first mode tip displacement is given by

$$\sigma_{1x}^2 = \left[C_f \rho \bar{U} \int_0^R \mu_1(r) c(r) dr \right]^2 \int_0^\infty S_u(n) |H_1(n)|^2 dn \quad (\text{A5.7})$$

A5.3.3 Background and resonant response

Normally the bulk of the turbulent energy in the wind is at frequencies well below the frequency of the first out-of-plane blade mode. This is illustrated in Figure A5.1, where a typical power spectrum for wind turbulence is compared with the square, $|H_1(n)|^2$, of an example frequency response function for a 1 Hz resonant frequency.

The power spectrum is that due to Kaimal (and adopted in Eurocode 1, 2005):

$$n S_u(n) = \sigma_u^2 \frac{6.8 n L_u^x / \bar{U}}{(1 + 10 \cdot 2 n L_u^x / \bar{U})^{\frac{5}{3}}} \quad (\text{A5.8})$$

and is plotted as the non dimensional power-spectral density function, $R_u(n) = n \underline{c} S_u(n) / \sigma_u^2$, against a logarithmic frequency scale. The time scale, $L_u^x / \bar{U}$, chosen is four seconds, based on a mean wind speed, $\bar{U}$, of 50 m/s and an integral length scale, L_u^x , of 200 m.

In view of the fact that the resonant response usually occurs over a narrow band of frequencies on the tail of the power spectrum, it is normal to treat it separately from the quasistatic response at lower frequencies, and to ignore the variation in $n \cdot S_u(n)$ on either side of the resonant frequency, n_1 . (See for example Wyatt, 1980). The variance of total tip displacement then becomes:

$$\sigma_x^2 = \sigma_B^2 + \sigma_{x1}^2$$

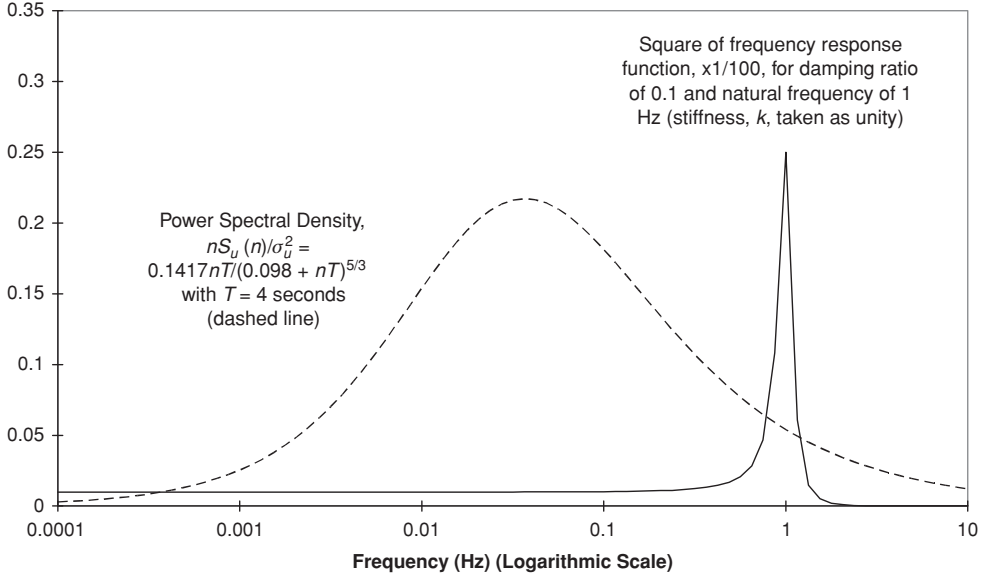


Figure A5.1 Power spectrum of wind turbulence and frequency response function

in which the variance of the first mode resonant response, σ_{x1} , is given by

$$\sigma_{x1}^2 = \left[C_f \rho \bar{U} \int_0^R \mu_1(r) c(r) dr \right]^2 S_u(n_1) \int_0^\infty |H_1(n)|^2 dn \quad (\text{A5.9})$$

and the resonant response of higher modes, σ_{x2}^2 , σ_{x3}^2 etc. are ignored. The non-resonant response, σ_B , is termed the background response, and can be derived from simple static beam theory.

It has been shown by Newland (1984) that $\int_0^\infty |H_1(n)|^2 dn$ reduces to $(\pi^2/2\delta)(n_1/k_1^2)$, where δ is the logarithmic decrement of damping. The logarithmic decrement, δ is 2π times the damping ratio, ξ_1 , defined as $\xi_1 = c_1/2m_1\omega_1$. Hence Equation A5.9 becomes

$$\sigma_{x1}^2 = \left[C_f \rho \bar{U} \int_0^R \mu_1(r) c(r) dr \right]^2 S_u(n_1) \frac{\pi^2}{2\delta} \frac{n_1}{k_1^2} \quad (\text{A5.10})$$

For comparison, the first mode component, $\bar{x}_1$, of the steady response is obtained by setting $\omega = 0$ and $q_0(r) = \frac{1}{2}\rho\bar{U}^2 C_f c(r)$ in Equation A5.3, yielding

$$\bar{x}_1 = \frac{1}{2} \rho \bar{U}^2 C_f \frac{1}{k_1} \int_0^R \mu_1(r) c(r) dr \quad (\text{A5.11})$$

Hence, the ratio of the standard deviation of the first mode resonant response to the first mode component of the steady response is

$$\frac{\sigma_{x1}}{\bar{x}_1} = 2 \frac{\sigma_u}{\bar{U}} \frac{\pi}{\sqrt{2\delta}} \sqrt{\frac{n_1 S_u(n_1)}{\sigma_u^2}} = 2 \frac{\sigma_u}{\bar{U}} \frac{\pi}{\sqrt{2\delta}} \sqrt{R_u(n_1)} \quad (\text{A5.12})$$

Note that towards the upper tail of the power spectrum of along wind turbulence, where n_1 is likely to be located, $\sqrt{R_u(n_1)}$ tends to $\sqrt{0.1417 / (n L_u^x / \bar{U})^{\frac{2}{3}}}$.

A5.4 Effect of across-wind turbulence distribution on resonant displacement response

In the foregoing treatment, the wind was assumed to be perfectly correlated along the blade. The implications of removing this simplifying assumption will now be examined.

The fluctuating load on the blade, $q(r, t)$, becomes $C_f \rho \bar{U} u(r, t) c(r)$ per unit length, and the generalised fluctuating load with respect to the first mode becomes

$$Q_1(t) = \int_0^R \mu_1(r) q(r, t) dr = C_f \rho \bar{U} \int_0^R u(r, t) c(r) \mu_1(r) dr \quad (\text{A5.13})$$

The standard deviation, σ_Q , of $Q_1(t)$ is given by

$$\begin{aligned} \sigma_{Q1}^2 &= \frac{1}{T} \int_0^T Q_1^2(t) dt = (\rho \bar{U} C_f)^2 \frac{1}{T} \int_0^T \left[\int_0^R u(r, t) c(r) \mu_1(r) dr \right] \\ &\quad \times \left[\int_0^R u(r', t) c(r') \mu_1(r') dr' \right] dt \\ &= (\rho \bar{U} C_f)^2 \int_0^R \int_0^R \left[\frac{1}{T} \int_0^T u(r, t) u(r', t) dt \right] c(r) c(r') \mu_1(r) \mu_1(r') dr dr' \end{aligned} \quad (\text{A5.14})$$

Now the expression within the square brackets is the cross correlation function, $\kappa_u(r, r', \tau) = E\{u(r, t) u(r', t + \tau)\}$ with τ set equal to zero. The cross correlation function is related to the cross spectrum, $S_{uu}(r, r', n)$, as follows:

$$\begin{aligned} \kappa_u(r, r', \tau) &= \frac{1}{2} \int_{-\infty}^{\infty} S_{uu}(r, r', n) \exp(i 2\pi n \tau) dn, \text{ giving} \\ \kappa_u(r, r', 0) &= \left[\frac{1}{T} \int_0^T u(r, t) u(r', t) dt \right] = \int_0^{\infty} S_{uu}(r, r', n) dn \quad \text{for } \tau = 0 \end{aligned} \quad (\text{A5.15})$$

Hence,

$$\sigma_{Q1}^2 = (\rho \bar{U} C_f)^2 \int_0^R \int_0^R \left[\int_0^{\infty} S_{uu}(r, r', n) dn \right] c(r) c(r') \mu_1(r) \mu_1(r') dr dr' \quad (\text{A5.16})$$

The normalised cross spectrum is defined as $S_{uu}^N(r, r', n) = S_{uu}(r, r', n)/S_u(n)$, and like $S_{uu}(r, r', n)$, is in general a complex quantity, because of phase differences between the wind speed fluctuations at different heights. As only in-phase wind speed fluctuations will affect the response, we consider only the real part of the normalised cross spectrum, known as the normalised co-spectrum, and denoted by $\psi_{uu}^N(r, r', n)$. Substituting in Equation A5.16, we obtain:

$$\sigma_{Q1}^2 = (\rho \bar{U} C_f)^2 \int_0^R \int_0^R \left[\int_0^\infty S_u(n) \psi_{uu}^N(r, r', n) dn \right] c(r)c(r')\mu_1(r)\mu_1(r') dr dr' \quad (\text{A5.17})$$

From this, it can be deduced that the power spectrum of the generalised load with respect to the first mode is

$$S_{Q1}(n) = (\rho \bar{U} C_f)^2 \int_0^R \int_0^R S_u(n) \psi_{uu}^N(r, r', n) c(r)c(r')\mu_1(r)\mu_1(r') dr dr' \quad (\text{A5.18})$$

Note that the power spectrum for the along wind turbulence shows some variation with height, and so should strictly be written $S_u(n, z)$ instead of $S_u(n)$. However, the variation along the length of a vertical blade is small, and is ignored here.

As for the initial case when wind loadings along the blade were assumed to be perfectly correlated, the power spectrum for first mode tip displacement is equal to the product of the power spectrum of the generalised load (with respect to the first mode) and the square of the frequency response function, that is,

$$S_{1x}(n) = S_{Q1}(n) |H_1(n)|^2 \quad (\text{A5.19})$$

As before, $S_{Q1}(n)$ is assumed constant over the narrow band of frequencies straddling the resonant frequency, and the standard deviation of resonant tip response becomes

$$\sigma_{x1}^2 = S_{Q1}(n_1) \int_0^\infty |H_1(n)|^2 dn = S_{Q1}(n_1) \frac{\pi^2 n_1}{2\delta k_1^2} \quad (\text{A5.20})$$

A5.4.1 Formula for normalised co-spectrum

It remains to evaluate $S_{Q1}(n_1) = (\rho \bar{U} C_f)^2 S_u(n_1) \int_0^R \int_0^R \psi_{uu}^N(r, r', n) c(r)c(r')\mu_1(r)\mu_1(r') dr dr'$. The normalised co-spectrum, $\psi_{uu}^N(r, r', n)$ must decrease as the spacing $|r - r'|$ between the two points considered increases, and intuitively it is to be expected that the decrease would be more rapid for the higher frequency components of wind fluctuation. On an empirical basis, Davenport (1962) has proposed an exponential expression for the normalised co-spectrum as follows:

$$\psi_{uu}^N(r, r', n) = \exp[-C |r - r'| n / \bar{U}] \quad (\text{A5.21})$$

where C is a non-dimensional decay constant. Davenport noted that measurements by Cramer (1958) indicated values of C ranging from 7 in unstable conditions to 50 in stable conditions, but recommended the use of the lower figure as being the more conservative despite the likelihood of stable conditions in high winds. Dyrbye and Hansen (1997) quote Risø measurements

reported by Mann (1994) which indicate a value of C of 9.4, and they recommend a value of 10 for use in design. A value of 11.5 is implicitly assumed in Eurocode 1.

There is an obvious inconsistency in the exponential expression for the normalised co-spectrum – when it is integrated up over the plane perpendicular to the wind direction, the result is positive instead of zero as it should be. This has led to the development of more complex expressions by Harris (1971) and Krenk (1995). However, the Davenport formulation will be used here, giving

$$\sigma_{x1}^2 = S_{Q1}(n_1) \frac{\pi^2}{2\delta} \frac{n_1}{k_1^2} = (\rho \bar{U} C_f)^2 S_u(n_1) \times \int_0^R \int_0^R \exp[-C|r - r'|n_1/\bar{U}] c(r) c(r') \mu_1(r) \mu_1(r') dr dr' \left[\frac{\pi}{2\delta} \frac{n_1}{k_1^2} \right] \quad (A5.22)$$

The resonant response can be expressed in terms of the first mode component, $\bar{x}_1$, of the steady response, $\frac{1}{2}\rho\bar{U}^2 C_f(1/k_1) \int_0^R \mu_1(r) c(r) dr$, from Equation A5.11, giving

$$\frac{\sigma_{x1}^2}{\bar{x}_1^2} = 4 \frac{\sigma_u^2}{\bar{U}^2} \frac{\pi^2}{2\delta} R_u(n_1) \times \frac{\int_0^R \int_0^R \exp[-C|r - r'|n_1/\bar{U}] c(r) c(r') \mu_1(r) \mu_1(r') dr dr'}{\left(\int_0^R c(r) \mu_1(r) dr \right)^2} \quad (A5.23)$$

Hence

$$\frac{\sigma_{x1}}{\bar{x}_1} = 2 \frac{\sigma_u}{\bar{U}} \frac{\pi}{\sqrt{2\delta}} \sqrt{R_u(n_1)} \sqrt{K_{Sx}(n_1)} \quad (A5.24)$$

where

$$K_{Sx}(n_1) = \frac{\int_0^R \int_0^R \exp[-C|r - r'|n_1/\bar{U}] c(r) c(r') \mu_1(r) \mu_1(r') dr dr'}{\left(\int_0^R c(r) \mu_1(r) dr \right)^2} \quad (A5.25)$$

is denoted the size reduction factor, which results from the lack of correlation of the wind along the blade. As an example, the size reduction factor, $K_{Sx}(n_1)$, is plotted out against frequency in Figure A5.2 for the case of a 40 m blade with chord $c(r) = 0.0961R - 0.06467r$ (Blade TR), assuming a decay constant C of 10, and a mean wind speed $\bar{U}$ of 50 m/s. The mode shape taken is the same as for the example in Section 5.6.3 (see Figure 5.4). Also shown for comparison is the corresponding parameter for a uniform cantilever.

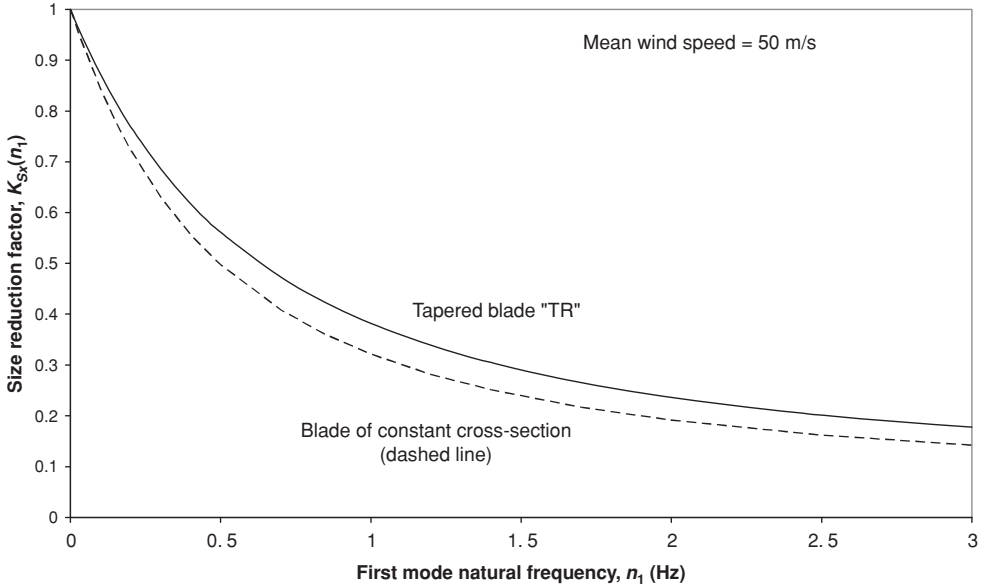


Figure A5.2 Size reduction factors for the first mode resonant response due to lack of correlation of wind loading along the blade: variation with frequency for 40 m blade

A5.5 Resonant root bending moment

For design purposes it is the augmentation of blade bending moments due to dynamic effects that is of principal significance. The ratio of the standard deviation of the first mode resonant root bending moment to the steady root bending moment (allowing for the lack of correlation of wind fluctuations along the blade) is derived below.

Defining $M_1(t)$ as the fluctuating root bending moment due to wind excitation of the first mode, we have

$$M_1(t) = \int_0^R m(r) \ddot{x}_1(t, r) r \, dr = \int_0^R m(r) \omega_1^2 x_1(t, r) r \, dr = \omega_1^2 f_1(t) \int_0^R m(r) \mu_1(r) r \, dr \quad (\text{A5.26})$$

Hence the standard deviation of $M_1(t)$,

$$\sigma_{M1} = \omega_1^2 \sigma_{x1} \int_0^R m(r) \mu_1(r) r \, dr \quad (\text{A5.27})$$

The steady root bending moment,

$$\bar{M} = \int_0^R \frac{1}{2} \rho \bar{U}^2 C_f c(r) r \, dr = \frac{1}{2} \rho \bar{U}^2 C_f \int_0^R c(r) r \, dr \quad (\text{A5.28})$$

Hence the ratio

$$\frac{\sigma_{M1}}{\bar{M}} = \frac{\omega_1^2 \sigma_{x1} \int_0^R m(r) \mu_1(r) r dr}{\frac{1}{2} \rho \bar{U}^2 C_f \int_0^R c(r) r dr} \quad (\text{A5.29})$$

Substituting the expression for σ_{x1} from Equation A5.22, we obtain

$$\begin{aligned} \frac{\sigma_{M1}}{\bar{M}} &= \frac{\omega_1^2 \rho \bar{U} C_f \frac{\pi}{\sqrt{2\delta}} \frac{\sqrt{n_1 S_u(n_1)}}{k_1} \int_0^R m(r) \mu_1(r) r dr \sqrt{\int_0^R \int_0^R \exp[-C|r - r'|n_1/\bar{U}] c(r) c(r') \mu_1(r) \mu_1(r') dr dr'}}{\frac{1}{2} \rho \bar{U}^2 C_f \int_0^R c(r) r dr} \end{aligned} \quad (\text{A5.30})$$

Noting that $R_u(n) = n S_u(n) / \sigma_u^2$, and that $k_1 = m_1 \omega_1^2$, this simplifies to

$$\begin{aligned} \frac{\sigma_{M1}}{\bar{M}} &= 2 \frac{\sigma_u}{\bar{U}} \frac{\pi}{\sqrt{2\delta}} \sqrt{R_u(n)} \frac{\int_0^R m(r) \mu_1(r) r dr}{m_1 \int_0^R c(r) r dr} \\ &\quad \times \sqrt{\int_0^R \int_0^R \exp[-C|r - r'|n_1/\bar{U}] c(r) c(r') \mu_1(r) \mu_1(r') dr dr'} \end{aligned} \quad (\text{A5.31})$$

where $m_1 = \int_0^R m(r) \mu_1^2(r) dr$ is the generalised mass with respect to the first mode, and the exponential expression within the double integral allows for the lack of correlation of wind fluctuations along the blade. Substituting $(\int_0^R c(r) \mu_1(r) dr) \sqrt{K_{Sx}(n_1)}$ for the square root of the double integral, using Equation A5.25, leads to

$$\frac{\sigma_{M1}}{\bar{M}} = 2 \frac{\sigma_u}{\bar{U}} \frac{\pi}{\sqrt{2\delta}} \sqrt{R_u(n)} \frac{\int_0^R m(r) \mu_1(r) r dr}{m_1 \int_0^R c(r) r dr} \left(\int_0^R c(r) \mu_1(r) dr \right) \sqrt{K_{Sx}(n_1)} \quad (\text{A5.32})$$

Defining the ratio of the integrals,

$$\frac{\int_0^R m(r) \mu_1(r) r dr}{m_1 \int_0^R c(r) r dr} \left(\int_0^R c(r) \mu_1(r) dr \right) = \frac{\sigma_{M1}}{\frac{\bar{M}}{\sigma_{x1}}}, \quad \text{as } \lambda_{M1} \quad (\text{A5.33})$$

we obtain

$$\frac{\sigma_{M1}}{\bar{M}} = 2 \frac{\sigma_u}{\bar{U}} \frac{\pi}{\sqrt{2\delta}} \sqrt{R_u(n_1)} \sqrt{K_{Sx}(n_1)} \lambda_{M1} \quad (\text{A5.34})$$

A5.6 Root bending moment background response

The root bending moment background response can be expressed in terms of the standard deviation of the root bending moment excluding resonant effects. If the wind is perfectly correlated along the blade, this is given by

$$\sigma_{MB} = C_f \rho \bar{U} \sigma_u \int_0^R c(r) r dr \quad (\text{A5.35})$$

However, if the lack of correlation of wind fluctuations along the blade is taken into account,

$$\sigma_{MB} = C_f \rho \bar{U} \sigma_u \sqrt{\int_0^R \int_0^R \rho_u(r - r') c(r) c(r') r r' dr dr'} \quad (\text{A5.36})$$

where $\rho_u(r - r')$ is the normalised cross correlation function between simultaneous wind speed fluctuations at two different blade radii, and is defined as

$$\rho_u(r - r') = \frac{1}{\sigma_u^2} E \{u(r, t) u(r', t + \tau)\} \quad \text{with } \tau \text{ set equal to zero.} \quad (\text{A5.37})$$

Measurements indicate that the normalised cross correlation function decays exponentially, so it can be expressed as

$$\rho_u(r - r') = \exp[-|r - r'|/L_u^r] \quad (\text{A5.38})$$

where L_u^r is the integral length scale for the longitudinal turbulence component measured in the across wind direction along the blade, and is thus defined as $\int_0^\infty \rho_u(r - r') d(r - r')$. As the integral length scale for longitudinal turbulence measured vertically in the across wind direction (L_u^z), is, if anything, less than that measured horizontally (L_u^y), it is conservative to treat it as being equal to that measured horizontally, with the result that L_u^r can be taken as equal to L_u^y also. Typically L_u^y is approximately equal to 30% of L_u^x , the integral length scale for longitudinal turbulence measured in the along wind direction. Observing that

$$\bar{M} = \frac{1}{2} \rho \bar{U}^2 C_f \int_0^R c(r) r dr$$

we can therefore write

$$\frac{\sigma_{MB}}{\bar{M}} = 2 \frac{\sigma_u}{\bar{U}} \sqrt{K_{SMB}} \quad (\text{A5.39})$$

where K_{SMB} , the size reduction factor for the root bending moment background response, is defined as

$$K_{SMB} = \frac{\int_0^R \int_0^R \exp[-|r - r'|/0.3L_u^x] c(r)c(r')rr' dr dr'}{\left(\int_0^R c(r)r dr \right)^2} \quad (A5.40)$$

For a blade with a uniform chord, the integral is straightforward, giving

$$K_{SMB} = 4 \left[\frac{2}{3\phi} - \frac{1}{\phi^2} + \frac{2}{\phi^4} - \exp(-\phi) \left\{ \frac{2}{\phi^3} + \frac{2}{\phi^4} \right\} \right] \quad \text{where} \quad \phi = \frac{R}{0.3L_u^x} \quad (A5.41)$$

As an example, K_{SMB} comes to 0.837 for the case of $R = 40$ m and $L_u^x = 189$ m indicating that the lack of correlation of the wind fluctuations reduces the root bending moment appreciably.

For blades with a normal tapering chord, K_{SMB} can be evaluated numerically. In the case of a blade with a tip chord equal to 33% of the maximum chord, K_{SMB} is 0.829 for the same value of ϕ as before. It is seen that the taper has an almost negligible effect on the end result.

A5.7 Peak response

One of the key parameters required in blade design is the extreme value of the out-of-plane bending moment. The 50-year return moment is defined as the expected maximum moment occurring during the mean wind averaging period when the mean takes the 50-year return value. Treating the moment as a Gaussian process, Davenport (1964) has shown that the expected value of the maximum departure from the mean is the standard deviation multiplied by the peak factor, g , where

$$g = \sqrt{2 \ln(vT)} + \frac{0.5772}{\sqrt{2 \ln(vT)}} \quad (A5.42)$$

In this formula, v is the mean zero-upcrossing frequency of the root moment fluctuations, and T is the mean wind speed averaging period. The variance of the root bending moment is, in the same way as for the tip displacement, equal to the sum of the variances of the background and resonant root bending moment responses, that is,

$$\sigma_M^2 = \sigma_{MB}^2 + \sigma_{M1}^2 \quad (A5.43)$$

Hence, from Equations A5.39 and A5.34, we obtain

$$\sigma_M^2 = \sigma_{MB}^2 + \sigma_{M1}^2 = 4\overline{M}^2 \frac{\sigma_u^2}{U^2} \left[K_{SMB} + \frac{\pi^2}{2\delta} R_u(n_1) K_{Sx}(n_1) \lambda_{M1}^2 \right] \quad (A5.44)$$

Thus,

$$M_{\max} = \bar{M} + g\sigma_M = \bar{M} \left(1 + 2g \frac{\sigma_u}{\bar{U}} \sqrt{K_{SMB} + \frac{\pi^2}{2\delta} R_u(n_1) K_{Sx}(n_1) \lambda_{M1}^2} \right) \quad (\text{A5.45})$$

The mean zero up-crossing frequency of the root moment fluctuations, ν , is defined as

$$\nu = \sqrt{\frac{\int_0^\infty n^2 S_M(n) dn}{\int_0^\infty S_M(n) dn}} \quad (\text{A5.46})$$

where $S_M(n)$ is the power spectrum of the root moment fluctuations. If we separate the power spectrum of the background response from the first mode resonant response at frequency n_1 , then the above expression can be written

$$\nu = \sqrt{\frac{\left(\int_0^\infty n^2 S_{MB}(n) dn \right) + n_1^2 \sigma_{M1}^2}{\sigma_{MB}^2 + \sigma_{M1}^2}} \quad (\text{A5.47})$$

Now

$$S_{MB}(n) = (C_f \rho \bar{U})^2 S_u(n) \int_0^R \int_0^R \psi_{uu}^N(r, r', n) c(r) c(r') r r' dr dr' \quad (\text{A5.48})$$

and

$$\bar{M} = C_f \frac{1}{2} \rho \bar{U}^2 \int_0^R c(r) r dr \quad (\text{A5.49})$$

so

$$S_{MB}(n) = 4 \frac{\bar{M}^2}{\bar{U}^2} \frac{S_u(n) \int_0^R \int_0^R \psi_{uu}^N(r, r', n) c(r) c(r') r r' dr dr'}{\left(\int_0^R c(r) r dr \right)^2} \quad (\text{A5.50})$$

Defining

$$K_{SMB}(n) = \frac{\int_0^R \int_0^R \psi_{uu}^N(r, r', n) c(r) c(r') r r' dr dr'}{\left(\int_0^R c(r) r dr \right)^2} \quad (\text{A5.51})$$

we obtain

$$S_{MB}(n) = 4 \frac{\overline{M}^2}{\overline{U}^2} S_u(n) K_{SMB}(n) \quad (\text{A5.52})$$

Substituting into Equation A5.47 gives

$$\nu = \sqrt{\frac{4 \frac{\overline{M}^2}{\overline{U}^2} \left(\int_0^\infty n^2 S_u(n) K_{SMB}(n) dn \right) + n_1^2 \sigma_{M1}^2}{\sigma_{MB}^2 + \sigma_{M1}^2}} \quad (\text{A5.53})$$

Noting from Equation A5.52 that $\sigma_{MB}^2 = 4(\overline{M}^2/\overline{U}^2) \int_0^\infty S_u(n) K_{SMB}(n) dn$, the expression for ν becomes

$$\nu = \sqrt{\frac{n_0^2 \sigma_{MB}^2 + n_1^2 \sigma_{M1}^2}{\sigma_{MB}^2 + \sigma_{M1}^2}} \quad (\text{A5.54})$$

where

$$n_0 = \frac{\int_0^\infty n^2 S_u(n) K_{SMB}(n) dn}{\int_0^\infty S_u(n) K_{SMB}(n) dn} \quad (\text{A5.55})$$

Substituting $\psi_{uu}^N(r, r', n) = \exp[-C(r - r')n/\overline{U}]$ into the expression for $K_{SMB}(n)$ in the numerator of Equation A5.55 gives

$$\int_0^\infty n^2 S_u(n) K_{SMB}(n) dn = \int_0^\infty n^2 S_u(n) \frac{\int_0^R \int_0^R \exp[-C(r - r')n/\overline{U}] c(r) c(r') r r' dr dr'}{\left(\int_0^R c(r) r dr \right)^2} dn \quad (\text{A5.56})$$

For high frequencies, the double integral is, in the limit, inversely proportional to frequency, so the integrand $n^2 S_u(n) K_{SMB}(n)$ is proportional to $n^2 n^{-\frac{5}{3}} n^{-1} = n^{-\frac{2}{3}}$ and the integral does not converge. Consequently it is necessary to take account of the chordwise lack of correlation of wind fluctuation at high frequencies, and if this is done, it is found that, in the limit, the integrand is proportional to $n^{-\frac{5}{3}}$ for which the integral is finite. The evaluation of the integral $\int_0^\infty n^2 S_u(n) K_{SMB}(n) dn$ taking chordwise lack of correlation into account is a formidable task, so the use of an approximate formula for the frequency, n_0 , is preferable, especially as the influence of n_0 on the peak factor, g , is slight. Dyrbye and Hansen (1997) give an approximate formula for a uniform cantilever is as follows:

$$n_0 = 0.3 \frac{\overline{U}}{\sqrt{L_u^x \sqrt{Rc}}} \quad (\text{A5.57})$$

Here R is the blade tip radius and c is the blade chord, assumed constant. For a tapering chord, the mean chord, $\bar{c}$, can be substituted.

A5.8 Bending moments at intermediate blade positions

A5.8.1 Background response

Denoting the standard deviation of the quasistatic or background bending moment fluctuations at radius r^* as $\sigma_{MB}(r^*)$, it is apparent that

$$\frac{\sigma_{MB}(r^*)}{\sigma_{MB}(0)} = \sqrt{\frac{K_{SMB}(r^*)}{K_{SMB}(0)}} \frac{\int_{r^*}^R c(r)[r - r^*]dr}{\int_0^R c(r)r dr} \quad (\text{A5.58})$$

The ratio of the steady moment at radius r^* to that at the root is $(\int_{r^*}^R c(r)[r - r^*]dr)/(\int_0^R c(r)r dr)$, so the ratio of the standard deviation of the quasistatic fluctuations at radius r^* to the steady value there is

$$\frac{\sigma_{MB}(r^*)}{\bar{M}(r^*)} = \frac{\sigma_{MB}(r^*)}{\sigma_{MB}(0)} \frac{\sigma_{MB}(0)}{\bar{M}(0)} \frac{\bar{M}(0)}{\bar{M}(r^*)} = \frac{\sigma_{MB}(0)}{\bar{M}(0)} \sqrt{\frac{K_{SMB}(r^*)}{K_{SMB}(0)}} \quad (\text{A5.59})$$

Generally, the square root will be close to unity, so $\sigma_{MB}(r^*)/\bar{M}(r^*)$ will be nearly constant.

A5.8.2 Resonant response

In section A5.5 it was shown that the standard deviation of the first mode resonant root bending moment is equal to $\omega_1^2 \sigma_{x1} \int_0^R m(r)\mu_1(r)r dr$ (Equation A5.27). The corresponding quantity at other radii can be derived similarly, giving

$$\sigma_{M1}(r^*) = \omega_1^2 \sigma_{x1} \int_{r^*}^R m(r)\mu_1(r)[r - r^*]dr \quad (\text{A5.60})$$

Hence, the ratio of the standard deviation of the first mode resonant root bending moment at radius r^* to the steady value there is

$$\frac{\sigma_{M1}(r^*)}{\bar{M}(r^*)} = \frac{\sigma_{M1}(r^*)}{\sigma_{M1}(0)} \frac{\sigma_{M1}(0)}{\bar{M}(0)} \frac{\bar{M}(0)}{\bar{M}(r^*)} = \frac{\int_{r^*}^R m(r)\mu_1(r)[r - r^*]dr}{\int_0^R m(r)\mu_1(r)r dr} \frac{\int_0^R c(r)r dr}{\int_{r^*}^R c(r)[r - r^*]dr} \frac{\sigma_{M1}(0)}{\bar{M}(0)} \quad (\text{A5.61})$$

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6

Conceptual design of horizontal axis wind turbines

6.1 Introduction

Within the general category of horizontal axis wind turbines for grid applications there exists a great variety of possible machine configurations, power control strategies and braking systems. This chapter looks at the different areas where design choices have to be made, and considers the advantages and disadvantages of the more conventional options in each case. Inevitably there are situations in which decisions in one area can impact those in another, and some of these are noted.

Alongside these discrete design choices there are several fundamental design parameters, such as rotor diameter, machine rating and rotational speed, which also have to be established at the start of the design process. Continuous variables such as these lend themselves to mathematical optimisation, as described in the opening sections of the chapter.

An illuminating overview of the evolution of turbine design, including the power control strategies and drive train configurations adopted by particular manufacturers, is provided in *Wind Energy – The Facts* (2009) published by the European Wind Energy Association.

6.2 Rotor diameter

The issue of what size of turbine produces energy at minimum cost has been fiercely debated for a long time. Protagonists of large machines cite economies of scale and the increase in wind speed with height in their favour. From the other camp, the ‘square-cube law’, whereby energy capture increases as the square of the diameter, whereas rotor mass (and therefore cost) increases as the cube, is advanced as an argument against.

In reality, both arguments are correct, and there is a trade-off between economies of scale and a variant of the ‘square-cube law’ which takes into account the wind shear

effect. This trade-off can be examined with the help of simple cost modelling, which is considered next.

6.2.1 Cost modelling

The sensitivity of the cost of energy to changes in the values of parameters governing turbine design can be examined with the aid of a model of the way component costs vary in response. The normal procedure is to start with a baseline design, for which the costs of the various components are known. In a rigorous analysis, the chosen parameter is then assigned a different value and a fresh design developed, leading to revised component weights, based on which new component costs can be assigned.

In general, the cost of a component will not simply increase pro rata with its mass, but will contain elements that increase more slowly. Examples are the tower surface protective coating and tower longitudinal welds (assuming the number required is constant), the costs of which increase approximately as the *square* of the tower height, if all dimensions are proportional to this height. If the design parameter variation considered is only about $\pm 50\%$, it is usually sufficiently accurate to represent the relationship between component cost and mass as a linear one with a fixed component, as follows:

$$C(x) = C_B \left(\mu \frac{m(x)}{m_B} + (1 - \mu) \right) \quad (6.1)$$

where $C(x)$ and $m(x)$ are the cost and mass of the component respectively when the design parameter takes the value x , and C_B and m_B are the baseline values. μ is the proportion of the cost that varies with mass, which will obviously differ for different baseline machine sizes and for different components.

The choice of the value of μ inevitably requires considerable expertise as regards the way manufacturing costs vary with scale, which may be limited in the case of products at the early stage of development. In view of this, the effort of developing fresh designs for different design parameter values may well not be justified, so resort is often made to scaling ratios based on similarity relationships. This approach is adopted in the investigation of optimum machine size which follows.

6.2.2 Simplified cost model for machine size optimisation: an illustration

The baseline machine design is taken as a 60 m diameter, 1.5 MW turbine, with the costs of the various components taken from the Risø publication 'Cost optimisation of wind turbines for large-scale offshore windfarms' (Risø-R-1000) by Fuglsang and Thomsen (1998). These are given in Table 6.1 as a percentage of the total.

Machine designs for other diameters are obtained by scaling *all* dimensions of all components in the same proportion, except in the case of the gearbox, generator, grid connection and controller. Rotational speed is kept inversely proportional to rotor diameter to maintain constant tip speed, and hence constant tip speed ratio at a given wind speed. As a result, the maximum rotor aerodynamic thrust increases with the square of rotor diameter and the peak aerodynamic bending moment in each structural element, which is assumed to govern its design, increases with rotor diameter cubed. Given the assumption that all cross-sectional

Table 6.1 Component costs expressed as a percentage of total machine cost for a 1.5 MW, 60 m diameter, fixed speed, stall-regulated wind turbine on land (from Risø-R-1000, Fuglsang and Thomsen, 1998)

Component	Cost as a percentage of total	Component	Cost as a percentage of total
Blades	18.3%	Controller	4.2%
Hub	2.5%	Tower	17.5%
Main shaft	4.2%	Brake system	1.7%
Gearbox	12.5%	Foundation	4.2%
Generator	7.5%	Assembly	2.1%
Nacelle	10.8%	Transport	2.0%
Yaw system	4.2%	Grid connection	8.3%
		TOTAL	100%

dimensions increase in proportion to rotor diameter, the bending section moduli increase as diameter cubed, so each critical stress remains invariant with diameter.

Maintenance of constant tip speed also means that all machine designs reach rated power at the same wind speed, so that rated power is proportional to diameter squared. However, the low speed shaft torque increases as diameter cubed, which is the basis for assuming the gearbox mass increases as the cube of rotor diameter, even though the gearbox ratio changes. In the following illustration a blanket value of μ of 0.9 for all components is adopted for simplicity. Accordingly, the cost of all components apart from generator, controller and the grid connection, for a machine of diameter D , is given by:

$$C_1(D) = 0.8C_T(60) \left(0.9 \left(\frac{D}{60} \right)^3 + 0.1 \right) \quad (6.2)$$

where $C_T(60)$ is the total cost of the 60 m diameter baseline machine.

The rating of the generator and the grid connection is proportional only to the diameter squared. It is assumed that Equation 6.1 applies to the cost of these components, but with mass replaced by rating. Thus, the cost of the generator and grid connection are given by:

$$C_2(D) = 0.158C_T(60) \left(0.9 \left(\frac{D}{60} \right)^2 + 0.1 \right) \quad (6.3)$$

The controller cost is assumed to be fixed, regardless of turbine size. Hence, the resulting turbine cost as a function of diameter is:

$$\begin{aligned} C_T(D) &= C_T(60) \left(0.8 \left\{ 0.9 \left(\frac{D}{60} \right)^3 + 0.1 \right\} + 0.158 \left\{ 0.9 \left(\frac{D}{60} \right)^2 + 0.1 \right\} + 0.042 \right) \\ &= C_T(60) \left(0.72 \left(\frac{D}{60} \right)^3 + 0.1422 \left(\frac{D}{60} \right)^2 + 0.1378 \right) \end{aligned} \quad (6.4)$$

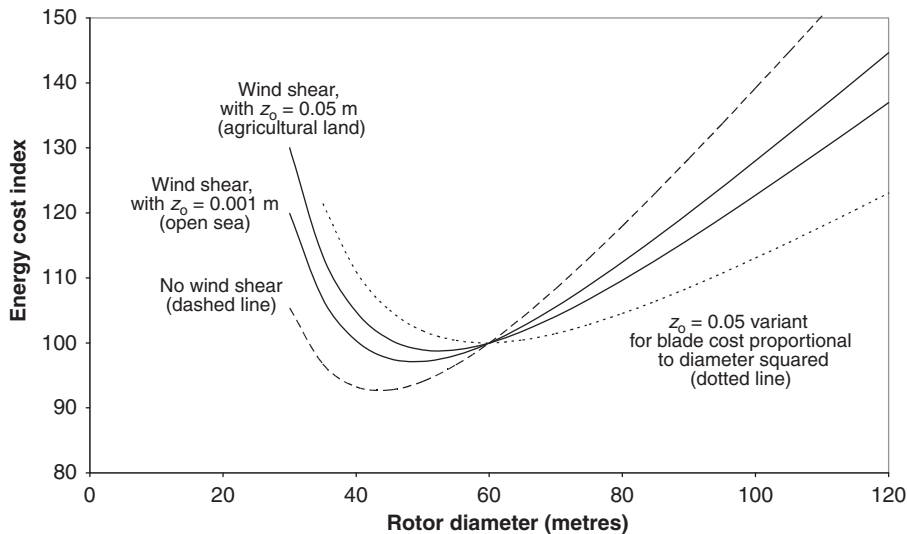


Figure 6.1 Variation of optimum turbine size with wind shear based on simplified cost model (assuming hub height equal to diameter)

As the tower height, along with all other dimensions, is assumed to increase in proportion to rotor diameter, the annual mean wind speed (*amws*) at hub height will increase with rotor diameter because of wind shear. This has a significant effect on energy yield, as the energy yield per unit of swept area is found to vary as the *amws* raised to the power of 1.9 for perturbations about the *amws* central value of 8 m/s taken in this example. The cost of energy (excluding operation and maintenance costs) can then be calculated in €/kWh/annum by dividing the turbine cost by the annual energy yield. The variation of energy cost with diameter, calculated according to the assumptions described above, is plotted in Figure 6.1 for two levels of wind shear corresponding to roughness lengths, z_0 , of 0.001 m and 0.05 m, the hub height wind speed being scaled according to the relation $\overline{U}(z) \propto \ln(z/z_0)$ (see Section 2.6.2). Also included is a plot for the case of zero wind shear.

It is apparent that the level of wind shear has a noticeable effect on the optimum machine diameter, which varies from 44 m for zero wind shear to 52 m for the wind shear corresponding to a surface roughness length of 0.05 m, which is applicable to farmland with boundary hedges and occasional buildings. Strictly, the impact of the increased annual mean wind speed with hub height on the fatigue design of the rotor and other components should also be taken into account, which would reduce the optimum machine size slightly.

It should be emphasised that the optimum sizes derived above depend critically on the value of μ adopted. For example, if μ were taken as 0.8 instead of 0.9, the optimum diameter would increase to 64 m for the wind shear corresponding to a surface roughness length of 0.05 m, although the minimum cost of energy would alter by only 0.9%. The correct approach would be to allocate different values of μ to different components, as is done in Fuglsang and Thomsen (1998). Ideally these would be based on cost data on components of the same design but different sizes.

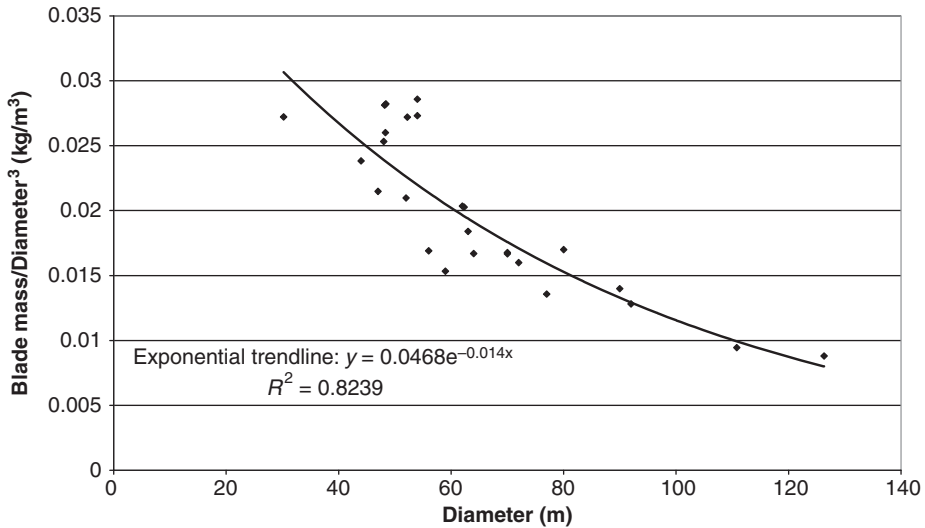


Figure 6.2 Variation of specific blade mass with diameter for LM blades available in 2004

The cost model outlined and illustrated above provides a straightforward means of investigating scale effects on machine economics for a chosen machine design. In practice, the use of different materials or different machine configurations may prove more economic at different machine sizes, and will yield a series of alternative cost versus diameter curves.

An example of the impact technological developments can have on simple scaling rules is provided by the trajectory of specific blade mass – defined as blade mass divided by turbine diameter cubed – as longer blade designs have evolved over time. Figure 6.2 shows a plot of specific blade mass against diameter for blades manufactured by LM Glasfiber that were available in 2004.

It is seen that the specific blade mass is approximately inversely proportional to diameter – that is, the blade mass has increased with diameter squared rather than with diameter cubed. The decline in specific blade mass is partly due to a lower practical limit on skin thicknesses, which made the smaller blades heavier than they would otherwise need to be.

The blade cost scaling rule adopted in the cost model obviously has a decisive effect on the optimum rotor diameter, which (for the case of wind shear corresponding to a surface roughness of 0.05 m) increases from 52 m to 59 m when the blade cost is scaled as the square of diameter, as indicated by the dotted line in Figure 6.1.

6.2.3 The NREL cost model

Research at NREL, reported in ‘Wind Turbine Design Cost and Scaling Model’ NREL (2006), has resulted in a useful set of mass and cost scaling rules for several wind turbine configurations, which serve as benchmarks against which innovations in the design of individual components can be judged. It is intended that the scaling rules are updated over time as additional data becomes available.

The baseline turbine design is a 70 m diameter, 1.5 MW variable speed pitch regulated machine, fitted with a three stage planetary gearbox. Many of the scaling rules are similar to those set out in the preceding section, with the main differences being as follows:

- Blade cost split into material and labour costs of similar magnitudes, with the former scaling as diameter cubed (D^3) and the latter scaling as $D^{2.5}$.
- Gearbox cost scaled as $D^{2.5}$ rather than D^3 .
- Nacelle cost scaled as $D^{1.95}$ rather than D^3 .
- Foundation cost scaled as $D^{1.2}$ rather than D^3 .
- An element of transport cost increasing as the sixth power of diameter.

Table 6.2 presents the component costs of the baseline turbine in 2005 dollars and the respective percentages of the total. Also shown are the corresponding percentages for the

Table 6.2 Component costs and percentages for NREL 1.5 MW 70 m diameter baseline machine

Component	Component costs for 1.5 MW, 70 m dia NREL baseline turbine (with 70 m hub height) \$1000 (2005)	Percentage component costs for NREL baseline turbine	Percentage component costs for Risø-R-1000 1.5 MW, 60 m dia turbine
Blades	151	11.4%	18.3%
Hub	47	3.6%	2.9%
Pitch bearings and mechanism	56	4.3%	N/A
Low speed shaft and main bearings	33	2.5%	4.2%
Gearbox	152	11.6%	12.9%
Generator	98	7.4%	7.5%
Variable speed electronics	119	9.0%	N/A
Nacelle	119	8.9%	10.8%
Yaw drive & bearing	20	1.5%	4.2%
Control system	35	2.7%	4.2%
Tower	158	12.0%	17.5%
Brake & HS coupling	3	0.2%	1.7%
Foundation	47	3.6%	4.2%
Assembly & installation	42	3.2%	2.1%
Transportation	51	3.9%	2.0%
Electrical connections inc grid connection	187	14.2%	8.3%
	1317	100%	100%

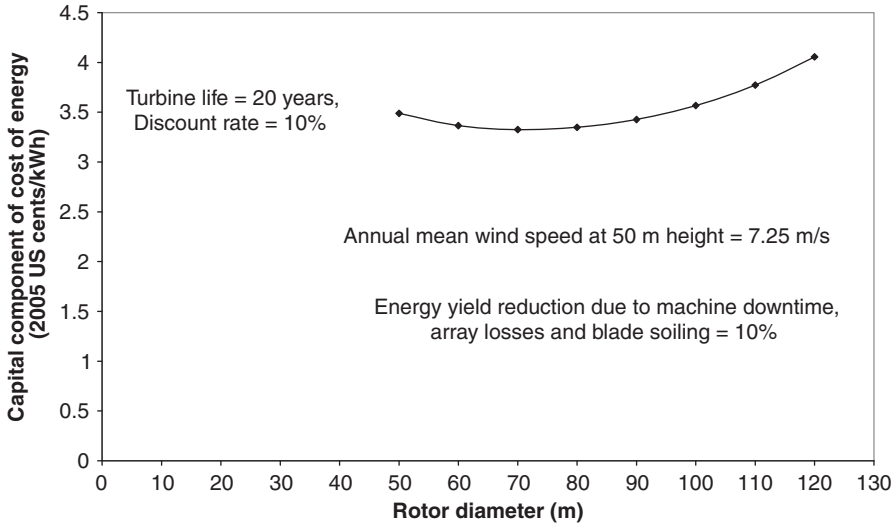


Figure 6.3 Variation of cost of energy with turbine diameter for NREL baseline machine—capital cost component only

60 m diameter 1.5 MW machine of the preceding section, where applicable. Caution should be exercised in making comparisons, however, as component definitions may vary.

Figure 6.3 presents the variation of the cost of energy capital component with turbine diameter, based on the NREL cost model for the machine described above. The annual mean wind speed at 50 m height is taken as 7.25 m/s and wind speed is assumed to vary with hub height according to the power law $\bar{U}(z) = \bar{U}(50) \cdot (z/50)^{0.14}$. Hub height is taken as equal to turbine diameter as before and the rated wind speed in all cases is 11.55 m/s. Turbine life is taken as 20 years and a discount rate of 10% is used.

It is seen that the model indicates that the optimum turbine diameter is just over 70 m – rather greater than that given by the cost model in the preceding section. This is to be expected in view of the reduced diameter exponents of the scaling rules for some components.

6.2.4 Machine size growth

During the 1980s and 1990s, the size of the largest turbines in commercial production doubled about every seven years. More recently the driver to increased diameters has undoubtedly been the extension of wind farm development offshore, where substantial fixed elements of support structure and undersea cable installation costs favour the deployment of much larger machines than on land. However, ‘Wind Energy – The Facts’ (2009) points out that the diameter of the largest commercially available wind turbine did not increase between 2004 and 2008 – a striking pause in the rapid growth in machine size hitherto.

The increasing popularity of 80 m and 90 m turbines for sites on land begs the question of why sizes larger than the apparent optimum are being chosen. Part of the answer may lie in imperfections of the cost models. However, a significant factor encouraging the selection of larger turbines is undoubtedly that their use enables better exploitation of sites of limited area.

For example, the total rated capacity that can be installed on a narrow ridge increases roughly linearly with turbine diameter, assuming (as is normally the case) that the minimum spacing permitted by the manufacturer is specified in terms of a fixed number of turbine diameters. Given that some site development costs, such as permitting and grid connection, do not vary significantly with wind farm rated capacity, there is always an incentive to maximise the installed capacity.

6.2.5 Gravity limitations

The simplified cost model described above was based on the assumption that blade design is governed solely by aerodynamic loads. However, as diameters increase, it is inevitable that edgewise moments due to blade self-weight will become increasingly important. For a family of blade designs derived from a baseline design by simply scaling the diameter and all other dimensions by the same amount, the blade root gravity moment will increase as the fourth power of diameter while the section modulus only increases by the third power. Although there will initially be some scope for catering for the increased gravity moment by redeployment of material closer to the leading and trailing edges, a limit on the practicable diameter for any particular blade material must eventually be reached.

6.3 Machine rating

The machine rating determines the wind speed (known as rated wind speed) at which rated power is reached. If the rating is too high for a given rotor diameter, the rated power will only be reached rarely, so the cost of the drive train and generator will not be justified by the energy yield. On the other hand, if the rating is reduced below the optimum then the cost of the rotor and its supporting structure will be excessive in relation to energy yield.

The investigation of the optimum relationship between rotor diameter and rated power can be carried out with the help of the cost modelling technique described in the previous section.

6.3.1 Simplified cost model for optimising machine rating in relation to diameter

The way in which the design of the various wind turbine components is influenced by changes in the rated speed is critically dependent on the nature of any accompanying changes in rotational speed. However, in view of the fact that the maximum rotational speed of land-based machines is generally restricted in order to limit noise emission (see Section 6.4), it is assumed here that the maximum tip speed is limited to 80 m/s, regardless of the rated wind speed.

The simplified cost model is applied to a pitch regulated, variable speed machine, as these are increasingly the turbine of choice. It is assumed that the machine is designed for optimum performance at a tip speed ratio of 8 and that it operates at this tip speed ratio up to a wind speed of 10 m/s, with the nominal tip speed remaining at 80 m/s at higher wind speeds.

Assuming that the blade planform and twist distribution are fixed, the annual energy yield can be calculated for a number of rated wind speeds, for a given annual mean wind speed and Weibull shape factor. The aim of the optimisation is to obtain the minimum cost of energy,

which requires knowledge of how the costs of the various turbine components would be affected by the rating change. Although, in theory, this could only be rigorously derived by carrying out a series of detailed turbine designs, in practice it is possible to obtain a useful indication of cost trends by identifying the parameters driving the design of each component category and investigating their dependence on the rated wind speed. If the cost split between various components is known for a baseline machine, these cost trends can then be applied to it in order to determine the optimum rating. In this case the cost shares given in Table 6.2 for the 70 m diameter, 1.5 MW NREL machine are used.

The manner in which the design of each of the major components is influenced by rated wind speed is set out below.

1. *Blade weight*: the following assumptions are made:

- The blade planform is constant.
- The blade design is governed by out-of-plane bending moments in fatigue.
- The out-of-plane bending moment fluctuations are proportional to the product of the wind speed fluctuation and the rotational speed (see Equation 5.25 in Section 5.7.5).
- The rotational speed is a function of prevailing wind speed, but is independent of rated wind speed as already stated.
- The blade skin thickness is independent of rated wind speed.

Hence the blade skin thickness and, therefore, the blade weight are unaffected by changes in the rated wind speed.

2. *Hub and pitch system weights*: it is assumed that each is proportional to the blade out-of-plane bending moments in fatigue – that is, independent of rated wind speed.
3. *Low speed shaft weight*: this is assumed to be governed by the shaft bending moment due to the cantilevered rotor and hub weights, which are unaffected by changes in rated wind speed.
4. *Gearbox and brake*: gearbox and brake design are governed by the rated torque, P/Ω . The rated power is proportional to the cube of the rated wind speed, and the maximum rotational speed is fixed, so the rated torque varies as the power rating. The weights of the gearbox and brake are therefore taken to be proportional to the rated power.
5. *Generator and variable speed electronics*: the design of the generator and the variable speed electronics are governed by rated power and the weight is assumed to be proportional to rated power in each case.
6. *Nacelle structure, yaw system, tower and foundation*: the design of these is governed principally by either extreme or fluctuating loads on the rotor, both of which are assumed to be independent of rated wind speed. The weights are therefore taken to be unaffected by rated wind speed.
7. *Grid connection*: the weight of cables, switchgear and transformers are assumed to be proportional to rated power.
8. *Controller, assembly and transport*: the costs of these items are taken as independent of rated speed.

Table 6.3 Percentage contribution of different components to machine cost for the 70 m dia, 1.5 MW NREL baseline machine, classified according to whether their cost varies with rated power or not

Components for which the weight/cost is independent of rated wind speed		Components for which the weight varies as rated power	
Component	Cost	Component	Cost
Blades	11.4%	Gearbox	11.6%
Hub and Spinner	3.6%	Generator	7.4%
Pitch bearings and mechanism	4.3%	Variable speed electronics	9%
Low speed shaft and bearings	2.5%	Brake & HS coupling	0.2%
Nacelle	8.9%	Internal cables and grid connection	14.2%
Yaw drive & bearing	1.5%		
Control System	2.7%		
Tower	12%		
Foundation	3.6%		
Assembly	3.2%		
Transport	3.9%		
TOTAL	57.6%	TOTAL	42.4%

The various components listed above are classified into two categories in Table 6.3, according to whether their weights are fixed or vary with the rated power. Also tabulated are the component costs as a percentage of the total for the baseline machine, together with the sum for each category.

Accordingly the following expression is obtained for machine cost as a function of the ratio of the rated power to that of the baseline machine, P_R/P_{RB} :

$$C_T = C_{TB} (0.576 + 0.424 (P_R/P_{RB})) \quad (6.5)$$

The capital component of the cost of energy is obtained by dividing the machine cost from Equation 6.5 by the discounted lifetime annual energy yield, which is calculated for each rated wind speed by combining the corresponding power curve with the Weibull distribution of wind speeds. This exercise has been carried out using the 70 m diameter, 1.5 MW pitch regulated, variable speed NREL machine as baseline, assuming an annual mean wind speed of 7.5 m/s, and taking the rated wind speed of the baseline machine as 11.55 m/s. The results are presented in Figure 6.4, which indicates that the optimum machine rating is very close to the 1.5 MW baseline. The variation in cost of energy with rated power on either side of the optimum is very small; with the maximum increase in the cost of energy over the range 1100–2000 kW being only 3%.

6.3.2 Relationship between optimum rated wind speed and annual mean

The optimum power rating is of course heavily dependent on the annual mean wind speed, U_{ave} . The optimum rated wind speed, U_{R0} for the above 70 m diameter pitch regulated machine

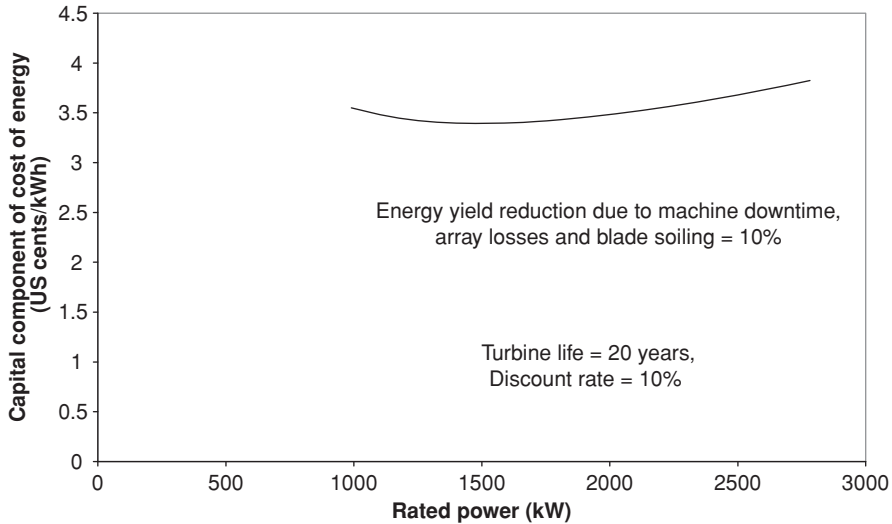


Figure 6.4 Variation in cost of energy with rated power for a 70 m diameter, pitch regulated variable speed machine for an annual mean wind speed of 7.5 m/s, based on simplified cost model

is given for a range of annual mean wind speeds in Table 6.4. The ratio U_{Ro}/U_{ave} is in the range 1.6–1.4, decreasing with increasing wind speed.

A similar exercise can be carried out to determine the optimum rated power of a stall regulated machine, and would yield similar results. However, because stall regulated machines reach rated power at a substantially higher wind speed than pitch regulated machines of the same rating, the U_{Ro}/U_{ave} ratio for stall regulated machines is typically about 2.

6.3.3 Specific power of production machines

It is instructive to investigate the relationship between rated power and swept area for production machines, and these quantities are plotted against each other in Figure 6.5 for

Table 6.4 Variation of optimum rated wind speed with annual mean for pitch regulated machines

Annual mean wind speed, U_{ave} (m/s)	Optimum rated wind speed, U_{Ro} (m/s)	Ratio U_{Ro}/U_{ave}	Optimum rated power (kW)	Specific power, defined as rated power per unit swept area (kW/sqm)	Cost index, with cost of energy for amws of 7.5 m/s taken as 100
7	11.1	1.59	1340	349	113
7.5	11.5	1.54	1495	388	100
8	11.9	1.49	1535	425	90
8.5	12.3	1.445	1770	460	81
9	12.65	1.41	1905	495	75

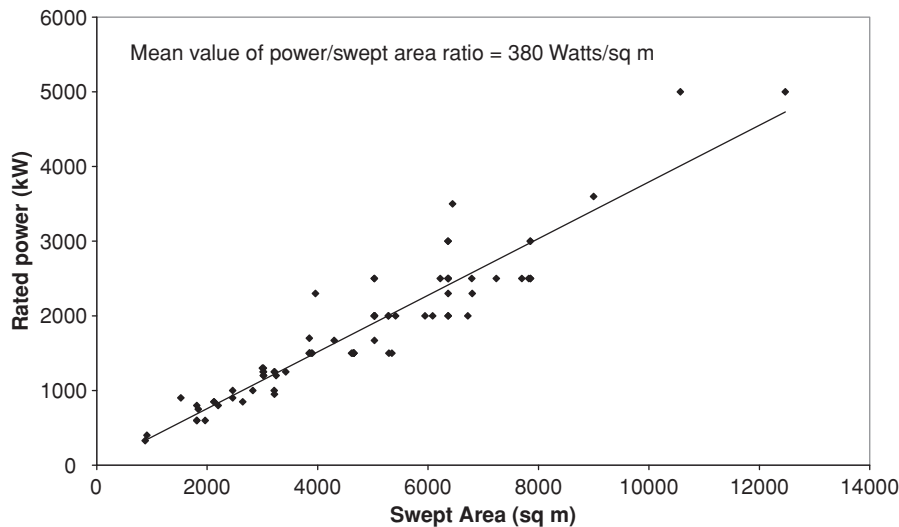


Figure 6.5 Rated power versus swept area for turbines in production in 2008

79 machines in production in 2008. Although different machines will have been designed for different annual mean wind speeds, the degree of scatter is not large, and a clear trend is apparent, with the line of best fit being close to a straight line passing through the origin. The mean specific power, defined as rated power divided by swept area, is 380 Watts/sq m for the 79 machines – close to the optimum value in Table 6.3 for an annual wind speed of 7.5 m/s.

6.4 Rotational speed

The aim of the wind turbine designer is the production of energy at minimum cost, subject to constraints imposed by environmental impact considerations. However, blade designs optimised for a number of different rotational speeds but the same rated power produce substantially the same energy yield, so the choice of rotational speed is based on machine cost rather than energy yield.

One of the key cost drivers is the rotor torque at rated power, as this is the main determinant of the drive train cost. For a given tip radius and machine rating, the rotor torque is inversely proportional to rotational speed, which argues for the adoption of a high rotational speed. However, increasing the rotational speed has adverse effects on the rotor design, which are explored in the following sections.

6.4.1 Ideal relationship between rotational speed and solidity

Equation 3.69a in Section 3.7.2 gives the chord distribution of a blade optimised to give maximum power at a particular tip speed ratio in terms of the lift coefficient, ignoring drag

and tip loss:

$$\sigma_r \lambda C_l = \frac{\frac{8}{9}}{\sqrt{\left(1 - \frac{1}{3}\right)^2 + \lambda^2 \mu^2 \left[1 + \frac{2}{9\lambda^2 \mu^2}\right]^2}} \quad (3.69a)$$

where λ is the tip speed ratio, σ_r is the solidity and $\mu = r/R$. Over the outboard half of the blade, which produces the bulk of the power, the local speed ratio, $\lambda\mu$, will normally be large enough to enable the denominator to be approximated as $\lambda\mu$, giving:

$$\sigma_r \lambda C_l = \frac{Nc(\mu)}{2\pi R} \lambda C_l = \frac{8}{9\lambda\mu} \quad (6.6)$$

where N is the number of blades.

After rearrangement, this gives

$$c(\mu) \left(\frac{\Omega R}{U_\infty}\right)^2 = \frac{16\pi R}{9C_l N} \cdot \frac{1}{\mu} \quad (6.7)$$

Hence it can be seen that, for a family of designs optimised for different rotational speeds at the same wind speed, the blade chord at a particular radius is inversely proportional to the square of the rotational speed, assuming that N and R are fixed and the lift coefficient is maintained at a constant value by altering the local blade pitch to maintain a constant angle of attack.

Note that Equation 6.7 does not apply if energy yield is optimised over the full range of operating wind speeds for a fixed speed pitch regulated machine. In this case, it has been demonstrated that the blade chord at a particular radius is approximately inversely proportional to rotational speed rather than to the square of it (Jamieson and Brown, 1992).

6.4.2 Influence of rotational speed on blade weight

The effect of rotational speed on blade weight can be explored with reference to the family of blade designs just described. As in Section 6.3.1, it is assumed that the blade design is governed by out-of-plane bending moments in fatigue and that the moment fluctuations are proportional to the product of the wind speed fluctuation, the rotational speed and the chord scaling factor (see Equation 5.25 in Section 5.7.5). By Equation 6.7 the chord scaling factor is inversely proportional to the square of the rotational speed, so the moment fluctuations simply vary inversely as the rotational speed.

The thickness to chord ratios at each radius are assumed to be unaffected by the chord scaling, so the blade section modulus for out-of-plane bending at a given radius is proportional to the product of the blade shell skin thickness, $w(r)$, and the square of the local chord. Thus,

$$Z(r) \propto w(r) \cdot (c(r))^2 \propto w(r)/\Omega^4 \quad (6.8)$$

In order to maintain the fatigue stress ranges at the same level, we require the blade section modulus, $Z(r)$, to vary as the moment fluctuations, which, as shown above, vary inversely as rotational speed. Thus,

$$Z(r) \propto 1/\Omega \quad \text{so} \quad w(r)/\Omega^4 \propto 1/\Omega \quad \text{and} \quad w(r) \propto \Omega^3 \quad (6.9)$$

Blade weight is proportional to the skin thickness times chord, and thus varies as rotational speed.

6.4.3 Optimum rotational speed

On the basis of the assumptions of Section 6.4.2 (which will by no means always apply), blade weight increases in proportion to rotational speed. However, the blade out-of-plane fatigue loads, which may govern the design of the nacelle structure and tower, vary *inversely* as the rotational speed. It is therefore likely that, as rotational speed is increased, there will be a trade-off between reducing costs of the drive train, nacelle structure and tower on the one hand and increasing rotor cost on the other, which will determine the optimum value.

6.4.4 Noise constraint on rotational speed

The aerodynamic noise generated by a wind turbine is approximately proportional to the fifth power of the tip speed. It is therefore highly desirable to restrict turbine rotational speed, especially when the wind speed and, therefore, ambient noise levels are low. Consequently manufacturers of turbines to be deployed at normal sites on land generally limit the tip speed of fixed speed machines to about 65 m/s.

In the case of variable speed machines the maximum tip speed is usually significantly higher – typically in the range 70–85 m/s, but of course these tip speeds are only reached at higher wind speeds, when ambient noise levels are higher also.

Offshore machines are not subject to the noise constraints on maximum tip speeds that apply on land, so machines designed primarily for offshore siting typically have somewhat higher tip speeds. However, Jamieson (2009) investigated a more substantial increase in tip speed to 120 m/s in order to realise savings in drive train cost, and proposed a down-wind configuration in order to avoid the risk of tower strike by the resulting flexible, low solidity blades.

6.4.5 Visual considerations

There is a consensus that turbines are more disturbing to look at the faster they rotate.

6.5 Number of blades

6.5.1 Overview

European windmills traditionally had four sails, perhaps because pre-industrial techniques for attaching the sail stocks to the shaft lent themselves to a cruciform arrangement in which the stocks for opposite sails formed a continuous wooden beam. By contrast the vast majority of horizontal axis wind turbines manufactured today have either two or three blades, although at

least one manufacturer used to specialise in one-bladed machines. As the latter are now only of theoretical or historical interest, consideration of them will be restricted to Section 6.5.7, and the rest of Section 6.5 will concentrate on two- and three-bladed machines.

In comparing the relative merits of machines with differing numbers of blades, the following factors need to be considered:

- performance;
- loads;
- cost of rotor;
- impact on drive train cost;
- noise emission; and
- visual appearance.

Some of these factors are strongly influenced by rotational speed and rotor solidity, and the ideal relationship between these parameters and the number of blades is considered in the next section. Section 6.5.3 investigates alternative two-bladed derivatives of a realistic three-bladed variable speed baseline design and compares their relative energy yields and notional costs. Section 6.5.4 reviews the differences in loading imposed by two- and three-bladed rotors on the supporting structure, and Section 6.5.5 considers the constraint on rotational speed imposed by noise emission. Visual appearance is considered briefly in Section 6.5.6.

6.5.2 Ideal relationship between number of blades, rotational speed and solidity

The effect of the number of blades on the blade chord and rotational speed of a machine optimised for a particular wind speed is given by Equation 6.7:

$$Nc(\mu) \left(\frac{\Omega R}{U_\infty} \right)^2 = \frac{16\pi R}{9C_l} \cdot \frac{1}{\mu}$$

Hence it can be seen that, if the number of blades is reduced from three to two, increasing the chord by 50% or the rotational speed by 22.5% are two of the options for preserving optimised operation at the selected wind speed. (It is assumed that the lift coefficient is maintained at a constant value by altering the local blade pitch to maintain a constant angle of attack.)

6.5.3 Some performance and cost comparisons

Clear-cut comparisons between two- and three-bladed machines are notoriously difficult because of the impossibility of establishing equivalent designs. Conceptually, the simplest option is to increase the chord by 50% at all radii and leave everything else – including rotational speed – unchanged. In the absence of tip-loss, the induction factors, and hence the annual energy yield, remain the same, but when tip-loss is included, the annual energy yield of a stall regulated machine drops by about 2.5%. However, retention of the same rotor solidity largely negates one of the main benefits of reducing the number of blades, namely reduction in rotor cost, and so this option will not be pursued further. Instead it is proposed to

take a realistic blade design for a three-bladed machine and look at the performance and cost implications of using the same blade on a two-bladed machine rotating at different speed.

Performance comparisons are affected both by the power rating in relation to swept area (Section 6.3) and by the aerofoil data used. In this case a 70 m diameter, 1.5 MW, pitch-regulated variable speed three-bladed turbine operating at a tip speed ratio of 8 for wind speeds below 10 m/s and constant tip speed above is adopted as the baseline machine. The maximum rotational speed is thus $80/35 = 2.29$ rad/s or 21.8 rpm. The blade planform and thickness distribution are scaled down from those for the T40 blade given in Figure 5.4(a). Empirical 3D aerofoil data for a LM 19.0 blade is used (see Figure 5.9), with maximum lift coefficient increasing from blade tip to blade root, as this results in more accurate power curve predictions. The data is taken from the Risø publication ‘Prediction of dynamic loads and induced vibrations in stall’ by Petersen, Madsen *et al.* (1998). The blade twist distribution is set to give maximum energy yield at a site where the annual mean wind speed is 7.5 m/s, resulting in an annual energy yield at 100% availability of 4937 MWh. The performance curve for the baseline turbine is shown in Figure 6.6.

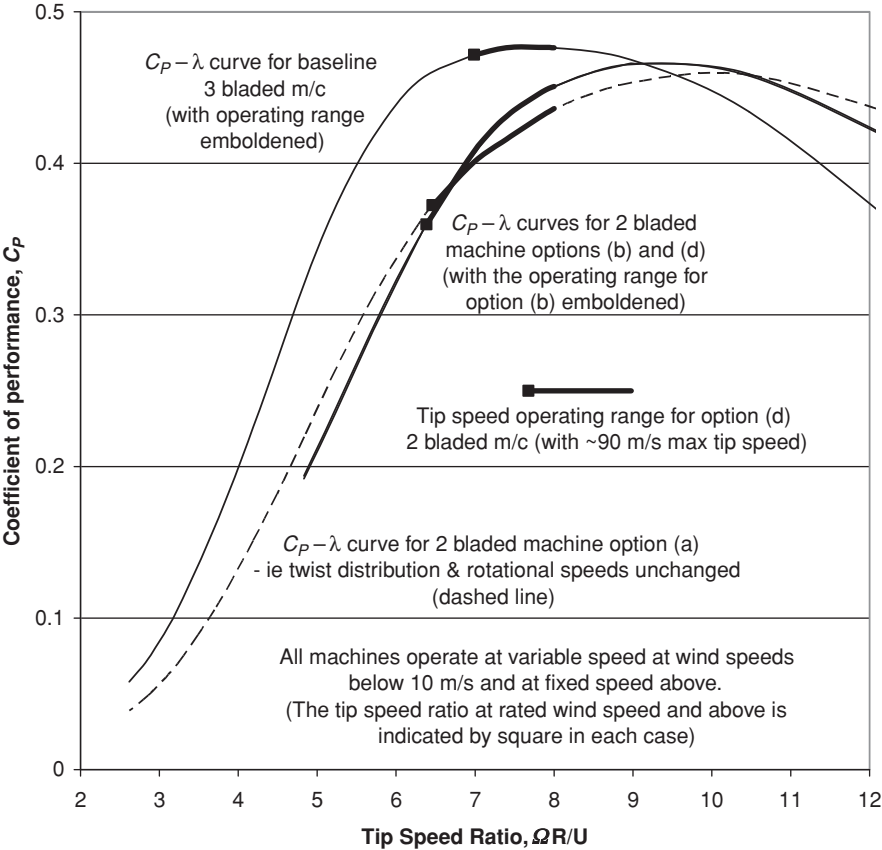


Figure 6.6 Comparison of $C_p - \lambda$ curves for three-bladed baseline machine and two-bladed options (a), (b) and (d)

Four options for a corresponding 70 m diameter pitch-regulated, variable speed, two bladed design at a site with the same annual mean wind speed are examined and the notional energy costs compared with that for the baseline three-bladed machine. The costs of the two bladed design options in relation to the baseline three-bladed machine are considered with reference to changes in the weights of the components, using the cost shares for the NREL baseline machine given in Table 6.2.

Blade design is assumed to be governed by out-of-plane fatigue bending moments, with the moment fluctuations increasing in proportion to rotational speed (see Equation 5.25 in Section 5.7.5). Accordingly the blade weight is assumed to increase linearly with rotational speed, but the total blade weight for the two-bladed machine at the baseline rotational speed is, of course, reduced by one third. The weights of the hub, pitch system, shaft and yaw system are also assumed to increase with rotational speed, but no account is taken of the increased loads on these components for a fixed-hub, two-bladed machine.

It is assumed initially that the design of the nacelle structure is fatigue driven and governed by the fluctuating moment on the nacelle due to differential blade out-of-plane root bending moment, which increases with rotational speed. Tower design is also assumed to be governed by fatigue in the first instance, so tower weight is similarly taken as proportional to rotational speed. The cyclic thrust loads on the rotor due to turbulence are virtually the same for two- and three-bladed machines rotating at the same speed if the blade planforms are the same, so the tower cost element at the baseline rotational speed is left unchanged.

The weights of the gearbox and brake are taken to be proportional to the rated torque, P_R/Ω , while those of the generator, the variable speed electronics and of the cables and equipment forming the grid connection are taken as proportional to rated power, P_R .

The various components are classified into different categories according to the way in which their costs vary with rotational speed and rated power in Table 6.5. Also tabulated are the two-bladed machine component costs as a percentage of the total for the baseline three-bladed machine, together with the sum for each category.

The following expression is obtained for machine cost as a function of rotational speed and rated power:

$$C_T = C_{TB}(0.134 + 0.4047\{\Omega/\Omega_B\} + 0.118\{\Omega_B/\Omega\}(P_R/P_{RB}) + 0.305(P_R/P_{RB})) \quad (6.10)$$

Here, P_{RB} and Ω_B are the baseline values of rated power and maximum nominal rotational speed, 1500 kW and 21.8 rpm respectively. The three design options can now be examined:

(a) *Planform, twist and rotational speed unchanged from baseline*

The reduction in the number of blades reduces the coefficient of performance at the tip speed ratio of 8, which applies at wind speeds less than 10 m/s, by 8.4%, but the energy yield reduction is less at 6.3%. The corresponding $C_P - \lambda$ curve is shown over the full range of tip speed ratios by the dashed line on Figure 6.6, with the much more limited operating range shown emboldened, and it is evident that the optimum tip speed ratio has increased from about 7.7 for the baseline three-bladed machine to about 10. Clearly the limit on the upper tip speed of 80 m/s imposes a significant penalty on performance.

Combination of the reduced energy yield with the machine capital cost saving of 3.8% due to the elimination of one blade results in an increased cost of energy of 2.6%.

Table 6.5 Contribution of different components to the cost of a two-bladed machine (expressed as percentages of three-bladed baseline machine cost) and classified according to the relationship assumed between the component cost and rotational speed/rated torque/rated power

Components for which the weight/cost is independent of rated power or rotational speed		Components for which the weight/cost varies as rotational speed, Ω		Components for which the weight/cost varies as rated torque, P_R/Ω		Components for which the weight/cost varies as rated power, P_R	
Component	Cost	Component	Cost	Component	Cost	Component	Cost
Foundation	3.6%	Blades	7.7%	Gearbox	11.6%	Generator	7.4%
Controller	2.7%	Hub	3.6%	Brake	0.2%	Variable	9.0%
Assembly	3.2%	Pitch	4.3%	system		speed	
Transport	3.9%	system				electronics	
		Low speed	2.5%			Grid	14.1%
		shaft & bearings				connection	
		Nacelle	8.9%				
		Yaw system	1.5%				
		Tower	12.0%				
TOTAL	13.4%	TOTAL	40.5%	TOTAL	11.8%	TOTAL	30.5%

(b) *As option (a), but twist distribution re-optimised*

Re-optimisation of the twist distribution reduces the energy yield penalty compared with the baseline three-bladed machine to 4.6%. As a result the increase in the cost of energy is reduced to 0.8%. The corresponding $C_P - \lambda$ curve is shown over the full range of tip speed ratios in Figure 6.6, with the much more limited operating range shown emboldened.

(c) *Tip speed ratio schedule scaled in conjunction with twist distribution optimisation to obtain minimum cost of energy based on machine cost function of Equation 6.10 (with tower and nacelle cost increasing with maximum rotational speed)*

The tip speed at each wind speed can be scaled up by the same ratio so as to produce maximum energy yield, with simultaneous re-optimisation of the twist distribution. This enables the energy yield penalty compared with the baseline three-bladed machine to be reduced to only 1.3%, for a tip speed scaling factor of 1.21. However, application of Equation 6.10 results in a capital cost increase of 2.5% relative to the baseline machine, resulting in a cost of energy increase – again relative to the baseline machine – of 3.8%.

Minimisation of the cost of energy, on the other hand, leads to a much lower tip speed scaling factor of 1.035. The energy yield penalty and capital cost reduction relative to the baseline machine are now 3.3% and 2.8% respectively, resulting in an increase in the cost of energy of only 0.5%.

- (d) *Tip speed ratio schedule scaled in conjunction with twist distribution optimisation to obtain minimum cost of energy based on machine cost function of Equation 6.10 modified so that tower and nacelle cost are fixed*

It is apparent that, with the weight of nacelle and tower assumed to increase in proportion to rotational speed the two-bladed variant c) considered above results in a small increase in the cost of energy relative to the three-bladed machine. However, if the weight of the nacelle structure is driven by the *size* of the enclosure required to accommodate the gearbox, generator and other equipment, rather than by fatigue loading, it will be constant for machines of fixed rating. If it is also assumed that the tower design is governed by extreme loads rather than fatigue loads, the expression for machine cost as a function of rotational speed and rated power becomes:

$$C_T = C_{TB}(0.254 + 0.2847\{\Omega/\Omega_B\} + 0.118\{\Omega_B/\Omega\}(P_R/P_{RB}) + 0.305(P_R/P_{RB})) \quad (6.11)$$

The tip speed scaling factor resulting in the minimum cost of energy then increases to 1.12 and the cost of energy reduces to 1.1% below that of the baseline machine. As shown in Figure 6.6, the corresponding $C_P - \lambda$ curve over the full range of tip speed ratios is almost identical to that for option (b), but the tip speed ratio operating range differs.

The above results are summarised in Table 6.6.

The results shown in Table 6.6 indicate that two-bladed, rigid hub machines have the potential to yield marginal cost benefits vis-à-vis three-bladed machines, if nacelle and tower design are not impacted by increases in rotational speed. However, the results should be treated with caution, because no account has been taken of increased component costs due to the increased loadings on the hub, low-speed shaft, yaw drive and nacelle inherent in a rigid hub two-bladed turbine. (Loads on rigid hub two-bladed machines are compared with those on three-bladed machines in more detail in the next section.)

The loadings on the nacelle of a two-bladed machine can be reduced significantly by the introduction of a teeter hinge between the rotor and the low speed shaft, with consequent potential cost benefits. The hinge eliminates the transfer of out-of-plane aerodynamic moments from the rotor to the low speed shaft, resulting in large reductions in the operational loadings on the shaft, nacelle and yaw drive. The dependence of these loads on rotational speed is also largely removed, with the result that the optimum rotational speed for a two bladed machine in energy cost terms is increased, approaching the value giving maximum energy yield.

Although teetering provides scope for significant cost savings on the shaft, nacelle and yaw drive (which account for nearly 20% of the baseline machine cost), these savings are offset by the additional costs associated with the teeter hinge and teeter restraint system.

6.5.4 Effect of number of blades on loads

Moment loadings on the low speed shaft and nacelle structure from three-bladed and rigid-hub two-bladed machines were examined in Sections 5.10 and 5.11, and are compared in Table 6.7 below for machines of the same diameter and rotational speed. The stochastic loading comparison is based on a turbulence length scale to rotor diameter ratio of 1.84.

Table 6.6 Comparison of two-bladed design variants on a 70 m diameter, 1.5 MW three-bladed variable speed, pitch regulated baseline machine, using blades of same planform and thickness/chord ratio distribution

Two-bladed 70 m diameter, 1.5 MW, machine design options	Maximum nominal speed and tip speed	Annual energy yield (MWh)	Reduction in annual energy yield cf baseline m/c	Reduction in overall machine cost cf three-bladed baseline	Increase/reduction in cost of energy
a) Same blade and tip speed schedule	21.8 rpm 80 m/s	4625	6.3%	3.8%	+2.6%
b) As above, but with optimisation of blade twist distribution	21.8 rpm 80 m/s	4709	4.6%	3.8%	+0.8%
c) Tip speeds & twist distribution varied to give minimum C of E as per Equation 6.10	22.6 rpm 83 m/s	4775	3.3%	2.8%	+0.5%
d) Tip speeds & twist distribution varied to give minimum C of E assuming nacelle and tower costs fixed – i.e. using Equation 6.11	24.5 rpm 90 m/s	4857	1.6%	2.7%	–1.1%

Table 6.7 Comparison of loads on shaft and nacelle for three-bladed and rigid-hub two-bladed machines (ψ is blade azimuth)

Location of moment loading	Deterministic loading arising from wind shear and/or yaw misalignment, in terms of blade root out-of-plane bending moment amplitude, M_o		Stochastic loading
	Three-bladed machine	Rigid-hub two-bladed machine	% increase for rigid-hub two-bladed machine compared with three-bladed m/c
Shaft bending moment amplitude	$1.5 M_o$	$2 M_o$	22%
Nacelle nodding moment	$1.5 M_o$	$M_o(1 + \cos 2\psi)$	22%
Nacelle yaw moment	Zero	$M_o \sin 2\psi$	22%

It is seen that loadings from a rigid-hub two-bladed rotor are significantly larger than from a three-bladed rotor. However, in most two-bladed machine designs, the rotor is allowed to teeter instead of being rigidly mounted, with the result that aerodynamic moments on the shaft and nacelle structure quoted in Table 6.7 are eliminated, and the blade out-of-plane root bending moments reduced. The benefits and drawbacks of teetering the rotor are examined in Section 6.6.

The rotor thrust variations at blade passing frequency due to stochastic loading, which are a dominant factor in tower fatigue design, are very similar for two- and three-bladed machines rotating at the same speed. However, two-bladed machines usually rotate faster than three-bladed machines of the same diameter, so the cyclic rotor thrust variations are higher.

6.5.5 Noise constraint on rotational speed

As noted in Section 6.5.3, there may be significant cost benefits to be gained from a two-bladed design with increased rotational speeds, because, in addition to the blade saving, the cost of the whole of the drive train is reduced because of the reduced torque. However, as noted in Section 6.4.4, it is normal to restrict the tip speeds of fixed speed and variable speed machines to about 65 m/s or 85 m/s respectively, in order to limit aerodynamic noise emission. At 80 m/s, the tip speed of the baseline machine discussed in Section 6.5.3 is within this limit, but the tip speed of option d) of 90 m/s would be less likely to be acceptable, except at remote sites or offshore. This subject is considered further in Section 6.9.

6.5.6 Visual appearance

Although the assessment of visual appearance is essentially subjective, there is an emerging consensus that three-bladed machines are more restful to look at than two-bladed ones. One possible reason for this is that the apparent ‘bulk’ of a three-bladed machine changes only slightly over time, whereas a two-bladed machine appears to contract down to a one-dimensional line element, when the rotor is vertical, twice per revolution. A secondary factor is that two-bladed machines generally rotate faster, which an observer can also find more disturbing.

6.5.7 Single-bladed turbines

Apart from the saving in rotor cost itself, the single-bladed turbine concept is an attractive one because of the reduction in drive train cost realisable through increased rotational speed (Section 6.5.2). An obvious disadvantage is the resulting increased noise emission resulting from the faster rotation, but this would not be an issue offshore. Another consideration is the reduced yield due to increased tip loss. For example, a single-bladed version of the 70 m diameter, 1.5 MW pitch regulated, variable speed three-bladed baseline design considered in Section 6.5.3, with the rotational speed at each wind speed scaled up by $\sqrt{3}$ in accordance with Equation 6.7 and with the twist distribution reoptimised to give maximum energy yield, would produce an annual energy output 5.5% less than the baseline machine.

The single blade must be counterweighted to eliminate torque fluctuations and any whirling tendency due to centrifugal loads. Furthermore, as a rigid hub would expose the nacelle to very large nodding and yawing moments in comparison with two- or three-bladed machines, it is customary to mount the rotor on a teeter hinge, so that the unbalanced aerodynamic

out-of-plane moment can be resisted by a centrifugal couple, thereby reducing the hub moment. However, the teeter motion of the blade is significantly greater than that of a two-bladed machine, so it is normal to mount the rotor downwind. Morgan (1994) reports that particular difficulties have been encountered in predicting teeter excursions after grid loss and emergency stops, leading to excessive risk of teeter stop impacts.

6.6 Teetering

6.6.1 Load relief benefits

Two-bladed rotors are often mounted on a teeter hinge – with hinge axis perpendicular to the shaft axis, but not necessarily perpendicular to the longitudinal axis of the blades – in order to prevent differential blade root out-of-plane bending moments arising during operation. Instead, differential aerodynamic loads on the two blades result in rotor angular acceleration about the teeter axis, with large teeter excursions being prevented by the restoring moment generated by centrifugal forces, as described in Section 5.8.8. However, when the machine is shutdown, the centrifugal restoring moment is absent, so differential blade loading will cause the rotor to teeter until it reaches the teeter end stops which need to be suitably buffered. Consequently the teeter hinge is unlikely to provide any amelioration of extreme blade root out-of-plane moments when the machine is shutdown.

The load relief afforded by the teeter hinge benefits the main structural elements in the load path to the ground in varying degrees, as outlined below:

- (a) *Blade*: The main benefit is the elimination of the cyclic variations in out-of-plane bending moment due to yaw (Figure 5.10), shaft tilt, wind shear (Figure 5.11) and tower shadow (Figure 5.14). By contrast, there is only a small reduction in blade root out-of-plane bending moment due to stochastic loadings – see the example in Section 5.8.8, where an 11% reduction is quoted. Thus, teetering results in a large overall reduction in out-of-plane fatigue loading, although the significance of this will be tempered by the influence of the unaltered edgewise gravity moment.
- (b) *Low speed shaft*: Low speed shaft design is governed by fatigue loading, which is normally dominated by the cyclic gravity moment due to the cantilevered rotor mass. On a rigid hub machine, the shaft moment Damage Equivalent Load or DEL (defined in Section 5.12.6) due to deterministic and stochastic rotor out-of-plane loadings combined can be of similar magnitude, so the insertion of a teeter hinge can produce a substantial reduction in overall shaft moment DEL. It should be noted, however, that the cyclic shaft moment due to wind shear *relieves* that due to gravity on a rigid hub machine, so teetering is not beneficial in respect of this load component.
A rough estimate of the overall shaft moment DEL on a rigid hub machine, excluding yaw error and tower shadow effects, can be obtained by taking the square root of the sum of the squares of the shaft moment DEL due to stochastic loads and that due to the combined cyclic loads due to gravity, wind shear and shaft tilt.
- (c) *Nacelle structure*: The provision of a teeter hinge should eliminate nodding and yawing moments on the nacelle completely during operation, leaving only rotor torque, thrust and in-plane loadings. This will benefit the fatigue design of the nacelle structure considerably, but not the extreme load design, for the reasons already explained.

- (d) *Yaw bearing and yaw drive*: Rigid hub machines experience severe yaw moments due to both deterministic and stochastic loads, which were underestimated on many early designs. The introduction of a teeter hinge dramatically reduces yaw moments during operation by eliminating rotor out-of-plane moments on the hub, but yaw moments due to in-plane loads on the rotor still remain.

The relative magnitude of the yaw moments due to in-plane as opposed to out-of-plane loads on a rigid hub rotor can be appreciated by comparing the effect of wind speed fluctuation, u , on the in-plane and out-of-plane loads on a blade element. Assuming that the blade is not stalled and that ϕ is small, the in-plane load per unit length is, from Equation 5.131a, given approximately by:

$$-F_Y = \left(\frac{1}{2} \rho \Omega \frac{dC_l}{d\alpha} \right) c(r) \cdot r \cdot u \left[\frac{C_l}{dC_l/d\alpha} + \sin \phi \right] \quad (6.12)$$

Whereas the out-of-plane load per unit length is, from Equation 5.25 approximately

$$F_X = \left(\frac{1}{2} \rho \Omega \frac{dC_l}{d\alpha} \right) c(r) \cdot r \cdot u \quad (6.13)$$

Defining the distance between the hub centre and the tower centre-line as e , it is seen that the yaw moment due to the in-plane rotor load is $(e/r)[C_l/(dC_l/d\alpha) + \sin \phi]$ times the maximum yaw moment due to out-of-plane load. As e is typically about one tenth of the tip radius, it is seen that the yaw moments due to in-plane loads are at least an order of magnitude smaller than those due to out-of-plane moments, so that the introduction of the teeter hinge results in a very significant reduction.

- (e) *Tower*: The fatigue loadings due to the M_Y moment and M_Z torque will clearly be significantly reduced at the top of the tower if the rotor is teetered, but the effect will be negligible towards the base where thrust loads dominate the moments.

6.6.2 Limitation of large excursions

Some limitation on teeter excursions has to be provided, if only to prevent collision between the blade and the tower. If the teeter hinge is located close to the axis of the blades, with the low speed shaft passing through an aperture in the wall of the hub shell (see Figure 6.7), then the maximum teeter excursion is governed by the size of the aperture.

The teeter response to deterministic and stochastic loads is considered in Section 5.8.8. Although it is evident that a permitted teeter angle range of the order of $+5^\circ$ or -5° will accommodate the vast majority of teeter excursions during normal operation, it is usually impracticable to accommodate the largest that can occur. Hence, in order to minimise the occurrence of metal-to-metal impacts on the teeter end stops, buffers incorporating spring and/or damping elements normally have to be fitted. These also perform an important role in limiting the much larger teeter excursions that would otherwise arise during start-up and shut-down, when the centrifugal restoring moment is reduced.

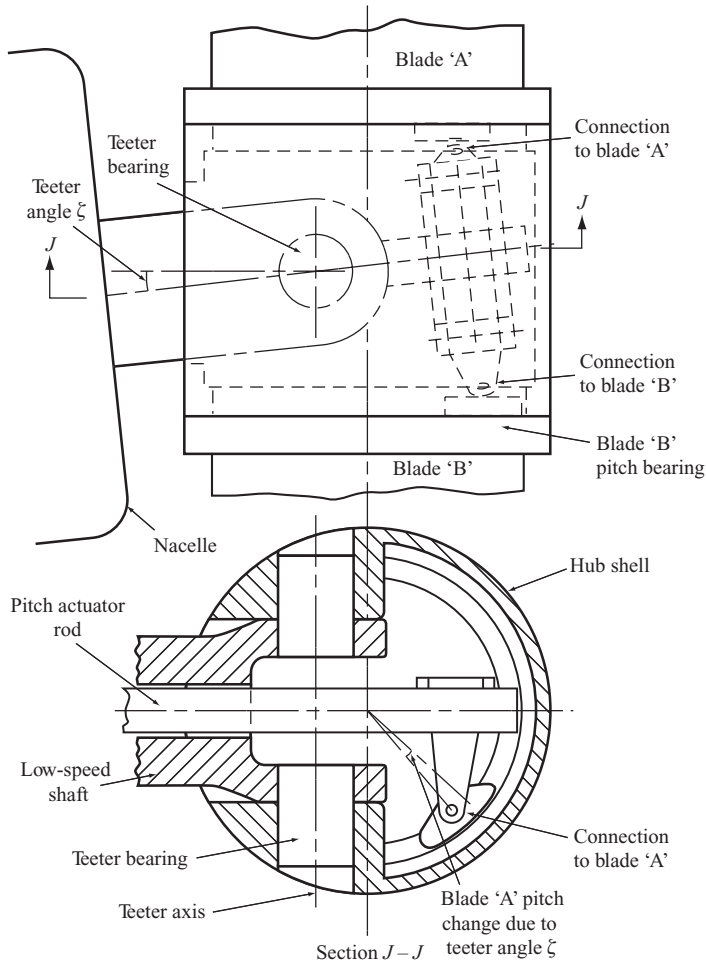


Figure 6.7 Pitch-teeter coupling

6.6.3 Pitch-teeter coupling

As described in Section 5.8.8, the magnitude of teeter excursions can be reduced by coupling blade pitch to teeter angle, in order to generate an aerodynamic restoring moment proportional to the teeter angle. This can be done simply by setting the teeter hinge at an angle, known as the Delta 3 angle, to the perpendicular to the rotor axis. Alternatively, on pitch-controlled machines, pitch-teeter coupling can be introduced by actuating the blade pitch by the fore-aft motion of a rod passing through a hollow low speed shaft. See Figure 6.7.

6.6.4 Teeter stability on stall-regulated machines

At first sight, it might be thought that the teeter motion of a stalled rotor would be unstable because of negative damping resulting from the negative slope of the $C_l - \alpha$ curve post-stall.

However, two-dimensional aerodynamic theory is a poor predictor of post-stall behaviour, and it has proved possible to design teetered rotors that are stable in practice, such as the Gamma 60 (Falchetta *et al.*, 1996) and Nordic 1000 (Engstrom *et al.*, 1997). The concept is explored in detail in investigations by Armstrong and Hancock (1991) and Rawlinson-Smith (1994).

6.7 Power control

6.7.1 Passive stall control

The simplest form of power control is passive stall control, which makes use of the post-stall reduction in lift coefficient and associated increase in drag coefficient to place a ceiling on output power as wind speed increases, without the need for any changes in blade geometry. The fixed blade pitch is chosen so that the turbine reaches its maximum or rated power at the desired wind speed.

Stall regulated machines suffer from the disadvantage of uncertainties in aerodynamic behaviour post-stall which can result in inaccurate prediction of power levels and blade loadings at rated wind speed and above. These aspects are considered in greater detail in Section 3.12.3.

6.7.2 Active pitch control

Active pitch control achieves power limitation above rated wind speed by rotating all or part of each blade about its axis in the direction which reduces the angle of attack and hence the lift coefficient – a process known as blade feathering. The main benefits of active pitch control are increased energy capture, the aerodynamic braking facility it provides and the reduced extreme loads on the turbine when shut-down. See also Sections 3.13 and 8.2.1.

The pitch change system has to act rapidly – that is, to give pitch change rates of 5° per second or better – in order to limit power excursions due to gusts enveloping the whole rotor to an acceptable value. However, it is not normally found practicable to smooth the cyclic power fluctuations at blade passing frequency due to blades successively slicing through a localised gust (Section 5.7.5), with the result that the large power swings of up to about 100% can sometimes occur in the case of fixed speed machines.

The extra energy obtainable with pitch control is not all that large. A pitch-regulated machine with the same power rating as a stall-regulated machine, utilising the same blades and rotating at the same speed will operate at a larger pitch angle below rated wind speed than the stall-regulated machine, in order to reduce the angle of attack and hence increase the power output at wind speeds approaching rated. For example, if the 1500 kW, 70 m diameter variable speed machine described in Section 6.5.3 were operated at a fixed speed of 17.1 rpm (corresponding to a tip speed of 62.8 m/s), a pitch regulated version would produce about 2.7% more energy than a stall regulated version for a 7.5 m/s annual mean wind speed, assuming optimisation of the blade twist distribution in each case. The power curves of the two fixed speed machines are compared in Figure 6.8. Also shown is the power curve for the variable speed pitch regulated machine described in Section 6.5.3. This would produce 5% more energy than the fixed speed, pitch regulated machine. Note that the knees in the power

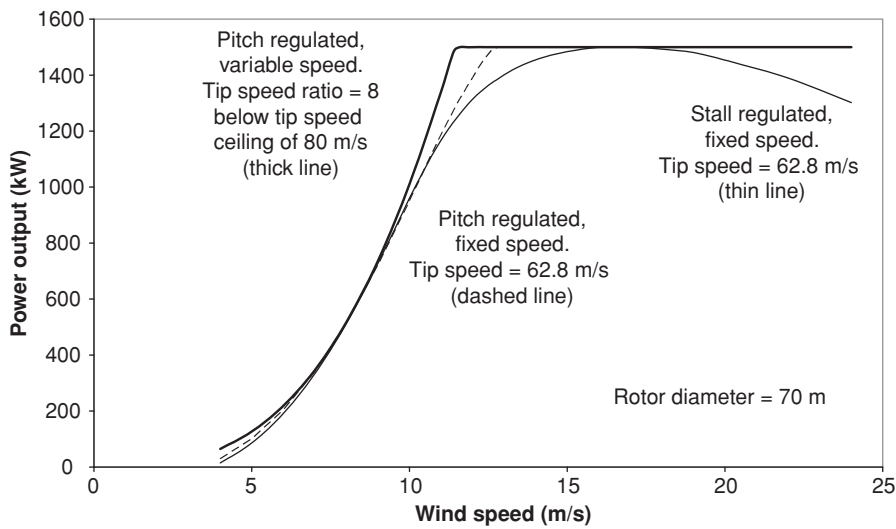


Figure 6.8 Comparison of power curves for (i) stall regulated, fixed speed; (ii) pitch regulated, fixed speed; and (iii) pitch regulated, variable speed 1.5 MW rated machines

curves of the pitch regulated machines at rated speed will be more rounded in practice because the pitch control will not keep pace with the higher frequency components of turbulence.

Figure 6.9 shows a family of power curves for a range of positive pitch angles for the 1500 kW, 70 m diameter pitch controlled machine rotating at 17.1 rpm. The intersections of these curves with the 1500 kW ordinate define the relationship between steady wind speed

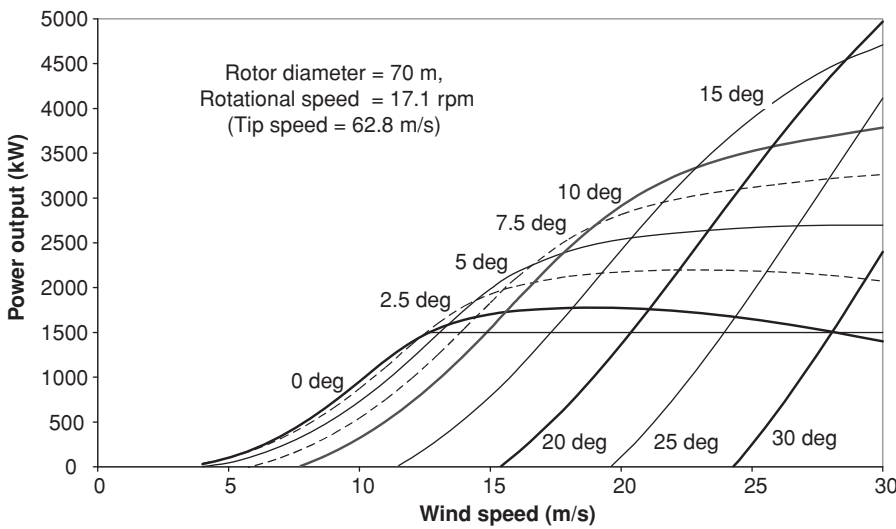


Figure 6.9 Power curves for different positive pitch angles: 70 m diameter three-bladed rotor rotating at 17.1 rpm

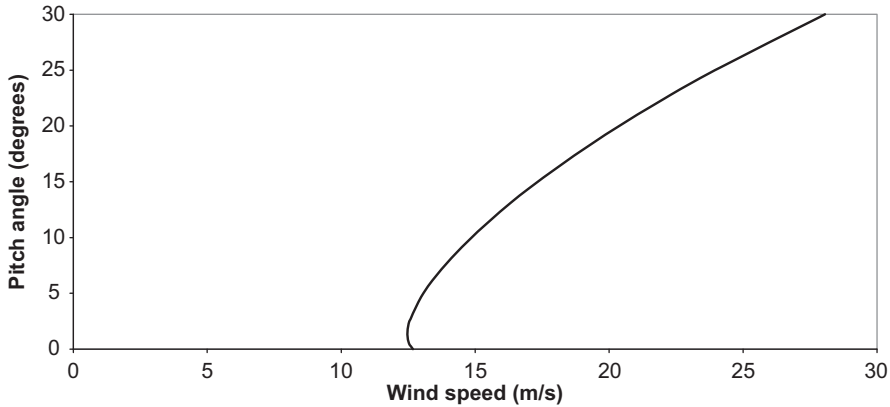


Figure 6.10 Schedule of pitch angles versus wind speed for limiting the power output of the machine featured in Figure 6.9 to 1.5 MW

and pitch angle required for power control (Figure 6.10). It is readily apparent from the power curve gradients at the intersection points that rapid changes of wind speed will result in large power swings when the mean wind speed is high.

The range of blade pitch angles required for power control is typically from 0° (often referred to as ‘fine pitch’), at which the tip chord is in the plane of rotation or very close to it, and about 35° . However, for effective aerodynamic braking, the blades have to be pitched to 90° or full feather, when the tip chord is parallel to the rotor shaft with the leading edge into the wind.

A variety of pitch actuation systems have been adopted (see also Section 8.5). They are divided between those in which each blade has its own actuator and those in which a single actuator pitches all the blades. The former arrangement has the advantage that it provides two or three independent aerodynamic braking systems to control overspeed, and the disadvantage that it requires very precise control of pitch on each blade in order to avoid unacceptable pitch angle differences during normal operation. An advantage of the latter arrangement is that the pitch actuator – for example, a hydraulic cylinder – can be located in the nacelle, producing fore-aft motion of the pitch linkages in the hub by means of a rod passing down the middle of a hollow low speed shaft (see Figure 6.11). Alternatively, the axial position of the rod can be controlled by means of a ball-screw and ball-nut arrangement, in which the ball-nut is driven by a servomotor. Normally the ball-nut is driven at the same speed as the rotor, but when a change of pitch is required the ball-nut rotational speed is altered temporarily. This system is arranged to be fail-safe, so that should the servomotor or its control system fail, the servomotor is braked automatically and the ball-nut drives the blade pitch to feather.

Where hydraulic cylinders are used to pitch blades individually, they are mounted within the hub and each piston rod is usually connected directly to an attachment on the blade bearing (see Figure 6.12). The attachment point follows a circular path as the blade pitches, so the cylinder has to be allowed to pivot. The alternative solution of employing an electric motor to drive a pinion engaging with teeth on the inside of the blade bearing consequently appears rather neater (see Figure 6.13). Both systems require a hollow shaft to accommodate either hydraulic hoses or power cables for pitch actuation together with signal cables for pitch angle sensing. In addition, appropriate slip rings are required at the rear end of the shaft.



Figure 6.11 Pitch linkage system used in conjunction with a single hydraulic actuator located in the nacelle. (The central triangular 'spider' is connected to the actuator by a rod passing through the hollow low-speed shaft. Links from the spider drive the blade pitch *via* braced arms cantilevering into the hub from each blade. Each arm is parallel to its blade axis, but eccentric to it.)

Methods of providing back-up power supplies to ensure blade feathering in the event of grid-loss are considered in Section 8.5.

Although full-span pitch control is the option favoured by the overwhelming majority of manufacturers, power control can still be fully effective even if only the outer 15% of the blade is pitched. The principal benefits are that the duty of the pitch actuators is significantly

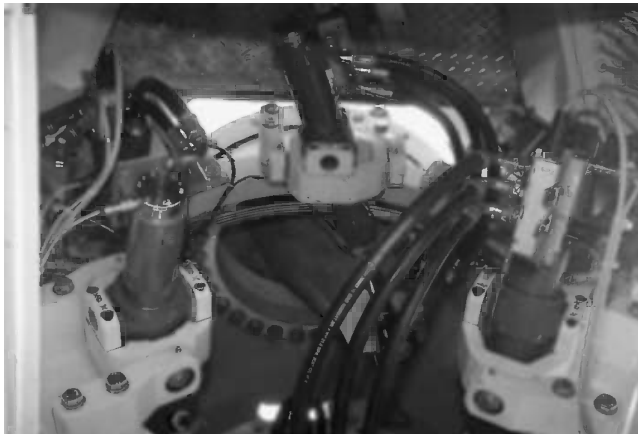


Figure 6.12 Blade pitching system using separate hydraulic actuators for each blade. (Each actuator cylinder is supported on a gimbal-type mounting bolted to the hub and its piston applies a pitching torque to the blade via a cantilevered conical tube eccentric to the blade axis. The blade is attached to the outer ring of the pitch bearing.)

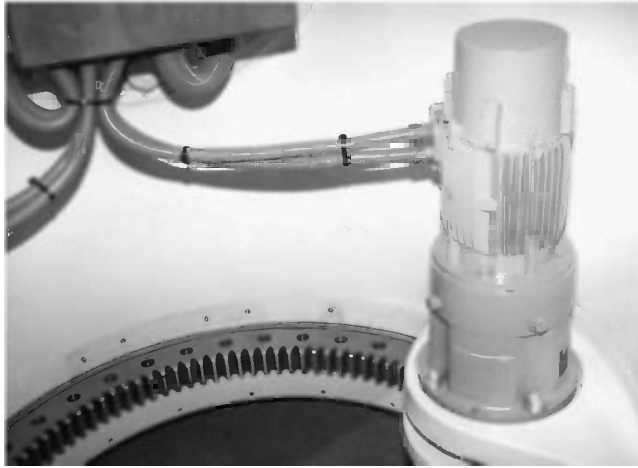


Figure 6.13 Blade pitching system using a separate electric motor for each blade. (A pinion, driven by the motor via a planetary gearbox, engages with gear teeth on the inside of the inner ring of the pitch bearing, to which the blade is bolted. The blade is not attached to the bearing in this photograph, so the fixing holes are visible.)

reduced, and that the inboard portion of the blade remains in stall, significantly reducing the blade load fluctuations. On the other hand partial-span pitch control has several disadvantages as follows:

- the introduction of extra weight near the tip;
- the difficulty of physically accommodating the actuator within the blade profile;
- the high bending moments to be carried by the tip-blade shaft;
- the need to design the equipment for the high centrifugal loadings found at large radii; and
- the difficulty of access for maintenance.

It should be apparent from the above brief survey of pitch actuation systems that the design of the hardware required for pitch-regulation is a significant task. Moreover, as regards the controller, pitch-regulation introduces the need for fast response closed loop control, which is not required for the supervisory functions on a stall-regulated machine. Thus, the benefits of pitch control have to be weighed carefully against all the additional costs involved, including the cost of maintenance.

Another factor that needs to be considered is fatigue loading. This increases significantly on full-span pitch-regulated machines, because the rate of change of lift coefficient with angle of attack remains at about 2π (see Equation A3.9 in Appendix A3) instead of reducing to zero as the blade goes into stall, with the result that rapid changes in wind speed above rated will cause bigger thrust load changes.

Pitch system controller design is considered in detail in Chapter 8.

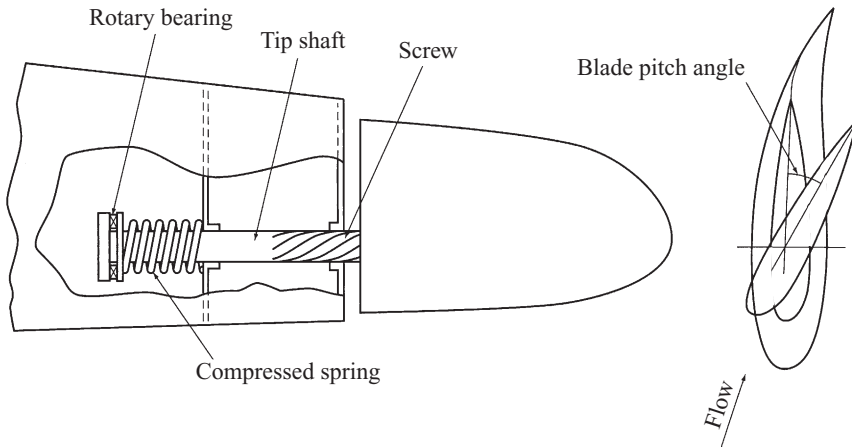


Figure 6.14 Passive control of tip blade, using screw on tip shaft and spring

6.7.3 Passive pitch control

An attractive alternative to active control of blade pitch to limit power is to design the blade and/or its hub mounting to twist under the action of loads on the blades in order to achieve the desired pitch changes at higher wind speeds. Unfortunately, although the principle is easy to state, it is difficult to achieve it in practice, because the required variation in blade twist with wind speed generally does not match the corresponding variation in blade load. In the case of stand-alone wind turbines, the optimisation of energy yield is not the key objective, so passive pitch control is sometimes adopted, but the concept has not been utilised as yet for many grid-connected machines.

Corbet and Morgan (1991) give a survey of how different types of blade loads might be utilised. Harnessing the centrifugal load is obviously promising in the case of variable speed machines, and this has been demonstrated using a screw cylinder and preloaded spring to passively control each tip blade, within the Dutch FLEXHAT programme. When the centrifugal load on the tip exceeds the preload, the tip blade is driven outwards against the spring and pitches (see Figure 6.14 for illustration of the concept).

Joose and Kraan (1997) have proposed replacing this mechanism by a maintenance-free ‘Tentortube’, which would twist under tension loading. This tube would be carbon-fibre reinforced with all the fibres set at an angle to the axis, so that centrifugal loading induced twist. It would be placed inside a hollow steel tip shaft, which would carry the aerodynamic loading on the tip blade.

6.7.4 Active stall control

Active stall control achieves power limitation above rated wind speed by pitching the blades initially *into* stall – that is, in the opposite direction to that employed for active pitch control, and is thus sometimes known as negative pitch control. At higher wind speeds, however, it is usually necessary to pitch the blades back towards feather in order to maintain power output at rated.

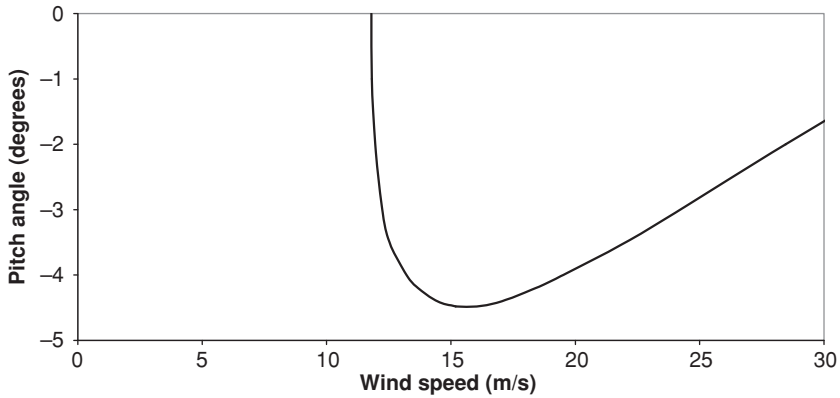


Figure 6.15 Schedule of pitch angles required to limit 70 m diameter turbine output to 1.5 MW at different wind speeds using active stall control

A significant advantage of active stall control is that the blade remains essentially stalled above the rated wind speed, so that gust slicing (see Section 6.7.2) results in much smaller cyclic fluctuations in blade loads and power output. It is found that only small changes of pitch angle are required to maintain the power output at rated, so pitch rates do not need to be as large as for positive pitch control. Moreover, full aerodynamic braking requires pitch angles of only about -20° , so the travel of the pitch mechanism is very much reduced compared with positive pitch control.

Figure 6.15 shows a schedule of pitch angle against wind speed for active stall control. The active stall control schedule is derived from the intersection of the family of power curves for different negative pitch angles for the 70 m diameter machine considered above with the 1500 kW ordinate in Figure 6.16. Note that the rotational speed has been increased by 10% so that the machine operates further away from stall below rated wind speed – otherwise the range of negative pitch angles utilised would be very small.

The principal disadvantage of active stall control is the difficulty in predicting aerodynamic behaviour accurately in stalled flow conditions. Active stall control is considered further in Section 8.2.1, Chapter 8.

6.7.5 Yaw control

As most horizontal axis wind turbines employ a yaw drive mechanism to keep the turbine headed into the wind, the use of the same mechanism to yaw the turbine out of wind to limit power output is obviously an attractive one. However, there are two factors which militate against the rapid response of such a system to limit power – firstly, the large moment of inertia of the nacelle and rotor about the yaw axis and, secondly, the cosine relationship between the component of wind speed perpendicular to the rotor disc and the yaw angle. The latter factor means that, at small initial yaw angles, yaw changes of, say, 10° only bring about reductions in power of a few percent, whereas blade pitch changes of this magnitude can easily halve the power output. Thus, active yaw control is only practicable for variable speed machines where the extra energy of a wind gust can be stored as rotor kinetic energy until the yaw drive has made the necessary yaw correction. This design philosophy has been exploited successfully

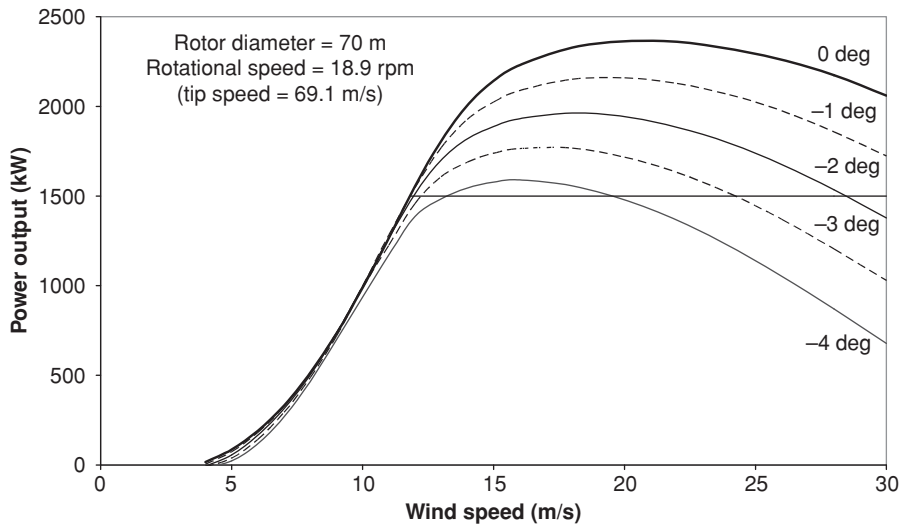


Figure 6.16 Power curves for different negative pitch angles: 70 m diameter three-bladed rotor rotating at 18.9 rpm

in Italy on the 60 m diameter Gamma 60 prototype, which has an impressive maximum yaw rate of 8° per second (Coiante *et al.*, 1989).

6.8 Braking systems

6.8.1 Independent braking systems: requirements of standards

The GL rules require that a wind turbine shall have two independent braking systems. On the other hand, IEC 61400–1 does not explicitly require the provision of two braking systems, but it does require the protection system to remain effective even after the failure of any non-safe-life protection system component.

IEC 61400–1 and the GL rules require that at least one of the braking systems should act on the rotor or low speed shaft.

Normal practice is to provide both aerodynamic and mechanical braking. However, if independent aerodynamic braking systems are provided on each blade, and each has the capacity to decelerate the rotor after the worst-case grid loss, then the mechanical brake will not normally be designed to do this as well. The function of the mechanical brake in this case is solely to bring the rotor to rest – that is, to park it – as aerodynamic braking is unable to do this.

6.8.2 Aerodynamic brake options

Active pitch control

Blade pitching to feather (i.e. to align the blade chord with the wind direction) provides a highly effective means of aerodynamic braking. Blade pitch rates of 10° per second are

generally found adequate, and this is of the same order as the pitch rate required for power control. The utilisation of the blade pitch system for start-up and power control means that it is regularly exercised with the result that the existence of a dormant fault is highly unlikely.

In machines relying solely on blade pitching for emergency braking, independent actuation of each blade is required, together with fail-safe operation should power or hydraulic supplies passing through a hollow low speed shaft from the nacelle be interrupted. In the case of hydraulic actuators, oil at pressure is commonly stored in accumulators in the hub for this purpose.

Pitching blade tips

Blade tips which pitch to feather have become the standard form of aerodynamic braking for stall-regulated turbines. Typically the tip blade is mounted on a tip shaft, as illustrated in Figure 6.14, and held in against centrifugal force during normal operation by a hydraulic cylinder. On release of the hydraulic pressure (which is triggered by the control system, or directly by an overspeed sensor), the tip blade flies outwards under the action of centrifugal force, pitching to feather simultaneously on the shaft screw. The length of the tip blade is commonly some 15% of the tip radius.

The ability of the control system to trigger blade tip activation is of crucial importance. On a number of early machine designs, the blade tips were centrifugally activated only, so there could be long periods without overspeed events when they did not operate. As a result there was a risk of seizure when operation was eventually required. With the now commonplace arrangement enabling the control system to activate the tip as well, the system can be routinely tested automatically. The penalty is that the low speed shaft needs to be hollow to accommodate the feed to the hydraulic cylinder.

Spoilers

Spoilers are hinged flaps, which conform to the aerofoil profile when retracted, and stick out at right angles to it when deployed. However, although such devices have been used in the past, they have to be of considerable length in order to decelerate the rotor adequately (Jamieson and Agius, 1990). Moreover, unless the design allows for their operation to be regularly tested, there is a risk that they will fail to deploy when actually needed.

Other devices

Various other devices have been suggested, such as:

- Ailerons.
- The sliding leading edge device or SLEDGE, in which a length of leading edge at the tip slides radially outwards.
- The flying leading edge device or FLEDGE, in which the whole leading edge together with an adjacent section of the camber face is pitched towards feather.

Jamieson and Agius (1990) and Armstrong and Hancock (1991) give useful surveys of these and other aerodynamic braking devices, and note that the SLEDGE device, which utilises only 2% or 3% of the blade area, is highly effective aerodynamically. Derrick (1992)

examines the capabilities of the SLEDGE and FLEDGE devices for both braking and power control in more detail. Despite their promise, these devices have not yet found commercial application.

6.8.3 Mechanical brake options

As noted in Section 6.8.1, the duty of the mechanical brake need only be that of a parking brake on machines where the aerodynamic brakes can be actuated independently. However, on pitch-regulated machines where blade position is controlled by a single actuator, full independent braking capability has to be provided by the mechanical brake. It is worth noting that several manufacturers of stall-regulated machines fitted with independent tip brakes ensure that the mechanical brake can stop the rotor unassisted. This may be to satisfy requirements in certain countries that two independent braking systems of a different type are provided.

A wind turbine brake typically consists of a steel brake disc acted on by one or more brake calipers. The disc can be mounted on either the rotor shaft (known as the low speed shaft) or on the shaft between the gearbox and the generator (known as the high speed shaft). The latter option is much the more common because the braking torque is reduced in inverse proportion to the shaft speeds, but it carries with it the significant disadvantage that the braking torques are experienced by the gear train. This can increase the gearbox torque rating required by as much as ca 50%, depending on the frequency of brake application – see Section 7.4.5, Chapter 7. Another consideration is that the material quality of brake discs mounted on the high-speed shaft is more critical, because of the magnitude of the centrifugal stresses developed.

The brake calipers are almost always arranged so that the brakes are spring applied and hydraulically retracted – that is, fail-safe.

Aerodynamic braking is much more benign than mechanical braking as far as loading of the blade structure and drive train is concerned, so it is always used in preference for normal shut-downs.

6.8.4 Parking versus idling

Although a mechanical parking brake is essential for bringing the rotor to rest for maintenance purposes, many manufacturers allow their machines to idle in low winds and some do so during high wind shutdowns. The idling strategy has two clear advantages – it reduces the frequency of imposition of braking loads on the gear train and gives the impression to members of the public that the turbine is operating even when it is not generating. On the other hand, gearbox and bearing lubrication must be maintained throughout.

6.9 Fixed speed, two speed or variable speed

Wind turbine rotors operate most effectively only at the one particular tip speed ratio that gives the maximum Power Coefficient (see Figure 3.15, Chapter 3). Hence, fixed rotational speed turbines operate sub-optimally except at the wind speed corresponding to this tip speed ratio. Energy capture can be increased by varying the rotational speed so that the turbine runs at optimum tip speed ratio over a range of wind speeds. A slightly reduced improvement can be obtained by running the turbine at one of two fixed speeds so that the tip speed ratio is closer to the optimum more often than with a single fixed speed.

Noise considerations are often of significance in the decision to opt for variable or two speed operation. As noted in Section 6.4.4, the aerodynamic noise generated by a wind turbine is approximately proportional to the fifth power of the tip speed. Both variable speed and two speed operation allow the rotational speed to be reduced in low winds, thus reducing turbine aerodynamic noise dramatically when it could otherwise be objectionable because of low background ambient noise.

Variable speed operation leads to a reduction in turbine mechanical loads as the control system gains can be reduced to allow small variations in the rotor speed in response to wind turbulence and cyclic torque variations (e.g. those caused by tower shadow). Also, larger wind farms are required by the electrical Transmission System Operators to comply with the Grid Code regulations that specify the electrical performance required. These regulations are designed to ensure that wind farms support the electrical power system by, for example, controlling their output voltage and continuing to operate when electrical network faults occur. These requirements are difficult to meet with fixed or two-speed induction generators.

6.9.1 Two speed operation

Two speed operation is implemented using two fixed speed induction generators. In low wind speeds, a smaller (lower power and speed) electrical generator is used to operate over a range of low wind speeds. When the wind speed rises and the power limit of this generator is reached the smaller generator is disconnected and a full power, higher speed, generator connected. Similarly when the wind speed drops the full power generator is disconnected and the smaller generator used. A hysteresis control system, measuring power, is used, to restrict the number of switching operations (Figure 6.17).

Either generators of differing numbers of poles (giving different speeds of rotation) are connected to gearbox output shafts rotating at the same speed, or generators with the same number of poles are connected to output shafts rotating at differing speeds. The power rating of the generator for low speed operation is normally around a third of the turbine rotor rating.

The development of induction generators with two sets of windings has allowed the number of poles within a single generator to be varied by connecting the windings together in different ways. Either two separate windings or a technique known as Pole Amplitude Modulation is used. Standard generators of this type are available which can be switched between four and six pole operation, giving a speed ratio of 1.5 (1800 and 1200 rpm for a 60 Hz system). Given correct selection of the gearbox ratio, this ratio produces close to the optimum increase of energy capture.

In the case of stall regulated wind turbines, only limited energy gains are to be had by converting from single fixed speed to two speed operation. The maximum rotational speed of the two speed machine is restricted to the rotational speed of the fixed speed machine in order to limit the aerodynamic power through aerodynamic stall. Energy gains are only of the order of 2 or 3%, but, nevertheless, two speed operation can be considered worthwhile on stall regulated machines because of noise considerations.

In the case of pitch regulated machines, the energy gain obtainable by moving to two speed operation depends on the power rating and rotational speed of the baseline fixed speed machine. Where these parameters have been chosen to be close to the optimum, in relation to rotor diameter and rotor chord, an energy gain of only about 3% is attainable. However, energy gains of up to 10% are possible when the baseline design is sub-optimal. Control of the rotor rotational speed during the transition from the low speed to the high speed generator

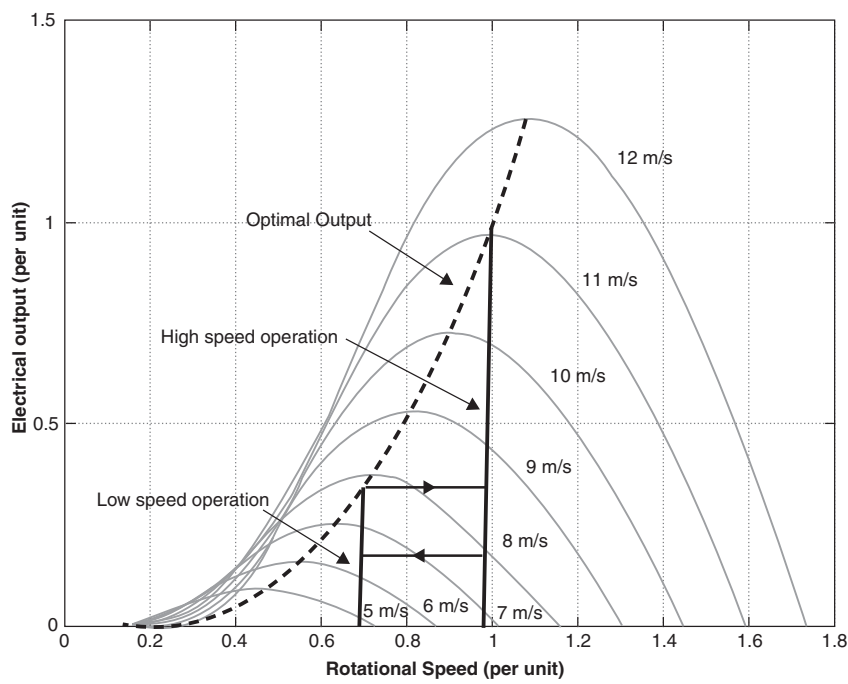


Figure 6.17 Locus of operation of two speed wind turbine

(and back) is much easier with rotor pitch control as the rotor speed can be controlled easily during the transitions.

Some disadvantages of two speed operation are as follows:

- Additional generator cost.
- Extra switchgear is required, which is subjected to a demanding duty in terms of frequency of operation.
- Control of turbine speed is required during each speed change.
- Energy is lost while the generator is disconnected during each speed change.

6.9.2 Variable slip operation (see also Chapter 8, Section 8.3.8)

Variable slip represents a compromise between fixed and variable speed operation (Pedersen, 1995). The variable slip generator is an induction generator with a variable resistor in series with the rotor circuit, controlled by a high frequency semiconductor switch. Below rated torque or power the external resistor is short-circuited and the generator acts as a conventional fixed speed induction machine. Above rated torque, however, varying the resistance allows the generator torque to be controlled and the generator speed to increase, so the behaviour is then similar to a variable speed system. A speed increase of up to about 10% is typical.

This arrangement is cheaper than a variable speed system, and gives some of the advantages, in particular the control of torque in the drive train and the smoothing of aerodynamic torque variations above rated power. It does not offer increased aerodynamic efficiency below

rated (although it does not suffer from frequency converter losses), and it does not allow any control of the power factor. Electrical flicker will, however, be reduced above rated power as the power output is smoothed.

The entire power circuit of the generator rotor can be mounted on the shaft including the variable resistors and electronic control switch (Heier, 2006). This avoids the use of slip rings carrying the rotor current. The signal controlling the power electronic switch is transmitted to the rotating shaft using a non-contact optical device.

An advantage of mounting the resistors externally connected via slip rings is that it is then easier to dissipate the extra heat which is generated above rated, and which may otherwise be a limiting factor with large generators.

6.9.3 Variable speed operation

By interposing a frequency conversion between the generator (which may be synchronous or induction type) and the network, it is possible to decouple the rotational speed of the generator and turbine from the network frequency. As well as allowing the rotor speed to vary and so maximise energy capture, the generator air-gap torque may be controlled and mechanical loads reduced. An alternative approach to varying the speed of a synchronous or induction electrical machine is to use a wound rotor induction generator, as with variable slip, and replace the external resistors in the rotor circuit with a frequency converter. This is known as the Doubly Fed Induction Generator (DFIG) and has become widely used.

Variable speed operation has a number of advantages:

- Below rated wind speed, the rotor torque and, hence, speed can be made to vary to maintain peak aerodynamic efficiency (C_p).
- The reduced rotor speed in low winds results in a significant reduction in aerodynamically generated acoustic noise. Noise is especially important in low winds, where ambient wind noise is less effective at masking the turbine noise.
- The rotor can act as a flywheel, smoothing out aerodynamic torque fluctuations before they enter the drive train. This is particularly important at the blade passing frequency.
- Direct control of the air-gap torque allows gearbox torque variations above the mean rated level to be kept small.
- Both active and reactive power can be controlled, so that either any particular power factor can be maintained or the terminal voltage controlled. In principle, it is possible to use a variable speed wind farm as a source of reactive power to compensate for the poor power factor of other consumers on the network. For large wind farms, it is much easier to meet the Grid Code requirements of the Transmission System Operators with variable speed wind turbines than with fixed speed turbines.
- Variable speed turbines will also give higher power (voltage) quality due to the smoother output power they develop. The use of power electronic converters to connect the generators gradually to the network minimises electrical transients on connection.

In practice, losses in the frequency converters may be several per cent of their rated power, counteracting the increased aerodynamic efficiency below rated wind speed. The load reduction possibilities, however, mean that most large MW-scale turbines now are variable

speed. Variations in aerodynamic torque at blade passing frequency are particularly significant in larger turbines, because of the size of the rotor compared to the lateral and vertical length scales of turbulence, and these are not translated into output power variations in variable speed turbines. The turbine control system is tuned to allow the aerodynamic power variations to be absorbed as slight changes in rotational speed, that is, as variations in kinetic energy.

There is a significant cost associated with the variable speed equipment, which must be weighed against the advantages. Other drawbacks include increased complexity and the generation of electrical noise and harmonics by the inverter system. Modern Pulse Width Modulated (PWM) inverters operating at high switching frequency using Insulated Gate Bipolar Transistors (IGBTs) produce much lower levels of lower order undesirable harmonics (e.g. fifth, seventh, eleventh, thirteenth etc.) than earlier, naturally commutated designs that used Thyristors. Electrical noise can be a problem for control signals within the turbine if insufficient care is taken to shield cables. Fibre optic transmission is increasingly being used, and this is not affected by induced currents from adjacent cables.

There are two principal methods of achieving variable speed operation. In ‘broad range’ or ‘full power conversion’ (FPC) variable speed, the generator stator is connected to the network via two fully rated frequency converters. In ‘narrow range’ variable speed using a Doubly Fed Induction Generator (DFIG) both the generator stator and rotor are connected to the network, the stator directly, and the rotor through slip rings and smaller frequency converters.

Broad range variable speed allows the speed to vary from close to zero to the full rated speed, but all the power of the wind turbine has to pass through the two frequency converters. Narrow range variable speed uses smaller, cheaper frequency converters, since only a fraction of the power passes through them, but the speed can only vary by a more limited amount, typically 20–40%, either side of synchronous speed.¹ In practice, this is enough to achieve almost all the advantages of variable speed operation. A disadvantage is the need to use a wound rotor induction machine with a small air-gap and the maintenance requirements of the slip rings.

In both broad and narrow speed range wind turbines, the frequency conversion is made by two back-back voltage source converters connected through a direct current (DC) link. These rectify the power into the DC link and then invert it to network frequency or to feed the generator. This rectification into direct current and inversion isolates the network frequency from either the generator speed, in the case of broad range, or the rotor frequency in the case of a DFIG wind turbine. In both cases the control system works by measuring the generator rotational speed and applying the torque required to keep the aerodynamic rotor at optimum tip speed ratio over a wide a range of wind speeds.

Figure 6.18 shows the control objective of a variable speed wind turbine. The rotor speed is measured and, once cut-in speed is reached, a torque is applied by the variable speed generator to stay close to the optimum curve (O-A). At maximum rotational speed (point A), this is maintained by the converter until the blades are pitched to control incoming power.

Figure 6.19 (a–e) shows the commonly used wind turbine generator systems.

- Figure 6.19 (a) shows a fixed speed (squirrel cage) induction generator with shunt capacitors to improve the output power factor. These capacitors may be switched to provide more reactive power as the wind power increases.

¹ Synchronous speed is fixed by the frequency of the network (50 or 60 Hz) and the number of magnetic poles of the generator construction

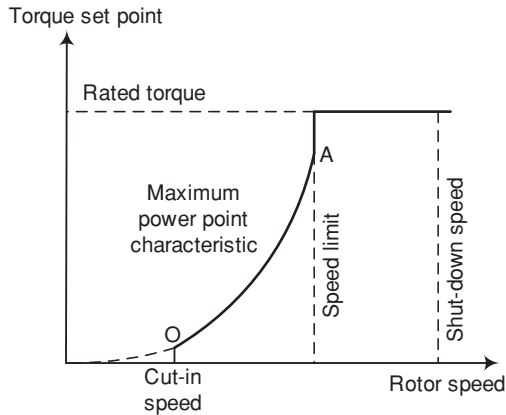


Figure 6.18 Control objective of a variable speed wind turbine. (See also Figure 8.3, Chapter 8)

- Similar power factor correction capacitors are connected to the stator of the wound rotor induction machine in Figure 6.19 (b). However the rotor circuit has additional rotor resistance controlled by a Pulse Width Modulated (PWM) switch to give variable slip operation.
- In Figure 6.19 (c) the controllable rotor resistance is replaced by back-back voltage source converters (using Insulated Gate Bipolar Transistors – IGBT) to connect the rotor to the network. This gives variable speed operation over a restricted speed range depending on the direction of power flow in the rotor circuit. Although there are switching losses in the converters, this arrangement avoids energy being dissipated in an external rotor resistance. Control of the generator speed and power factor is by the rotor side converter while the network side converter maintains the DC link voltage. The crow-bar circuit is to protect the generator side converter when faults occur on the AC network and to assist in ensuring continuous operation of the wind turbine during these faults.
- All the power from the generator (that may be synchronous or induction type) is rectified to DC in the arrangement of Figure 6.19 (d). Again control of the generator torque (and hence speed) and excitation is by the machine side converter while the network side converter maintains the DC link voltage. The network side converter can be used to control the network voltage or reactive power flow into the network.
- An alternative arrangement is shown in Figure 6.19 (e) with the network side converter controlling the output power of the generator (and hence the wind turbine speed) while the machine side is a simple diode rectifier followed by a DC:DC voltage converter. The generator must then be of the synchronous type, either wound rotor or permanent magnet, but the diode rectifier has lower losses than a controllable IGBT converter.

6.9.4 Other approaches to variable speed operation

There are other possible approaches to variable speed operation, although none of these has found wide-spread commercial application. They include:

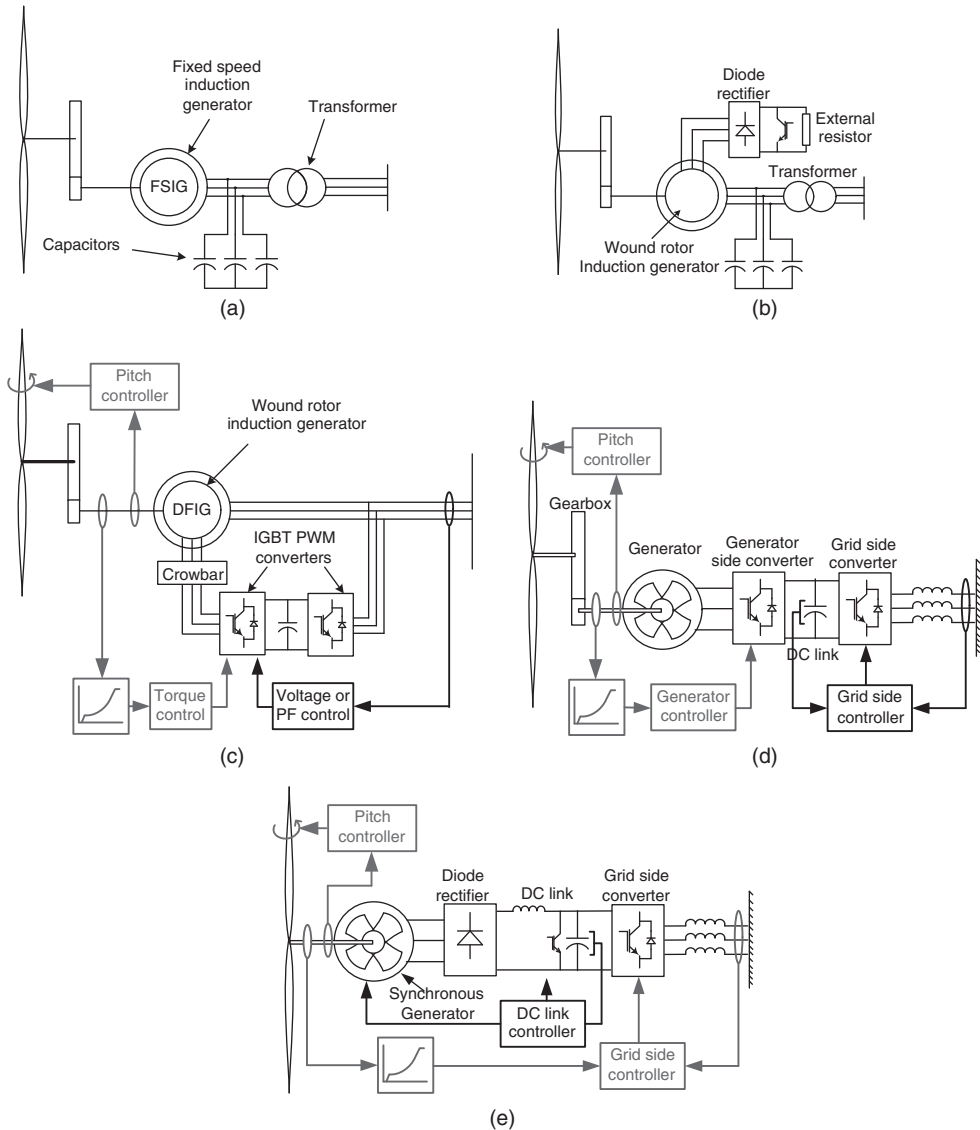


Figure 6.19 Wind turbine architectures. (a) Fixed speed induction generator. (b) Variable slip generator. (c) Doubly fed induction generator. (d) Full power converter wind turbine. (e) Full power converter wind turbine (diode rectifier). (Anaya Lara, 2009)

- Use of a differential gearbox, with the third shaft controlled by a variable speed electric motor/generator (Law *et al.*, 1984; Burton *et al.*, 1990) or by a hydraulic pump/motor (Henderson *et al.*, 1990).
- Mechanical continuously variable transmission systems such as have been developed for automotive applications.

6.10 Type of generator

Fixed speed wind turbines differ from almost all conventional generating plant by using induction rather than synchronous generators. This choice is driven by the need for significant torsional compliance and extraction of damping energy in the drive train due to the cyclic variations in the torque developed by the aerodynamic rotor.

Both synchronous and induction generators have similar winding arrangements on the stator which, when connected to the three-phase network voltage, produce a fixed speed, rotating magnetic field. However, the rotors of the two machines are quite different (Hindmarsh, 1984; McPherson, 1990). A synchronous machine has magnets mounted on its rotor and the rotor magnetic field then locks into that produced by the stator leading to operation at synchronous speed. For large scale power generation applications, electro-magnets are used on the rotor excited by an externally applied direct current. Although the rotor operates at the same speed as the stator magnetic field it leads the stator field by an angle depending on the applied torque. In contrast the rotor of a conventional induction machine has a 'squirrel cage' winding into which currents are induced as the rotor bars cut the magnetic field produced by the stator. Hence, an induction generator can only develop torque at a rotational speed slightly greater than that of the stator field. This 'slip speed' is proportional to the applied torque.

Therefore, to a first approximation, the behaviour of a synchronous machine may be considered to be analogous to a torsional spring. The torque is proportional to the angle between the rotor and the stator field. This angle is known as the load or power angle. In contrast an induction generator can be thought of as a torsional damper where the torque is proportional to the difference in speed between the rotor and the stator field (the slip speed). This is illustrated in simple schematic form in Figure 6.20. It may be seen that if the simple model of a fixed speed wind turbine, equipped with a synchronous generator, is excited by the cyclic torque from the wind turbine rotor then there is no damping energy extracted by the drive train to control the torsional oscillations. It is a simple two spring, two mass system. In contrast, with an induction generator, the connection of the generator to the network is represented by a torsional damper. The main cyclic torque of the wind turbine rotor will be at blade passing frequency and it is an unfortunate coincidence that this often matches quite closely the natural frequency of oscillation of a small synchronous generator connected to an electrical distribution network.

In practice, synchronous generators are often fitted with cage damper windings but it is not practical to provide the degree of damping required for wind turbine applications. Also at higher ratings (above, say, 1 MW) second order effects tend to reduce the damping energy that can be extracted by an induction generators (Saad-Saoud and Jenkins, 1999). However, the basic principle remains that the damping energy extracted from the drive train by induction generators is necessary for operation of fixed speed wind turbines.

In contrast, the generator of a variable speed wind turbine is not connected directly to the network but is de-coupled through the DC link of the frequency converters. Hence either synchronous or induction generators may be used with full power conversion variable speed systems.

6.10.1 Historical attempts to use synchronous generators

All large hydro, fossil fired or nuclear power stations use synchronous generators. Induction generators are much less useful than synchronous generators for large-scale power generation.

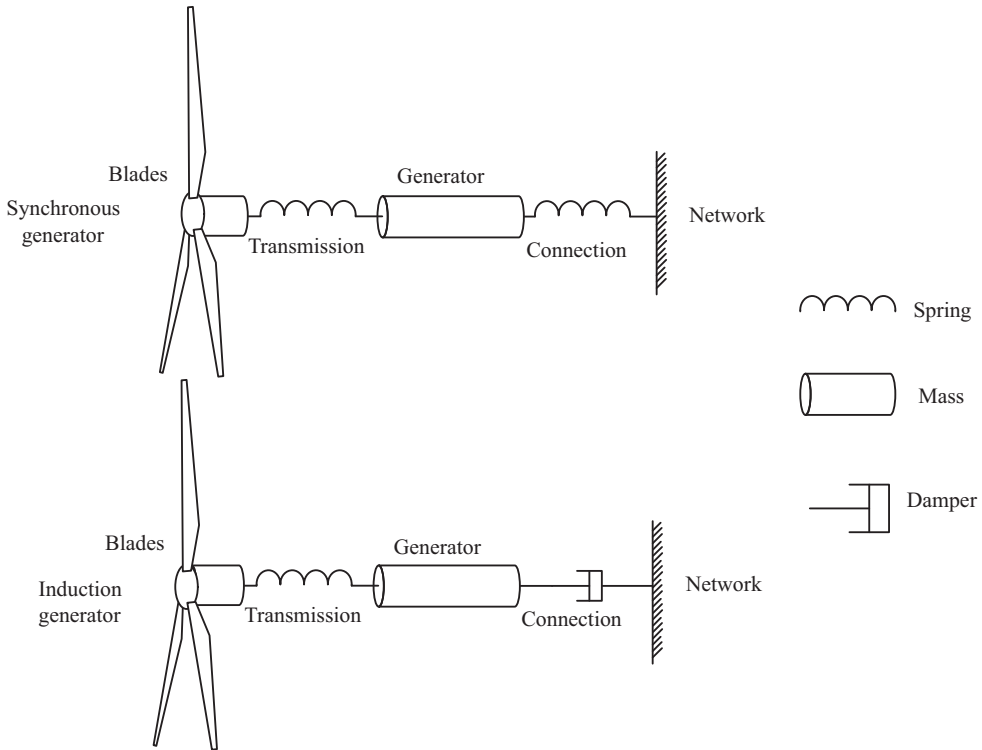


Figure 6.20 Mechanical analogues of directly connected generators

- The damping action results in higher energy losses in the rotor than with synchronous generators. It is then, of course, necessary to arrange for the removal of the heat dissipated in the rotor.
- All the reactive power necessary to energise the magnetic circuits must be supplied from the network (or by local capacitors). If local capacitors are used then there is the danger of self-excitation.
- There is no direct control over the terminal voltage or reactive power flow.
- Induction generators do not produce sustained fault current for three-phase faults on the network.
- They suffer from problems of voltage instability. This was not an important issue with limited wind generation but with large wind farms on weak networks can be of concern.

Hence, in the early development of wind turbines considerable efforts were made to use synchronous generators. These involved a number of innovative solutions to the provision of damping. For example, both Westinghouse in the USA and Howden in the UK used fluid couplings in the drive train to provide damping. The Wind Energy Group in the UK mounted a 250 kW synchronous generator flexibly using a spring-damper system and connected a 3 MW synchronous generator through a sophisticated variable speed mechanical gearbox

(Law *et al.*, 1984). These and other similar approaches using synchronous generators on large wind turbines are now mainly of historical interest but such concepts continue to be pursued by some designers.

6.10.2 Direct drive generators

There has been considerable development of generators driven directly by the wind turbine rotor without a speed increasing gearbox and a number of manufacturers offer such wind turbines. The obvious advantage is the elimination of the gearbox. However, the power output of any rotating electrical machine may be described generally by (Laithwaite and Freris, 1980):

$$P = K D^2 L n$$

where

- D is the rotor diameter
- L is the length
- n is the rotational speed
- K is a constant

Thus, it may be seen that if the rotational speed is reduced then it is necessary either to lengthen the generator in proportion or to increase the diameter. It is cheaper to increase the diameter as this raises the power by the square rather than linearly. Thus, direct drive generators for wind turbines tend to have rather large diameters, and hence weights, but with limited length.

Induction generators require a small radial distance between the surface of the rotor and the stator (known as the air-gap). This is necessary to ensure an adequate air-gap magnetic flux density as all the excitation is provided from the stator. In contrast, synchronous generators have excitation systems on the rotor and so can operate with larger air-gaps. It is difficult to manufacture large diameter electrical machines with small air gaps for mechanical and thermal reasons. Hence, direct drive wind turbines use synchronous generators (either with permanent magnet excitation or with a wound rotor and electro-magnets providing the field). The use of a synchronous generator, in turn, leads to the requirement for solid-state frequency conversion equipment to de-couple the generator from the network and full power conversion variable speed operation. If permanent magnets are used, then there is no control over the excitation of the synchronous machine and its output voltage will vary with rotational speed. Then it may be necessary to use an additional power electronic converter to control the DC link voltage. The manufacture of very large, permanent magnet generators can be difficult as the forces developed by the magnets can be very great.

Direct drive, radial flux synchronous generators (Figure 6.27) with electrical excitation are used at up to the highest ratings. Generators with permanent magnet excitation may be either radial or axial flux designs but in order to limit the cost of the rare earth permanent magnets the air gap between the stator and rotor is kept as small as possible and so very high forces can develop. Double air-gap designs are used to balance the forces of the permanent magnets (Heier, 2006).

An alternative to complete elimination of the gearbox is a hybrid approach with a gearbox of one or two stages and a medium-speed generator connected through a full power converter. This allows a simpler and more reliable gearbox to be used with the generator of a similar

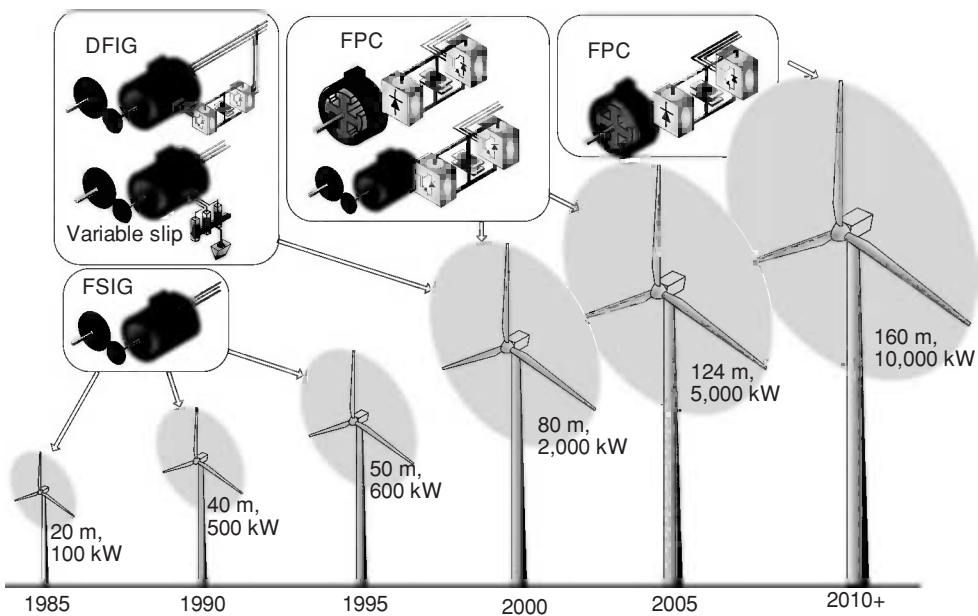


Figure 6.21 Evolution of wind turbine generator systems

size (EWEA, 2009). A further alternative design approach is to use multiple high speed shafts from the gearbox and multiple generators. This again leads to a compact and balanced drive train but multiple power converters are required in parallel. The medium-speed generators are generally permanent magnet synchronous machines.

6.10.3 Evolution of generator systems

Figure 6.21 shows the evolution of wind turbine generator systems. Early wind turbines used Fixed Speed Induction Generators (FSIG) either single or two speed. These operated at effectively constant speed (with slip less than 3%) using either stall regulation of the aerodynamic rotor or pitch regulation of the blades. The arrangement was applied to wind turbines as large as 1.5 MW but above this rating there are increasing difficulties in controlling drive train oscillations and limiting mechanical loads. In addition FSIG wind turbines have difficulties meeting the requirements of the Grid Codes that are applied to large wind farms. Variable slip wind turbines using rotor resistors were then developed but these have significant losses when the rotor resistance is used.

From around 2000, Doubly-Fed Induction Generators (DFIG) became increasingly common in large wind turbines where the benefits of limited variable-speed operation were required but at the reduced cost of controlling only a fraction of the output power. However, slip rings are required on the generator rotor and the air-gap of a wound rotor induction generator must be kept small. A possible future development is the use of the brushless doubly-fed generator where rather than use slip rings the rotor is excited by a second controlled stator winding. This approach eliminates the requirement for slip-rings and brushes but is not yet widely used commercially.

At the same time as the DFIG architecture was being implemented widely, Full Power Conversion (FPC) equipment was being used with both generators driven through a gearbox and those driven directly by the aerodynamic rotor. Both wound rotor and permanent magnet low speed, large diameter synchronous generators have been used for direct drive applications. At present, there is little consensus on the architecture of future electrical generator systems for very large wind turbines.

6.11 Drive train mounting arrangement options

6.11.1 Low speed shaft mounting

The functions of the low speed shaft are the transmission of drive torque from the rotor hub to the gearbox, and the transfer of all other rotor loadings to the nacelle structure. Traditionally the mounting of the low speed shaft on fore and aft bearings has allowed these two functions to be catered for separately: the gearbox is hung on the rear end of the shaft projecting beyond the rear bearing and the drive torque is resisted by a torque arm. The front bearing is positioned as close as possible to the shaft/hub flange connection, in order to minimise the gravity moment due to the cantilevered rotor mass, which usually governs shaft fatigue design. The spacing between the two bearings will normally be greater than that between front bearing and rotor hub in order to moderate the bearing loads due to shaft moment. See Figure 6.22 for an illustration of a typical arrangement.

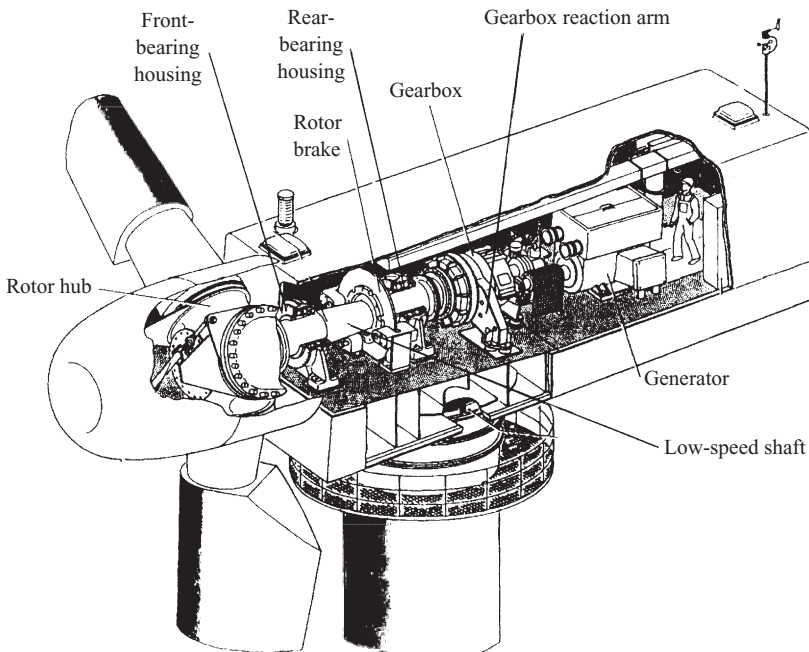


Figure 6.22 View of nacelle showing traditional drive shaft arrangement

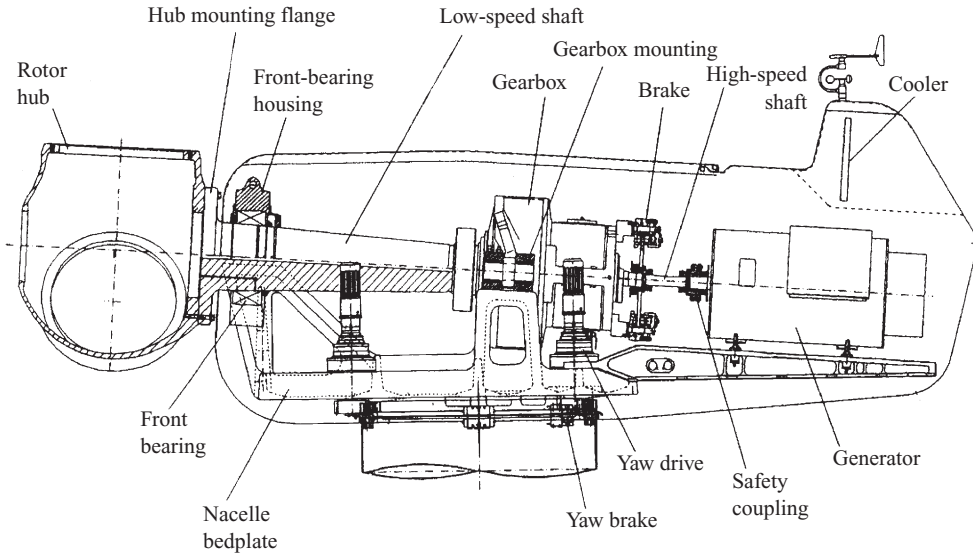


Figure 6.23 Nacelle arrangement for the Nordex N60 turbine. Reproduced by permission of Nordex

The opposite approach is to make the gearbox an integral part of the load path between the low speed shaft and tower top, that is, an ‘integrated gearbox’. The fore and aft low speed shaft bearings are absorbed within the gearbox, which moves to the front of the nacelle in order to minimise the rotor cantilever distance, and the gearbox casing then transmits the loads to the nacelle bedplate (Figure 6.28). Clearly this approach requires a much more robust gearbox casing, which must not merely resist the rotor loads, but do so without deflecting sufficiently to impair its functioning. Moreover its fore-aft length has to be increased in order to moderate the bearing loads due to shaft moment. The benefits lie in the reduced extent of the bedplate and the elimination of separate bearings requiring separate provision for lubrication, but a significant disadvantage is that gearbox replacement requires the removal of the rotor.

A configuration which is becoming increasingly popular is one intermediate between the two extremes described above, in which only the rear low speed shaft bearing is absorbed into the gearbox. The gearbox is usually set well back from the front bearing in order to reduce the rear bearing loads, and is rigidly fixed to supporting pedestals positioned on either side of the nacelle. Typical arrangements are shown in Figure 6.23, which shows a cross-section through the nacelle of the Nordex N-60 turbine, and in Figure 6.24. Note that the shaft tapers down in diameter towards the rear reflecting the reducing bending moment. The advantage of this arrangement is that the gearbox casing is not called upon to carry any moments due to cantilevered rotor mass or rotor out-of-plane loadings.

Figures 6.25 and 6.26 are aerial views of the nacelle of a NEG-Micon 1.5 MW machine with a similar drive train arrangement, after installation of the low speed shaft.

In the case of wind turbines with direct drive generators, the low speed shaft arrangement is dramatically different. The low speed shaft, which now connects the rotor hub to the rotor of the generator, is hollow, so that it can be mounted on a concentric fixed shaft cantilevered out from the nacelle bedplate. See Figure 6.27.

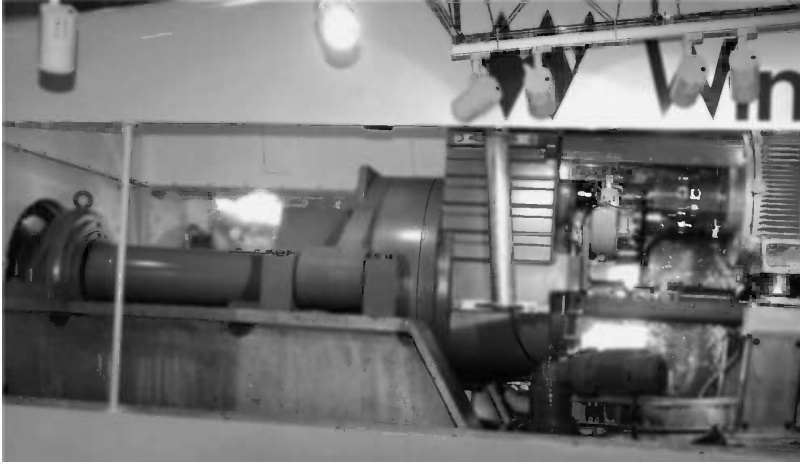


Figure 6.24 Drive train side view. From left to right the components visible through the cut-out in the nacelle wall are: (1) low-speed shaft front bearing, (2) low speed shaft, (3) gearbox mountings, (4) gearbox, (5) high-speed shaft with brake, (6) generator. The fabricated bed-plate is also visible.



Figure 6.25 Turbine assembly in the air (1); View of nacelle of 1.5 MW NEG Micon turbine after installation of low-speed shaft (front) and gearbox. The ring of bolt holes in the low-speed shaft for hub mounting are clearly visible. Reproduced by permission of NEG Micon



Figure 6.26 Turbine assembly in the air (2); View of low-speed shaft and front bearing after installation on 1.5 MW NEG Micon turbine. Reproduced by permission of NEG Micon

6.11.2 High speed shaft and generator mounting

The generator is normally mounted to the rear of the gearbox on an extension of the nacelle bedplate and the connecting drive shaft – the ‘high speed shaft’ – is fitted with flexible couplings at each end, to cater for small misalignments between the generator and gearbox.

The generator axis is normally offset from the low speed shaft axis. This is because, except in the case of machines fitted with a mechanical brake acting on the rotor, access is required to the rear end of the low speed shaft for hydraulic pipes, electrical cables or an actuator rod passing through the shaft, which is made hollow for the purpose, for blade pitch control or the activation of tip brakes. Usually the generator is either offset to one side of the nacelle, which introduces asymmetry into the nacelle bedplate, or it is offset vertically upwards, which requires a vertical step in the bedplate.

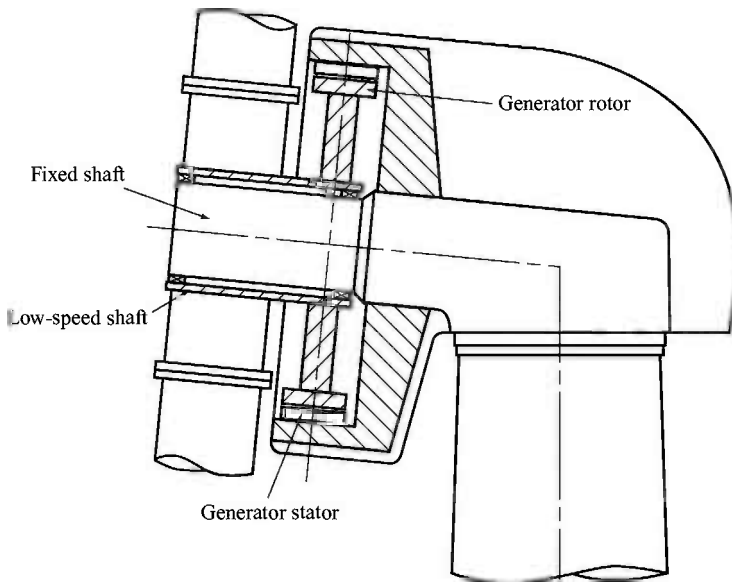


Figure 6.27 Radial flux direct-drive generator arrangement

A much more compact arrangement can be obtained by bolting the generator rigidly onto the rear of the gearbox via an adaptor tube (see Figure 6.28). The surfaces of the mating interfaces have to be carefully machined to ensure shaft alignment, and suitable access has to be provided to the coupling between the generator and gearbox output shafts. Despite the neatness of this layout, it has only been adopted by one or two manufacturers.

One consequence of locating the generator in the nacelle is that power cables running down the tower are required to twist as the nacelle yaws. On some large machines, the problems associated with the twisting of heavy cables have been avoided by mounting the generator vertically in the top of the tower, and driving the high speed shaft via a bevel gear. An alternative solution to the problem of heavy twisting cables, however, is to leave the generator in the nacelle and to transform to a higher voltage there as well.

6.12 Drive train compliance

The rotational dynamics of the drive train can have a major effect on loading. The effect is very different in fixed and variable speed turbines, but in each case the consequence of ignoring drive train dynamics at the design stage can be very severe.

In the variable speed case, the dynamics may be quite simple: the drive train may be modelled as a rotor and a generator inertia, separated by a torsional spring. Typically the natural frequency of this resonant system is quite high, of the order of 3–4 Hz. However, this mode is subject to very little damping, especially above rated where the generator torque is held constant. (Below rated the torque will be varied as a function of rotational speed, thus providing a small amount of damping.) There is very little aerodynamic damping from the rotor; and this mode of vibration can potentially generate very large gearbox torque



Figure 6.28 Integrated gearbox on Zond Z-750 turbine. (The gearbox is mounted on a circular nacelle bedplate, with the hub to the left and generator at the rear. An electrically driven yaw drive can be seen beneath the generator)

oscillations. Chapter 8 explains how the control system can be used to damp this mode by appropriate control of the generator torque, but it is important to ensure that the resonant frequency does not coincide with a significant forcing frequency such as $6P$, which can make it very difficult to achieve sufficient damping through the control system.

In the fixed speed case, the directly-coupled induction generator provides a lot of damping since the air-gap torque increases steeply with generator speed. The smaller the slip, the greater this damping. This might be expected to be beneficial, but in practice the reverse is likely to be true. Consider the drive train as a two degree of freedom system, with four elements in series: the rotor inertia, the shaft torsional spring, the generator inertia and, finally, the damper representing the slip curve, connected to ground (the constant-frequency grid). If the damping is very large, the generator can almost be considered to be locked (i.e. rotating at nearly constant speed), and the dynamics are dominated by the degree of freedom represented by the rotor inertia and the torsional spring, which has very little damping as in the variable speed case. With higher generator slip, the lower generator damping allows more movement of the generator inertia, causing more coupling between the two degrees of freedom. This gives a system in which there are two coupled torsional modes, each involving some movement of both rotor and generator inertias, and involving both the spring and the damper. Effectively the generator damping now affects both modes significantly: instead of a very lightly damped spring mode and a very heavily damped damper mode, we now have

two modes with intermediate damping, so the peak dynamic magnification is much lower. Thus, 0.5% rated generator slip can give a peak dynamic magnification of perhaps 2–5 at the resonant frequency, whereas with 2% slip the peak magnification may be no more than 1–1.5. The position of the peak with respect to blade passing frequency is critical – if the blade passing frequency is close to the peak, very large gearbox torque and electrical power oscillations will occur at this frequency, and it is very difficult to reduce these significantly using pitch control. Ideally the blade passing frequency should be well above the resonant frequency so that the dynamic amplification will be less than one, but it is not uncommon for power and torque oscillations at the blade passing frequency to be as large as ± 50 –100% of rated in high winds.

The use of a high slip generator greatly improves the situation, but there are two main drawbacks: Firstly each 1% of slip corresponds to 1% of extra losses, which significantly reduces the energy yield below rated wind speed. Secondly, these extra losses equate with heat dissipation in the generator, making it more difficult to keep the generator cool, especially in large machines.

An alternative to high generator slip which has occasionally been used is a fluid coupling between the gearbox and the generator. This is also a device which generates a torque proportional to slip speed, and it suffers from the same drawbacks as a high slip generator.

Another technique which has sometimes been used is to reduce the resonant frequency by introducing additional torsional flexibility into the drive train. This can be done by means of a quill shaft, a flexible low-speed coupling, or flexible mounts for the gearbox or even for the whole bedplate. The frequency reduction is, however, accompanied by a further loss of damping, and it may therefore be necessary to incorporate additional mechanical damping with the torsional flexibility, which is not always easy to engineer. Torsional flexibility in the high speed shaft is not usually practical because of the large angular movement required to achieve the necessary flexibility – half a revolution may be necessary, compared to just one or two degrees at the low speed shaft. An interesting variant (Leithead and Rogers, 1995) is to mount the generator on flexible mounts. This system can be tuned to absorb energy at the blade passing frequency through an additional mode of vibration of the generator casing against its mountings. This mode also affects the generator slip speed (the difference between rotor and casing speeds) and is, therefore, damped by the slip curve. Nevertheless, generator casing displacements would still need to be of the order of 10–15°, which is still not easy to engineer.

6.13 Rotor position with respect to tower

6.13.1 Upwind configuration

The upwind configuration is the one most commonly chosen. The principal advantage is that the tower shadow effect is much less for the same blade-tower spacing, reducing both dynamic loads on the blade and rhythmic noise effects. Set against this is the need to take great care to avoid the risk of blade-tower strikes with upwind machines, requiring accurate prediction of blade deflections under turbulent wind loading.

The clearance between the undeflected blade and the tower can be increased by tilting the low speed shaft upwards or by increasing the rotor overhang. It is desirable to keep the rotor overhang small in order to minimise low speed shaft and nacelle bedplate bending moments, so

the low speed shaft is normally tilted upwards by 5° or 6° to provide the necessary blade-tower clearance, at the cost of a very small reduction in power output.

6.13.2 Downwind configuration

The wind velocity deficit behind a wind turbine tower is much greater than that in front of it, to the extent that Powles (1983) has reported a turbulent region with essentially no forward velocity extending up to four tower diameters downstream of an octagonal tower. Beyond this distance, recovery is relatively rapid, with the deficit reduced to about 25% at seven tower diameters downstream.

In addition to the mean wind speed velocity deficit behind the tower, vortex shedding results in additional wind speed fluctuations over and above those already present due to turbulence. The two effects combine to present a harsh environment to the blades immediately behind the tower. The blades are subjected to a large negative impulsive load each time they pass the tower, which contributes significantly to blade fatigue damage, and the audible tower ‘thump’ that results is liable to be unwelcome. Designers usually mitigate both effects by positioning the rotor plane well clear of the tower, but this inevitably increases nacelle costs somewhat.

An important benefit of the downwind configuration is that it allows the use of very flexible blades without the risk of tower strike. Such blades benefit by being less severely unloaded by the tower shadow, because wind loading deflects them further from the tower in the first place.

6.14 Tower stiffness

A key consideration in wind turbine design is the avoidance of resonant tower oscillations excited by rotor thrust fluctuations at rotational or blade passing frequency. The damping ratio may be only 2–3% for tower fore-aft oscillations and an order of magnitude less for side-to-side motion, so unacceptably large stresses and deflections could develop if the blade passing frequency or rotational frequency coincided with tower natural frequency. This section begins by looking at the relative magnitudes of some of the excitations.

6.14.1 Stochastic thrust loading at blade passing frequency

Whereas the deterministic variations in blade loading due to wind shear, yaw etc. largely cancel out when the loadings on three blades are added together, the stochastic loadings due to turbulence do not, resulting in a significant rotor thrust load component at blade passing frequency. The magnitude of this quantity can be estimated fairly easily by assuming a linear relationship between fluctuations in the incident wind speed and the resultant load fluctuations according to Equation 5.25.

For the example three-bladed machine considered in Section 5.12.4, the total variance of rotor thrust is only about 20% less than it would be if the wind speed variations across the rotor were fully correlated. Thus, from Equation 5.129:

$$\sigma_T = \left(\frac{1}{2} \rho \Omega \frac{dC_l}{d\alpha} \right) \sigma_u \oint c r dr \quad (0.8) \quad (6.14)$$

where the integral sign $\oint$ signifies that the integration is carried out over the whole rotor. Using the expression for the power spectrum of rotor thrust in Equation 5.130, it can be shown that the variance of the thrust fluctuations within $\pm 10\%$ of blade passing frequency is about 1.4% of the total for the case considered. Although this is a small proportion (as can be seen from inspection of Figure 5.40), the standard deviation of thrust fluctuations in this frequency range is a much higher proportion of σ_T , that is, $\sqrt{0.014} \cong 12\%$. Denoting the standard deviation of thrust fluctuations within $\pm 10\%$ of blade passing frequency as $\sigma_{T.3p}$, we have

$$\sigma_{T.3p} \cong 0.1 \left(\frac{1}{2} \rho \Omega^2 \frac{dC_l}{d\alpha} \right) \sigma_u \oint cr dr \quad (6.15)$$

How does this compare with the maximum steady operational thrust load? Considering a pitch-regulated machine operating at rated wind speed, the rotor thrust is approximately

$$T = \frac{1}{2} \rho \Omega^2 \oint C_l(r) . cr^2 dr$$

and its maximum steady value can be estimated by setting $C_l(r)$ equal to 1.5, giving

$$T_{Max} = \frac{1}{2} \rho \Omega^2 1.5 \oint cr^2 dr \quad (6.16)$$

Hence the ratio of thrust fluctuations within $\pm 10\%$ of blade passing frequency to the maximum steady thrust is

$$\frac{\sigma_{T.3p}}{T_{Max}} \cong \frac{0.1 \left(\frac{1}{2} \rho \Omega^2 \frac{dC_l}{d\alpha} \right) \sigma_u \oint cr dr}{\frac{1}{2} \rho \Omega^2 1.5 \oint cr^2 dr} = \frac{0.1 \left(\frac{dC_l}{d\alpha} \right) \sigma_u \oint cr dr}{1.5 \Omega \oint cr^2 dr} \quad (6.17)$$

Setting $dC_l/d\alpha$ equal to 2π and noting that $\oint cr^2 dr / \oint cr dr$ is approximately equal to $0.6R$, one obtains:

$$\frac{\sigma_{T.3p}}{T_{Max}} \cong \frac{0.1(2\pi)\sigma_u}{1.5\Omega(0.6R)} \cong 0.7 \frac{\sigma_u}{\Omega R} \quad (6.18)$$

If the standard deviation of turbulence, σ_u , is 1.8 m/s and the tip speed is 65 m/s, the moment ratio approximates to 0.02.

Noting that the damage equivalent fatigue stress range for a material with an $S-N$ curve log-log plot inverse slope of $m = 4$ is 3.36 times the standard deviation of stress for a narrow banded process (see Equation 11.68 in Chapter 11), it is seen that, in relation to the design of the tower against fatigue, the damage equivalent load range of the thrust fluctuations close to blade passing frequency is about 6.5% of the maximum thrust in the case considered.

6.14.2 Tower top moment fluctuations due to blade pitch errors

Unintended differences between the pitches of the three blades will cause a permanent difference in the blade root moments M_{Y1} , M_{Y2} and M_{Y3} (see Figure 5.36), which will translate into a rotating moment applied to the nacelle. This in turn will impose a sinusoidal M_Y moment on the tower top at the rotational frequency.

Consider a turbine with blade pitch errors of $\Delta\theta$, $-\Delta\theta$ and zero on blades one, two and three. The angle of attack is reduced by $\Delta\theta$ on blade one, changing the blade root moment by

$$\Delta M_{Y1} = \frac{1}{2} \rho \Omega^2 \frac{dC_l}{d\alpha} \{-\Delta\theta\} \int_0^R cr^3 dr \quad (6.19)$$

The fluctuating tower top moment reaches its maximum and minimum when blades one and two are at 30° to the vertical and its amplitude is

$$\Delta M_Y = \frac{1}{2} \rho \Omega^2 \cdot 2\pi \{\Delta\theta\} \int_0^R cr^3 dr (2 \cos 30^\circ) \quad (6.20)$$

It is instructive to compare this moment with the average value in the tower due to the maximum steady thrust. At a depth of R below the hub, the moment ratio is

$$\frac{\Delta M_Y}{T_{Max} R} = \frac{\frac{1}{2} \rho \Omega^2 \cdot 2\pi \{\Delta\theta\} \int_0^R cr^3 dr (\sqrt{3})}{\frac{1}{2} \rho \Omega^2 1.5 \oint cr^2 dr R} \cong \frac{2\pi \{\Delta\theta\} 0.7 (\sqrt{3})}{1.5(3)} \cong 1.7 \{\Delta\theta\} \quad (6.21)$$

as $\int_0^R cr^3 dr / \oint cr^2 dr$ is approximately equal to $0.7 R/3$.

The moment ratio equates to 0.9% if the default blade pitch error of 0.3° specified in the 2003 edition of the GL ‘Guidelines for the certification of wind turbines’ is used. Thus, the moment *range* at rotational frequency due to this blade pitch error is about 1.8% of that due to rated thrust at a depth R below the hub – that is, significantly less than the damage equivalent moment range at this location due to stochastic thrust loading at blade passing frequency of about 6.5%.

6.14.3 Tower top moment fluctuations due to rotor mass imbalance

Unintended differences between the masses of the three blades will result in a sinusoidal gravitational moment about the low speed shaft axis, which will be transmitted to the tower. While IEC 61400–1 Edition 3 and the 2003 edition of the GL ‘Guidelines for the certification of wind turbines’ stipulate that the rotor mass imbalance taken into account should be based on the manufacturer’s specification, the 1999 edition of the latter specified a rotor mass eccentricity of $0.005R$ in the case of a ‘balanced’ rotor, which will be considered here.

In the case of an 80 m diameter turbine, the blade mass is typically about 7.5 tonnes, so, taking the hub mass equal to the that of three blades, the total rotor mass comes to 45 tonnes. With an eccentricity of $0.005 \times 40 = 0.2$ m, the tower top side-to-side moment *range* comes to about 180 kNm.

As before, it is instructive to compare this moment with the average value in the tower due to the maximum steady thrust. Taking the latter as 250 kN (Figure 5.39), the moment in the tower at a depth of R below the hub is 10,000 kNm. Hence the tower moment *range* at rotational frequency due to rotor mass imbalance is about 1.8% of that due to rated thrust at a depth R below the hub.

Based on the above it would appear that the impact of mass imbalance is much the same as that of blade pitch error for the mass eccentricity and pitch error considered. However, when dynamic magnification is taken into account, the effects of mass imbalance are potentially much more damaging, because the damping ratio for side-to-side tower oscillations may be an order of magnitude less than for fore-aft oscillations. This is because aerodynamic damping makes a negligible contribution to the damping of side-to-side motion.

6.14.4 Tower stiffness categories

Wind turbine towers are customarily categorised according to the relationship between the tower natural frequency and the exciting frequencies. Towers with a natural frequency greater than the blade passing frequency are said to be stiff, while those with a natural frequency lying between rotational frequency and blade passing frequency are said to be soft. If the natural frequency is less than rotational frequency, the tower is described as soft-soft.

If the tower is designed to meet strength requirements and no more, its frequency category is primarily determined by the ratio of tower height to turbine diameter, with the higher ratios producing the softer towers. The principal benefits of stiff towers are modest – they allow the turbine to run up to speed without passing through resonance, and tend to radiate less sound. However, since stiff towers usually require the provision of extra material not otherwise required for strength, soft towers are generally preferred.

6.15 Personnel safety and access issues

An integral part of wind turbine design is the inclusion of the necessary safety equipment for operation and maintenance staff. Minimum requirements include the following:

- Provision of ‘Emergency Stop’ push buttons located at key locations in the tower, nacelle and hub to enable personnel to stop the turbine and its systems operating in the event of an emergency.
- Provision of a ‘Remote/Local’ switch placed at the bottom of each tower. This enables a technician to take full control of the turbine when entering to carry out maintenance by changing the switch to the ‘Local’ position. This eliminates the risk of a third party trying to command the turbine to restart remotely.
- Provision of a fall-arrest system on the tower ladder(s). This consists of a fall arrester that slides on either a steel cable running the full length of the ladder in the middle or on a rigid rail bolted to the ladder in sections. In normal use the fall arrester is pulled up or lowered down by the anchor line attached to the climber. The tension in this anchor line releases a clamp, which locks onto the cable or rail again in the event of a fall.
- Provision of intermediate landings or smaller rest platforms in the tower to allow personnel to rest while climbing.

- An alternative means of egress from the nacelle, for use in case of fire in the tower. This can take the form of an inertia reel device, enabling personnel to lower themselves through a hatch in the nacelle floor.
- Locking devices for immobilising the rotor and the yawing mechanism. Rotor brakes and yaw brakes are not considered sufficient, because of the risk of accidental release and the occasional need to deactivate them for maintenance purposes. Ideally, the rotor locking device should act on the low speed shaft, so that its effectiveness is not dependent on the integrity of the gearbox. However, it is usually physically easier to engage a rotor lock acting on the high speed shaft. Typically the device consists of a pin mounted in a fixed housing, which can be engaged in a hole in a shaft-mounted disc.
- Guards to shield any rotating parts within the nacelle.
- Suitable fixtures for the attachment of safety harnesses for personnel working outside the nacelle.

Careful attention needs to be paid to the route between the tower top and nacelle to avoid hazards arising from sudden yawing movements. Some modern turbines have safety systems in place that only allow access to the nacelle in the event that the turbine has been shut down.

The designer needs to assess the requirement for all weather access to the nacelle at an early stage. Lattice towers afford no protection from the weather when climbing, so the number of days on which access for maintenance is possible will be restricted. Similar restrictions will arise if the nacelle cover has to be opened to the elements in order to provide space for personnel to enter.

Consideration also needs to be given to the means of raising and lowering tools and spares. If the interior of the tower is interrupted by intermediate platforms, these operations have to be performed outside, with consequent weather limitations. Many wind turbines are equipped with lifting hoists and/or cranes for this purpose.

With the increase of turbine diameters and, consequently, of tower heights, unassisted climbing of towers is becoming a physically demanding activity. Accordingly, EN 50308:2004 'Wind turbines – Protective measures: Requirements for design, operation and maintenance' stipulates lifts as the preferred method of turbine access as opposed to ladders. However, the reduced diameter at the top of tapering towers usually means it is impracticable for a lift to extend right to the top of the tower and, in any case, a significant number of ladder climbs are still required during lift installation and maintenance. In larger turbines, generally over 60 m hub height, the provision of lifts are a legislative requirement in some countries with a ladder only used in the event of an emergency.

An alternative approach to reducing the physical demands on maintenance personnel is the use of a climb assist device, whereby a cable attached to a powered hoist bears a significant proportion of the climbers bodyweight during ascent.

A TUV/NEL report (2007) provides a useful review of turbine access options.

Standard rules for electrical safety apply to all electrical equipment. However, particular care must be taken with the routing of electrical cables between tower and nacelle, in order to avoid potential damage due to chafing when they twist. If the power transformer is located in the tower base or nacelle instead of in a separate enclosure at ground level, it should be partitioned off to minimise the fire risk to personnel.

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Component design

7.1 Blades

7.1.1 Introduction

A successful blade design must satisfy a wide range of objectives, some of which are in conflict. These objectives can be summarised as follows:

1. Maximise annual energy yield for the specified wind speed distribution.
2. Limit maximum power output (in the case of stall regulated machines).
3. Resist extreme and fatigue loads.
4. Restrict tip deflections to avoid blade/tower collisions (in the case of upwind machines).
5. Avoid resonances.
6. Minimise weight and cost.

The design process can be divided into two stages: the aerodynamic design, in which objectives 1 and 2 are satisfied, and the structural design. The aerodynamic design addresses the selection of the optimum geometry of the blade *external* surface – normally simply referred to as the *blade geometry* – which is defined by the aerofoil family and the chord, twist and thickness distributions. The structural design consists of blade material selection and the determination of a structural cross section or *spar* within the external envelope that meets objectives 4–6. Inevitably there is interaction between the two stages, as the blade thickness needs to be large enough to accommodate a spar which is structurally efficient.

The focus of Section 7.1 is on blade structural design. After a brief consideration of the aerodynamic design in Section 7.1.2, practical constraints on the optimum design are noted in Section 7.1.3 and forms of blade structure surveyed in Section 7.1.4. An overview of the properties of some potential blade materials is given in Section 7.1.5 and the properties of glass fibre reinforced plastic (GFRP) and laminated wood are considered in more detail in

Sections 7.1.6 and 7.1.7. Governing load cases are considered in Section 7.1.8 with reference to both stall and pitch regulated machines. Subsequent sections touch upon blade resonance, panel buckling design and blade root fixings.

7.1.2 Aerodynamic design

The aerodynamic design encompasses the selection of aerofoil family and optimisation of the chord and twist distributions. The variation of thickness to chord ratio along the blade also has to be considered, but this ratio is usually set at the minimum value permitted by structural design considerations, as this minimises drag losses.

A survey of aerofoil families designed for wind turbine use is given in Section 3.17.

The process for optimising the blade design of machines operating at a fixed tip speed ratio is described in Section 3.7.2, where analytical expressions for the blade geometry parameter,

$$\sigma_r \lambda C_l = \frac{Nc(\mu)}{2\pi R} \lambda C_l$$

and the local inflow angle, ϕ , are derived as a function of the local tip speed ratio, $\lambda\mu = \lambda r/R$ (Equations 3.69a and 3.70a). If $\lambda\mu \gg 1$, the expressions can be approximated by

$$\sigma_r \lambda C_l = \frac{Nc(\mu)}{2\pi R} \lambda C_l = \frac{8}{9\lambda\mu} \quad \text{and} \quad \phi = \frac{2}{3\lambda\mu} \quad (7.1)$$

If it is decided to maintain the angle of attack, α , and hence the lift coefficient, C_l , constant along the blade, then these relations translate to

$$c(\mu) = \frac{16\pi R}{9C_l N \lambda^2} \cdot \frac{1}{\mu} \quad \text{and} \quad \theta = \frac{2}{3\lambda\mu} - \alpha \quad (7.2)$$

so that both the chord and twist are inversely proportional to radius.

In the case of machines operating at constant rotational speed, and hence at varying tip speed ratio, no parallel analytical solution for the optimum blade geometry exists. Instead resort must be made to numerical methods based on Blade Element – Momentum theory – for example, using Equations 3.54b and 3.55a in Section 3.8.6.

For pitch regulated machines, the annual energy capture attributed to the annular ring swept out by each blade element is determined for the chosen wind speed distribution, and the variation of energy capture with blade chord and twist at each ‘blade station’ computed. In this way the values of blade chord and twist at each ‘blade station’ yielding maximum energy capture are identified.

For stall regulated machines, the method is similar, but the total annual energy capture has to be maximised within the constraint of limiting the maximum total power output to the machine rating. The results of such an investigation are reported in ‘A design study of a 1 MW stall regulated rotor’ by Fuglsang and Madsen (1995).

7.1.3 Practical modifications to optimum design

The result of the optimisation described in the previous section is typically a blade geometry in which both blade chord and blade twist vary approximately inversely with radius, as

illustrated in Figure 3.19, Chapter 3. However, because the inboard section of the blade makes only a small contribution to total power output (Figure 3.39, Chapter 3), the aerofoil section is generally not continued inboard of about 15% radius in practice, and the chord at this radius is substantially reduced, to perhaps half the theoretical optimum. It is then often found expedient to taper the chord uniformly over the active length of the blade, with the tip chord and chord taper set so that the chord distribution approximates closely to the optimum over the outboard half of the blade (Figure 3.20, Chapter 3).

The blade root area is normally circular in cross-section in order to match up with the pitch bearing in the case of pitchable blades, or to allow pitch angle adjustment at the bolted flange (to compensate for non-standard air density) in the case of stall regulated blades. The transition from the root section to the aerofoil section outboard of 15% radius should be a smooth one for structural reasons, with the result that the latter section will have a high thickness to chord ratio of up to about 50%.

7.1.4 Form of blade structure

A hollow shell corresponding to the defined blade envelope clearly provides a simple, efficient structure to resist flexural and torsional loads and some blade manufacturers adopt this form of construction (see Figure 7.1). However, in the case of small and medium size machines, where the out-of-plane loads dominate, there is greater benefit in concentrating skin material in the forward half of the blade, where the blade thickness is a maximum, so that it acts more efficiently in resisting out-of-plane bending moments (see Figures 7.2 and 7.3). The weakened areas of the shell towards the trailing edge are then typically stiffened by means of sandwich construction utilising a PVC foam filling.

The hollow shell structure defined by the aerofoil section is not very efficient at resisting out-of-plane shear loads, so these are catered for by the inclusion of one or more shear webs oriented perpendicular to the blade chord.

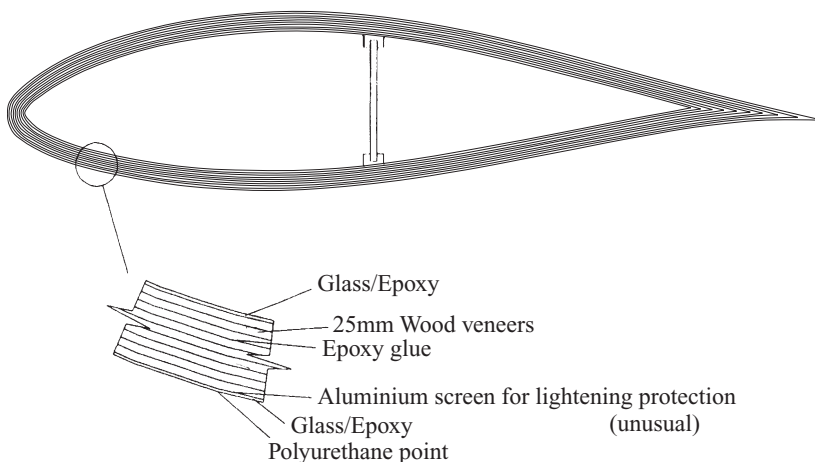


Figure 7.1 Wood/epoxy blade construction utilizing full blade shell. Reproduced from Corbet (1991) by permission of the DT1 Renewable Energy R&D Programme

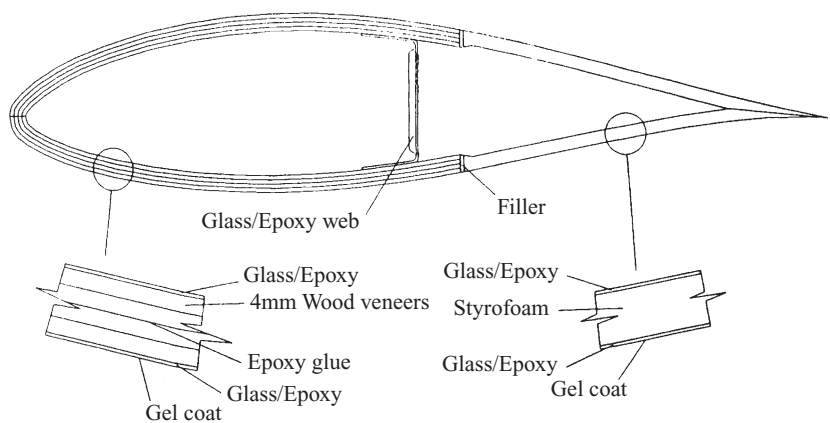


Figure 7.2 Wood/epoxy blade construction utilizing forward half of blade shell. Reproduced from Corbet (1991) by permission of the DT1 Renewable Energy R&D Programme

If the load-bearing structure is limited to a compact closed hollow section spar, consisting of two shear webs and the skin sections between them (see Figure 7.4), then a GFRP blade lends itself to semi-automatic lay-up on a rotating mandrel which can be withdrawn after curing.

7.1.5 Blade materials and properties

The ideal material for blade construction will combine the necessary structural properties – namely high strength to weight ratio, fatigue life and stiffness – with low cost and the ability to be formed into the desired aerofoil shape.

Table 7.1 lists the structural properties of the materials in general use for blade manufacture and those of some other candidate materials. For comparative purposes, values are also presented of:

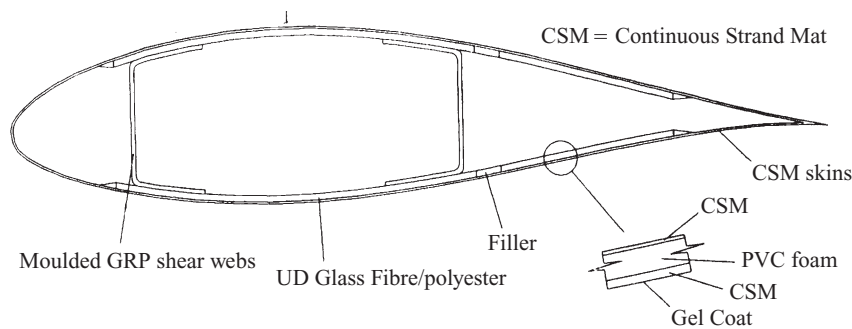


Figure 7.3 Glass-fibre blade construction using blade skins in forward portion of blade cross section and linking shear webs. Reproduced from Corbet (1991), by permission of the DT1 Renewable Energy, R&D Programme

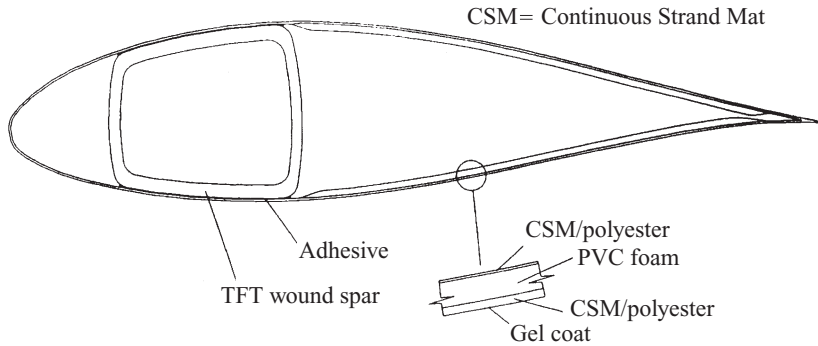


Figure 7.4 Glass-fibre blade construction using compact spar wound with transverse filament tape (TFT) on mandrel. Reproduced from Corbet (1991), by permission of the DT1 Renewable Energy R&D Programme

- compressive strength to weight ratio;
- fatigue strength as a percentage of compressive strength;
- stiffness to weight ratio; and
- a panel stability parameter, $E/(UCS)^2$.

It is evident that glass and carbon fibre composites (GFRP and CFRP) have a substantial higher compressive strength to weight ratio compared with the other materials. However, this apparent advantage is not as decisive as it appears, for two reasons. First of all, the fibres of some of the plies making up the laminated blade shell have to be aligned off-axis (typically at $\pm 45^\circ$) to resist shear loads, giving reduced strengths in the axial direction. Secondly, the relatively low Young's modulus of these composites means that resistance to buckling of the thin skins governs the design rather than simple compression yielding. The likelihood that buckling will govern is inversely related to the panel stability parameter, $E/(UCS)^2$, given in the last column of Table 7.1, so that materials with high values, such as wood composites will be least sensitive to buckling. As a result wood composite blades are generally lighter than equivalent glass fibre composite blades. Design against buckling is considered in Section 7.1.10.

It should be noted that the low strength of wood laminate compared with other materials renders it unsuitable for blades with slender chords operating at high tip speed, where the flapwise bending moments during operation are inevitably high in relation to blade thickness. For example, Jamieson and Brown (1992) have shown that, in the case of a family of stall-regulated machines, the blade stress is highly sensitive to rotational speed, increasing as the fourth power, if the skin thickness to chord ratio is maintained constant. Although stresses can be reduced by increasing the skin thickness, this represents a less and less efficient use of the additional material beyond a skin thickness to chord ratio of 3–4%, especially in the outboard part of the blade, where the blade thickness to chord ratio is low.

Fatigue performance is conveniently measured by mean fatigue strength at 10^7 cycles, as a percentage of ultimate compressive strength. Clearly, carbon fibre and khaya/epoxy perform best with a value of about 30%. The low value for welded steel (10%), combined with steel's

Table 7.1 Structural properties of materials for wind turbine blades

Material (NB: UD denotes unidirectional fibres – i.e., all fibres running longitudinally)	Ultimate tensile strength (UTS) (MPa)	Ultimate compressive strength (UCS) (MPa)	Specific gravity (s.g.)	Compressive strength to weight ratio UCS/s.g.	Mean fatigue strength at 10 ⁷ cycles (amplitude) (MPa)	Mean fatigue strength as percentage of UCS	Young's Modulus, <i>E</i> (GPa)	Stiffness to weight ratio <i>E</i> /s.g. (GPa)	Panel stability parameter <i>E</i> /(UCS) ² (MPa) ⁻¹
	(mean for composites, minimum for metals)								
1. Glass/polyester <i>ply</i> with 50% fibre volume fraction and UD lay-up	860–900 [1][2]	~720 [1]	1.85	390	140 [3]	19%	38 [2]	20.5	0.07
2. Glass/epoxy <i>ply</i> with 50% fibre volume fraction and UD lay-up	Properties are generally very close to those for GRP given above								
3. Glass/polyester <i>laminates</i> with 50% fibre volume fraction and 80% of fibres running longitudinally	690–720	~580	1.85	310	120	21%	33.5	18	0.1
4. Carbon fibre/epoxy <i>ply</i> with 60% fibre volume fraction and UD lay-up	1830 [4]	1100 [4]	1.58	700	350 [5]	32%	142 [4]	90	0.12

5. Khaya ivorensis/epoxy laminates	82 [6]	50 [6]	0.55	90	15 [7]	30%	10 [8]	18	4
6. Birch/epoxy laminates	117 [9]	81 [10]	0.67	121	16.5 [7]	20%	15 [10]	22.5	2.3
7. High Yield Steel (Grade Fe 510)	510	510	7.85	65	50 [11]	10%	210	27	0.81
8. Weldable aluminium alloy AA6082 (formerly) H30	295 [12]	295 [12]	2.71	109	17 [13]	6%	69 [12]	25.5	0.79

Sources:

[1] Mayer (1996) Figure 2.4.
[2] Barbero (1998) Table 1.1.
[3] Mayer (1996) Fig. 14.4.
[4] Carbon fibres exhibit a wide range of properties; figures given here are for one example only, taken from [2].
[5] Based on $S-N$ curve index of $m = 14$, taken from GL rules.
[6] Bonfield and Ansell (1991) Moisture content = 10%
[7] Based on $S-N$ curve index of $m = 13.4$ for scarf-jointed wood laminates, taken from Hancock and Bond (1995).
[8] Bonfield *et al.* (1992).
[9] Mayer (1996) Table 7.3.
[10] Hancock (Personal Communication). Moisture content = 10%.
[11] Mean value for butt-welded joints with weld profile ground smooth (Class C), taken from BS 5400, Part 10 (1980).
[12] CP 118: 1969 'Code of practice for structural aluminium'.
[13] Mean value estimated from mean minus two standard deviations value for ground butt welded joint with shallow thickness transition, Detail Cat 221, in IIW 'Fatigue design of welded joints and components'

low strength to weight ratio, renders it uncompetitive for large diameter machines where gravity fatigue loading becomes important, although it was chosen for some of the early prototype megawatt scale machines when the fatigue properties of composite materials were less well understood.

The stiffness to weight ratio determines blade natural frequency. Apart from CFRP, the values in the table are all in a relatively small range (18–27 GPa), indicating that material choice will generally only have a marginal effect on dynamic behaviour.

From the above brief survey, it is apparent that the material with the best all-round structural properties is carbon fibre composite. However, it has not found common use, because it is an order of magnitude more costly than other materials. Instead, the most popular material is glass/polyester, followed by glass/epoxy and wood/epoxy.

Steel is the cheapest material in the raw state, and can be formed into tapering, curved panels following the aerofoil profile, except in the sharply curved region near the leading edge. However, it is much harder to introduce a twist into such panels, and this consideration, together with the poor fatigue properties, means that steel is rarely used. By contrast, glass and carbon fibre composites lend themselves to wet lay-up in half-moulds profiled to give the correct aerofoil shape, planform and twist. Laminated wood composite blades are built up in a similar way, but the veneer thickness has to be restricted to enable the veneers to flex to the required curvature during lay-up.

In the following paragraphs, the properties of the materials in most common use for blade manufacture are considered in more detail.

7.1.6 Properties of glass/polyester and glass/epoxy composites

As noted in Table 7.1, the properties of glass/polyester and glass/epoxy plies with the same fibre volume fraction and lay-up are generally very similar – that is, the influence of the matrix is slight. They will, therefore, be treated as the same material in the discussion that follows, except in relation to fatigue where some differences have been noted. The glass used in blade construction is E-glass, which has good structural properties in relation to its cost.

The plate elements forming the spar of a GFRP blade are normally laminates consisting of several plies, with fibres in different orientations to resist the design loads. Within a ply (typically 0.5–1.0 mm in thickness), the fibres may all be arranged in the same direction – that is, UD or unidirectional – or they may run in two directions at right angles in a wide variety of woven or non-woven fabrics.

Although the strength and stiffness properties of the fibres and matrix are well defined, only some of the properties of a ply can be derived from them using simple rules. Thus, for a ply reinforced by UD fibres, the longitudinal stiffness modulus, E_1 , can be derived accurately from the rule of mixtures formula

$$E_1 = E_f V_f + E_m (1 - V_f) \quad (7.3)$$

where E_f is the fibre modulus (72.3 GPa for E-Glass), E_m is the matrix modulus (in the range 2.7–3.4 GPa) and V_f is the fibre volume fraction. On the other hand, the inverse form of this formula – for example,

$$\frac{1}{E_2} = \frac{(1 - V_f)}{E_m} + \frac{V_f}{E_f} \quad (7.4)$$

significantly underestimates the transverse modulus, E_2 , and the in-plane shear modulus, G_{12} . More accurate formulae based on more sophisticated models are given in Barbero (1998).

The longitudinal tensile strength of a ply reinforced by UD fibres, σ_{1t} , can be estimated from

$$\sigma_{1t} = \sigma_{fu} \left[V_f + \frac{E_m}{E_f} (1 - V_f) \right] \quad (7.5)$$

where σ_{fu} is the ultimate tensile strength of the fibres. However, the tensile strengths of E-glass single fibres (3.45 GPa) cannot be realised in a composite, where fibre strength reductions of up to 50% have been measured. Accordingly, a value of σ_{fu} of 1750 MPa should be used in Equation 7.5.

The longitudinal compressive strength of a ply reinforced by UD fibres is always significantly less than the tensile strength because of microbuckling of the fibres, which is governed by the shear strength of the matrix and the degree of fibre misalignment. A strength reduction of at least 15% should be allowed for, assuming minimum fibre misalignment.

Clearly, longitudinal stiffness and strength are both limited by the fibre volume fraction obtainable. For hand lay-up, fibre volume contents of 30–40% are typical, but the use of ‘vacuum bagging’, in which trapped air and excess volatile compounds, such as residual solvent, are extracted, consolidates the composite and allows a volume fraction of 50% or more to be achieved. The use of pre-pregs, which are unidirectional fibres or woven fabric pre-impregnated with partially cured epoxy resin, results in similar increased fibre volume fractions.

Fatigue properties

When expressed in terms of stress, the fatigue properties of composite laminates extend over a wide range, depending on fibre volume fraction and the number of plies with fibres in the longitudinal direction. However, data from constant stress amplitude fatigue test results becomes much more intelligible if stress ranges are converted into initial strain ranges, allowing the fatigue properties of composites with different lay-ups to be compared. (The Young’s modulus of a composite reduces over time during a fatigue test – hence the need to specify that the strain range is measured at the start of the test).

The fatigue behaviour of composites depends on both the stress range *and* the mean stress level, which can both be described in terms of the maximum stress, $\sigma_{\max}$, and the ratio of minimum to maximum stress, R . It is convenient initially to consider fatigue behaviour under reverse loading, that is, with $R = -1$, for which the mean stress is zero, and then relate behaviour at other R ratios to it.

The constant amplitude fatigue behaviour of glass fibre composites can best be characterised by a linear relationship between the logarithm of the number of cycles and the logarithm of the stress or strain amplitude, viz:

$$\varepsilon = \varepsilon_0 N^{-1/m} \quad \text{or,} \quad N = K \varepsilon^{-m} \quad \text{where} \quad K = (\varepsilon_0)^m \quad \text{or,} \quad \log N = \log K - m \log \varepsilon \quad (7.6)$$

Echtermeyer, Hayman and Ronold (1996) carried out a regression analysis on a total of 111 constant amplitude, reverse loading fatigue test results for ten different laminates tested at

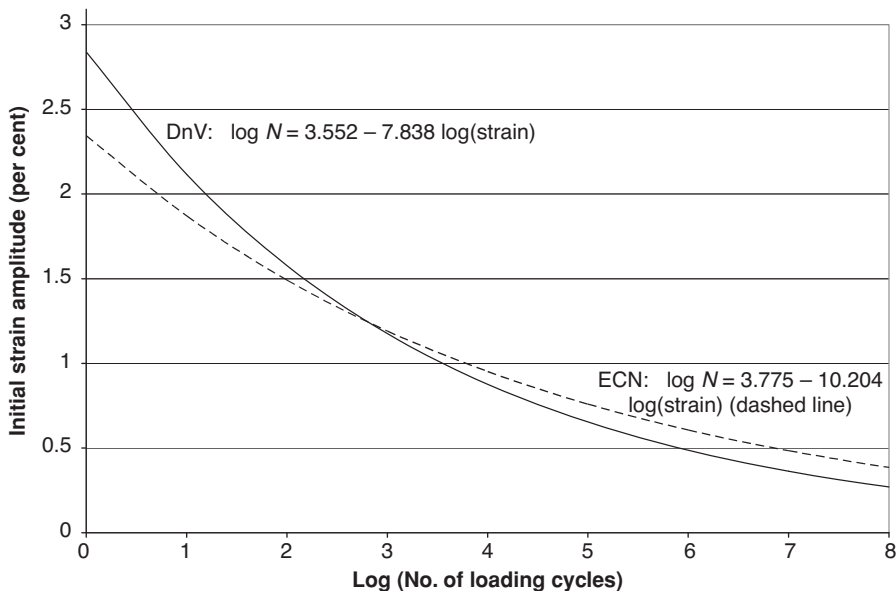


Figure 7.5 Strain-life regression lines fitted to results of constant amplitude, reverse loading fatigue tests on GFRP composites

DnV, assuming that they all conformed to the same $\varepsilon - N$ curve, and obtained values for ε_0 , $\log K$ and m of 2.84%, 3.552 and 7.838 respectively, with a standard deviation of $\log N$ of 0.437. The DnV regression line is compared with another derived from 19 tests on a $0^\circ/+45^\circ$, -45° laminate at ECN, giving $\varepsilon_0 = 2.34\%$, $\log K = 3.775$ and $m = 10.204$, in Figure 7.5. The researchers did not constrain the regression lines to pass through the strain value at either UTS or UCS at $\log N = 0$ (ca 2.4% and 2.0%): had they done so the DnV line would have had a shallower slope – that is, a larger value of m . After comparison with regressions on other fatigue test datasets, they concluded that the DnV line provided a reasonable basis for initial design.

Constant amplitude tests at other R ratios generally show reducing fatigue lives as the mean stress increases above zero – whether in tension or in compression. It is customary to represent the results on a constant life diagram (also known as a Goodman diagram), in which the stress range to failure is plotted against mean stress for different fatigue lives. Regression analyses can be carried out on families of test results at different R ratios to give a series of $\varepsilon - N$ relations in the form of Equation 7.6 which can be used to plot the constant life diagram. Such an exercise was carried out on the Dutch ‘FACT’ database of fatigue tests on composites for wind turbines (Joose and van Delft, 1996), and some results of this work have been reproduced in Figure 7.6 (dashed lines).

In the preparation of design rules, it is common practice to make the simplifying assumption that the strain amplitude reduces linearly with increasing mean strain for a given fatigue life, reaching zero at a mean strain corresponding to either the ultimate tensile or compressive strength. Such linear constant life lines are shown in Figure 7.6 as heavy lines. Constant life lines for design are obtained from characteristic material properties divided by appropriate

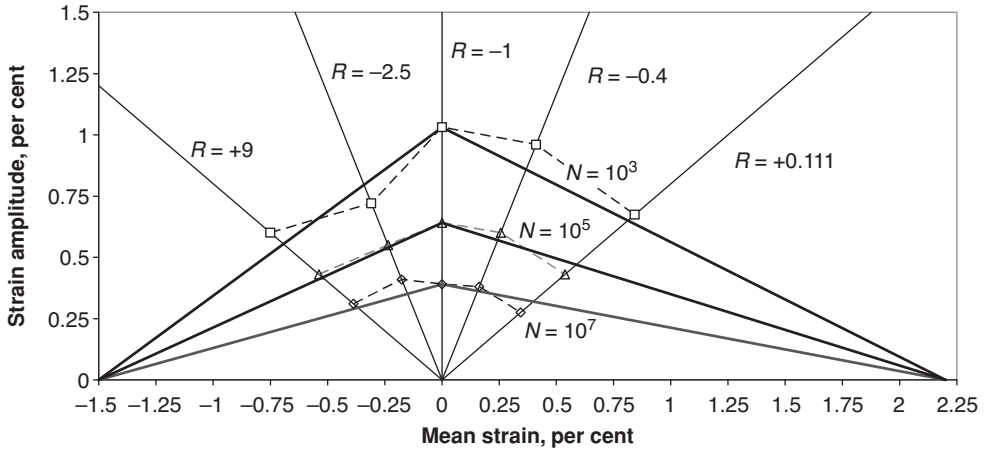


Figure 7.6 Constant life diagram showing variation in fatigue strain amplitude with mean strain for lives of 10^3 , 10^5 and 10^7 cycles for GFRP composites. Dashed lines show experimental results and full lines show linearised formulations for design

partial safety factors, as opposed to the characteristic properties used in Figure 7.6. Thus, the design strain amplitude when the mean stress is compressive becomes

$$\varepsilon_d(\bar{\sigma}, N) = \varepsilon_{0d} N^{-\frac{1}{m}} \left(1 - \frac{\bar{\sigma}}{\sigma_{cd}} \right) \quad (7.7)$$

where $\varepsilon_{0d} = \varepsilon_{0k}/\gamma_{mf}$, $\sigma_{cd} = \sigma_{ck}/\gamma_{mu}$, ε_0 is the value of ε given by the $\varepsilon - N$ curve when $\log N$ is zero, $\bar{\sigma}$ is the mean stress for the loading cycles under consideration, and σ_{cd} is the design compressive stress. γ_{mf} is the partial safety factor for fatigue strength, γ_{mu} is the partial safety factor for ultimate strength, and the suffices d and k signify design and characteristic values respectively.

Equation 7.7, together with its equivalent for mean tensile loading, can be used to calculate the permissible number of load cycles, N_i , for each strain range in the fatigue loading spectrum for the point in the blade cross section under examination – incorporating the appropriate partial safety factor for the consequences of failure. These are then combined with the predicted number of cycles for each strain range, n_i , to yield Miner's damage sum, $\sum_i (n_i/N_i)$, which is normally required to be less than unity.

In the GL rules, under the 'simplified assumptions', the $\varepsilon - N$ curve for design takes the same form as Equation 7.6, but the constant life diagram is symmetrical about a strain level midway between the ultimate tensile and compressive strains, with ε_0 set equal to half the difference between the ultimate tensile and compressive strains. Partial materials safety factors are included. GL specifies different values of the index, m , for composites with polyester and epoxy matrixes – 9 and 10 respectively – although opinion is divided as to whether the difference is justified. The constant life diagram is illustrated in Figure 7.7 for $m = 10$.

The small circle on Figure 7.7 indicates that the number of cycles permitted for a strain range of $0.6/\gamma_M$ % and a factored mean strain of 1% would be 1000, where γ_M is the

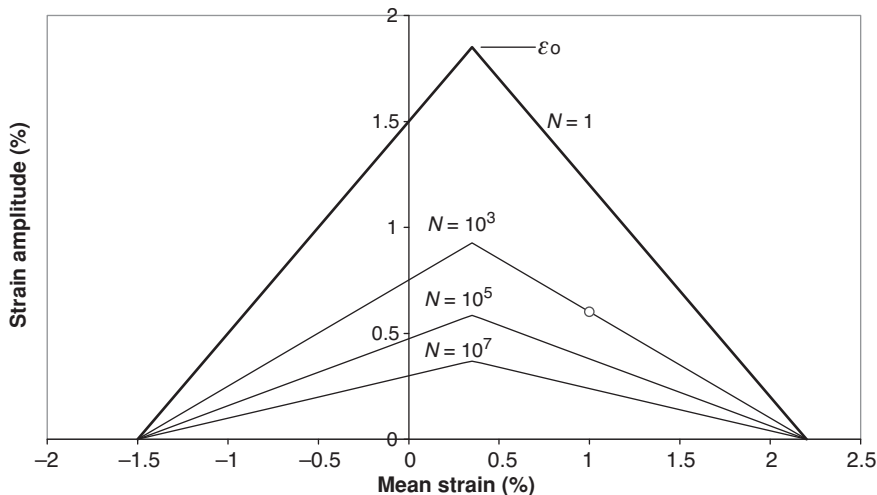


Figure 7.7 Constant life diagram based on GL rules (simplified assumptions) showing variation in fatigue strain amplitude with mean strain for lives of one, 10^3 , 10^5 and 10^7 cycles for GFRP composites

appropriate partial safety factor for material strength in fatigue and the mean strain is factored by the partial safety factor for material strength under *extreme* loads.

There is inevitably a degree of uncertainty as regards the accuracy of Miner’s rule in predicting the fatigue damage due to variable amplitude loading from constant amplitude test data. In order to investigate this, fatigue test programmes have been carried out using the WISPER (Wind SPectrum Reference) and WISPERX variable amplitude fatigue load spectra, which have been devised to be representative of those experienced by wind turbine blades. (WISPERX is a modification of WISPER in which the large number of small cycles, accounting for approximately 90% of the total, are omitted to reduce test durations.) For each test specimen, the WISPER (or WISPERX) load sequence is scaled to give a chosen maximum stress level and is applied repeatedly until the specimen fails.

Van Delft *et al.* (1996) analysed the results of a series of tests carried out at ECN and Delft Technical University on a 0° , $\pm 45^\circ$ laminate and found that, for a maximum stress of about 150 MPa, the actual fatigue lives of specimens subjected to repetitions of the WISPER or WISPERX load sequences were about one hundred times less than predicted for these sequences on the basis of constant amplitude, reverse loading test data and Miner’s rule, with the effect of mean stress allowed for using the linear relation described above. The $R = -1$ test data led to an $S-N$ curve given by $N = (\sigma/\sigma_{tu})^{-10}$, where σ is the amplitude of the stress cycles and σ_{tu} is the ultimate tensile strength, so the number of cycles to failure for constant amplitude loading at other R values was taken as $N = (\sigma/\sigma_{tu}(1 - \bar{\sigma}/\sigma_{tu}))^{-10}$ for a tensile mean and $N = (\sigma/\sigma_{tu}(1 - \bar{\sigma}/\sigma_{cu}))^{-10}$ for a compressive mean in calculating the Miner’s damage sum. The difference in fatigue lives at the stated maximum stress level quoted above translates to an approximate ratio of 1:1.5 between actual and predicted maximum stress levels of the WISPER sequence to cause failure over the design fatigue life, which would clearly use up a substantial proportion of the safety factors used in design. However, other investigators

working with different laminates have found reasonable agreement between measured and predicted fatigue lives under WISPER loading.

Material safety factors

Limit state design requires that the characteristic strength of a material be divided by a partial safety factor for material strength. In the case of GFRP, this factor needs to take account of degradation of the material over time, as well as the material's inherent variability.

The GL rules lay down that the material safety factor for calculating the design strength of GFRP under extreme loads is to be calculated as the product of:

- a basic factor, γ_{MO} , of 1.35;
- a factor, $C_{2a} = 1.35$, to account for the influence of ageing;
- a factor, $C_{3a} = 1.1$ to account for strength reduction at higher temperatures;
- a factor, $C_{4a} = 1.2$ for hand lay-up laminates or 1.1 where manufacture is partially automated;
- a factor, $C_{5a} = 1.1$ if the laminate is not post cured.

These rules result in a material safety factor in the range 2.2–2.65.

In the case of fatigue loads, the ageing factor of 1.35 is omitted and the factor accounting for lay-up is replaced by one taking account of the type of fibre reinforcement.

7.1.7 Properties of wood laminates

Although laminated wood/epoxy is classed as a composite, it is markedly different in form from GFRP. Individual plies are made up of large sheets of wood veneer (Figure 7.8) instead of a multiplicity of fibres laid up in a matrix, and the epoxy behaves as an adhesive rather than a matrix, bonding the sheets together at the longitudinal and transverse joints and bonding each ply to its neighbour. Thus, the fibre volume fraction is close to 100% and the anisotropic properties of the wood laminate derive principally from the anisotropic properties of the wood itself.

Wood strength properties are much greater in the direction parallel to the grain, so all the veneers are orientated with the grain parallel to the blade axis, in order to resist blade bending loads efficiently. However, the veneers cannot be produced in lengths much greater than 2.5 m, so transverse joints have to be included, which introduce lines of weakness not normally found in GFRP blades. The effect is minimised by staggering the joints, and by using scarf joints in preference to butt joints.

The epoxy adhesive has a secondary function of sealing the veneers against moisture ingress; additional moisture protection is provided by a layer of glass/epoxy on both the external and internal surfaces. It is important to maintain moisture content at a low level, because veneer strength decreases about 6% for every 1% rise in moisture content.

A comparison of some of the properties of wood laminates used, or considered for use, in wind turbine blades is given in Table 7.2. *Khaya ivorensis*, an African mahogany and Douglas fir used to be the main species used for blade manufacture in the UK and US respectively, but environmental pressures have led to the phasing out of *Khaya* in favour of European species such as poplar and birch.



Figure 7.8 Blade Production. View of veneer lay-up in mould to make one blade skin. The blade is completed by glueing face and camber skins together. Reproduced by permission of NEG-Micon. See Plate 3 for the colour figure

Table 7.2 gives tensile strengths of unjointed specimens. Bonfield *et al.* (1992) report the results of tests on jointed specimens, which showed a significant reduction in tensile strength to 50 MPa for butt jointed Khaya. Scarf jointed Khaya specimens, with a 1:6 length to thickness ratio, performed much better, achieving a tensile strength of 75 MPa. In all cases the joints in the different veneers making up the laminate were staggered.

An important consideration for design is the variability of strength properties, particularly as wood is an inherently variable material. Strength tends to increase with density, and density varies according to the growing conditions of the tree and the part of the tree from which the wood is taken. Such variability can be reduced by careful grading and the rejection of damaged veneers before laminating. Bonfield and Ansell (1991) report compression tests on 32 carefully selected Khaya samples which yielded the compression strength of 50 MPa given in the table with a standard deviation of only 3 MPa. It should be noted that the lack of annual growth rings in equatorially grown wood may reduce the degree of scatter. Wood strengths perpendicular to the grain are typically much less than those along the grain – for example, the compressive strength of transversely loaded Khaya is only 12.6 MPa.

Table 7.2 Properties of unjointed wood/epoxy laminates

Species	Specific gravity	Mean tensile strength along the grain (MPa)	Mean compression strength along the grain (MPa)	Young's modulus along the grain (GPa)	Shear strength (MPa)
Khaya ivorensis	0.55	82	50	10	9.5
Poplar	0.45	63	52	10	9
Baltic pine	0.55	105	40	16	
Birch	0.67	117	81	15	16
Beech	0.72	103	69	10	16
Douglas fir	0.58	100	61	15	12

Fatigue properties

The fatigue properties of wood laminates have been the subject of a sustained programme of work at Bath University, starting with Khaya and then extending to other species (Bonfield *et al.*, 1992). A useful summary of this work appears in Bond and Ansell (1998). The general conclusion is that wood performs very well in fatigue with a shallow S – N curve slope, and that fatigue strengths at high cycles do not vary greatly between species.

If the S – N curve for constant amplitude, reverse loading ($R = -1$) fatigue is normalised with respect to the ultimate compressive strength, σ_{cu} – that is, $\sigma = \sigma_{cu}N^{-1/m}$, then the results of tests on unjointed Khaya indicate a value of the index m of about 20. However the value of m reduces to about 16 for scarf jointed khaya, poplar and beech, and to about 13 for butt-jointed specimens. Hancock and Bond (1995) have proposed the use of an index of 13.4 for design purposes for scarf jointed wood laminates in general.

Testing at other R ($= \sigma_{\min}/\sigma_{\max}$) ratios allows constant life diagrams to be plotted – see, for example, the diagram for scarf jointed poplar in Figure 7.9, taken from Bond and Ansell (1998). Note the relatively low stress ranges at $R = -0.84$ ($=$ UCS/UTS), which may be due to simultaneous occurrence of compressive and tensile damage. Despite this, the simplification of the constant life diagram to a series of straight lines between the $R = -1$ stress range for each fatigue life and either the UTS or UCS is reasonably accurate.

Material safety factors

The material safety factor applied when timber is used in building construction is normally high – for example, about 3–4. However, there are a number of reasons for adopting a much lower value in blade design – as follows:

1. Laminated construction is used, so any defects are very localised.
2. The moisture content is carefully controlled during manufacture, and the blade skin is then very effectively sealed against further moisture ingress.
3. Creep effects are negligible as the gravity loads change direction because of blade rotation and the wind loads are temporary in nature.

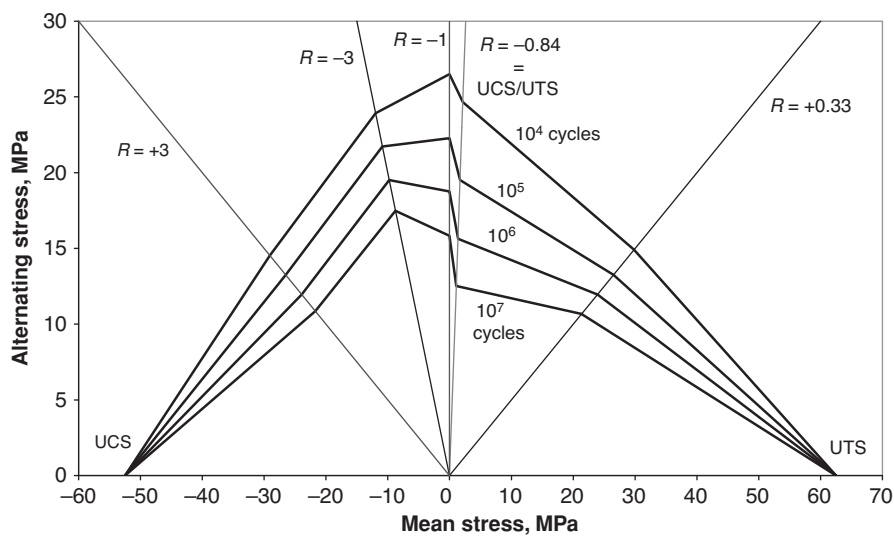


Figure 7.9 Constant life diagram for scarf jointed poplar derived from 50% median regression lines on *S–N* fatigue test data

Accordingly a partial safety factor of only about 1.5 is normal for design against extreme loads.

7.1.8 Blade loading overview

This section explores the variation of extreme and fatigue loading with wind speed and yaw angle, focussing on flapwise bending at 60% radius by way of example. The turbine considered is a 40 m diameter 500 kW machine, with the 20 m radius blades scaled down from the 40 m radius T40 blade shown in Figure 5.4(a).

Extreme loading during operation: stall-regulated machines

The stall-regulated machine considered operates at a single rotational speed of 30 rpm, with rated wind speed, V_r , and cut out speed, V_o , of 16 m/s and 25 m/s respectively. The shaft tilt with respect to the horizontal is taken as 5° , so, allowing for a 8° inclination of the flow to the horizontal, as specified in the code, the maximum shaft tilt with respect to the flow is 13° .

The blade loadings are calculated using empirical 3D data taken from Petersen, Madsen *et al.* (1998), with extrapolation of the lift and drag coefficient curves beyond 30° angle of attack (the upper limit of the data). The 3D data displays a gentler stall than typical 2D data, so there is no significant reduction in blade out-of-plane bending moment as the blade goes into stall. Above about 20 m/s, the out-of-plane bending moment begins to increase progressively once again as drag begins to become significant. The predicted variation of blade 12 m radius out-of-plane bending moment with wind speed is plotted out for a 0.2 shear exponent, zero shaft tilt and a range of yaw angles on Figure 7.10, with the yaw direction defined as positive

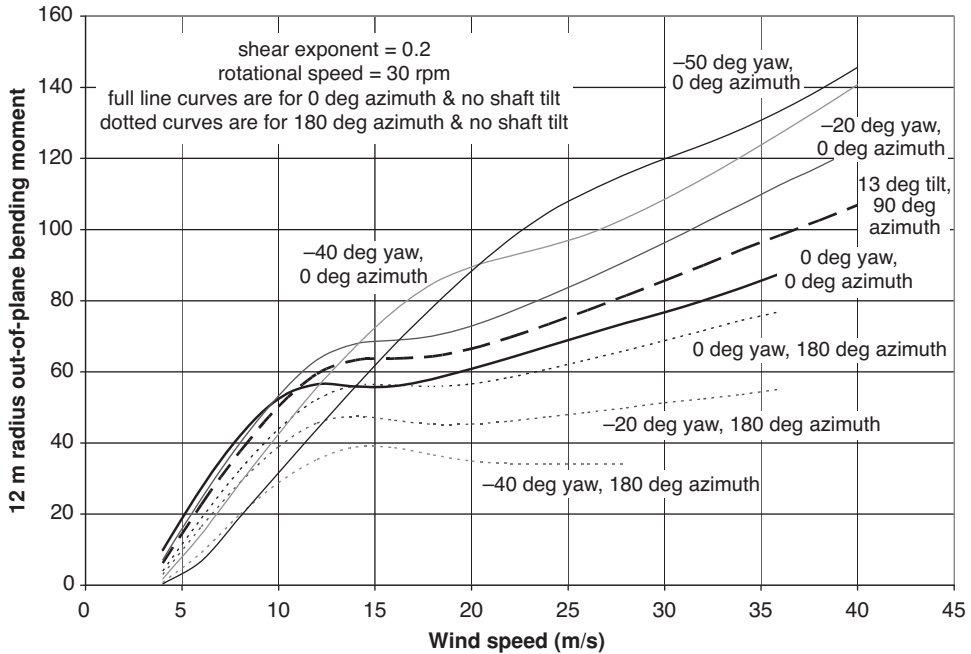


Figure 7.10 Variation of 12 m radius out-of-plane bending moment with wind speed at various yaw angles for an example 40 m diameter stall regulated machine

when the lateral component of air flow with respect to the rotor disc is in the same direction as the blade movement at zero azimuth (i.e. at 12 o'clock). The effect of this increase in relative velocity outweighs that of the reduction of angle of attack at wind speeds beyond stall, so the bending moment at 0° azimuth is increased by negative yaw. Maximum moments occur at negative yaw angles and 0° azimuth rather than at positive yaw angles and 180° azimuth, because wind shear augments the wind speed in the former case. Also plotted is the variation of bending moment with wind speed for a 13° shaft tilt with respect to the air flow and 90° azimuth, which is the critical configuration for load cases not involving a change in wind direction. The 13° shaft tilt with respect to the flow is made up of 5° shaft tilt relative to horizontal and an 8° inclination of the flow to the horizontal, as specified in IEC 61400-1.

The plots of the extreme 12 m radius out-of-plane bending moment in Figure 7.10 are conservative on three counts, because no allowance is made for the following:

- Lack of correlation of the wind over the outer 40% of the blade.
- Limitation on maximum wind speed seen during operation by high wind cut-out.
- Limitation on maximum yaw angle by yaw control.

The alleviation of extreme loadings by high wind cut-out and yaw control depend on the averaging times applied to the wind speed and direction signals by the control system.

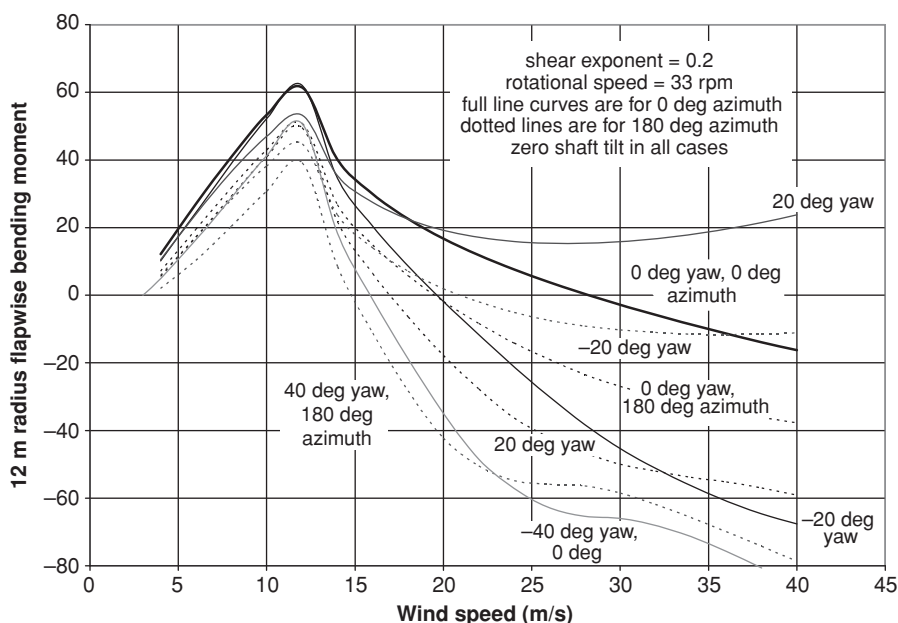


Figure 7.11 Variation of 12 m radius flapwise bending moment with short-term mean wind speed at various yaw angles, for an example 40 m diameter pitch regulated m/c

Extreme loading during operation: pitch-regulated machines

The characterisation of extreme operational loadings on pitch-regulated machines is inevitably more complicated than for stall-regulated machines, although at the same time it should be more accurate because of the avoidance of uncertainties associated with stall. It is instructive to focus comparisons on the blade bending moment about the weak axis at 60% radius once again. This time it is referred to as the flapwise bending moment rather than the out-of-plane (of rotation) moment because of blade pitching.

Figure 7.11 presents the variation of 60% radius flapwise bending moment with short-term mean wind speed at several yaw angles for a 500 kW, 40 m diameter pitch-regulated machine rotating at 33 rpm. The rated speed, V_R , is 12 m/s and other parameters, including the wind shear exponent, are the same as in the stall-regulated example above. The blade loadings are calculated using empirical 3D data taken from Petersen, Madsen *et al.* (1998) as before, with extrapolation of the lift and drag coefficient curves to negative angle of attack. The figure only shows the bending moments resulting from slow variations in wind speed – that is, those which can be followed by the pitch control system – so moments arising from faster wind speed fluctuations must be added to obtain the total.

The curves are very different in shape from those obtained for the stall-regulated machine. The 12 m radius flapwise bending moment reaches a peak at rated wind speed, and then drops off sharply, becoming negative by about 24 m/s in the case of zero yaw and zero wind shear. This is because the blade has pitched to such an extent that the outboard section of the blade is providing a braking torque to counteract the increased torque from the inboard section. At high wind speeds and yaw angles, large negative bending moments are developed, which

approach the magnitude of the peak positive moment at rated speed. Note that the bending moment *reduces* with negative yaw angle at zero azimuth, instead of increasing as it does for stall-regulated operation. This is because blade pitching renders angle of attack, which is reduced under these conditions, more critical than relative velocity. Plots of the variation of flapwise bending moment with short-term mean wind speed at inboard blade cross sections are essentially similar to those in Figure 7.11, because moments are dominated by loadings on the outboards portion of the blade.

To the extent that the pitch control system can keep pace with the wind speed transients, the curves in Figure 7.11 can be used to provide an approximate indication of the extreme BMs arising from some of the IEC 61400-1 deterministic load cases. It is seen that the extreme moments are only about one half of the maximum value for the stall-regulated machine.

The spectrum of the longitudinal wind speed fluctuations will contain significant energy at frequencies above the level at which the pitch control system can respond, and these have to be considered in the analysis of the ‘Normal turbulence model’ load case. Figure 7.12 illustrates the perturbations in the 12 m radius flapwise bending moment for the above machine, as a result of such high frequency wind speed fluctuations, with respect to a sharp rise above rated wind speed (12 m/s) and sharp falls below steady winds of 24 and 28 m/s. In the first case considered, the yaw angle is -20° and the azimuth 0° , as this configuration yields the largest positive bending moment at rated wind speed. In the second case, the yaw angle is -40° , which exceeds the maximum value predicted over the design life, and provides an upper bound on the largest negative moment at short-term mean wind speeds around the cut-out value.

It is apparent from Figure 7.12 that rapid wind speed increases above rated wind speed will produce a significant increase in bending moment, but rapid reductions in wind speed

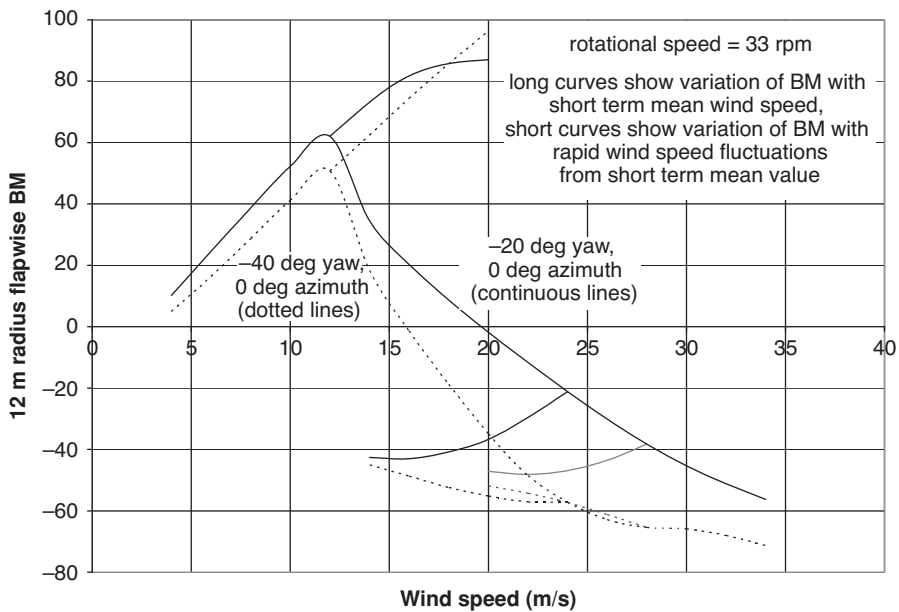


Figure 7.12 Effect of rapid wind speed fluctuations on 12 m radius flapwise BM for the example 40 m diameter pitch regulated machine

below, say, a 24 m/s steady value will not, as in this case the blade goes into negative stall. Over the machine lifetime, the maximum increase in wind speed above rated that does not produce a blade pitch response can be estimated using

$$\begin{aligned}
 u_{\max} &= \sigma_u \sqrt{\frac{\int_{\Omega/2}^{\infty} S_u(n)}{\int_0^{\infty} S_u(n)}} \left[\sqrt{2 \ln(\Omega T)} + \frac{\gamma}{\sqrt{2 \ln(\Omega T)}} \right] \\
 &= (\sigma_u)_{n > \Omega/2} \left[\sqrt{2 \ln(\Omega T)} + \frac{0.5772}{\sqrt{2 \ln(\Omega T)}} \right] \quad (7.8)
 \end{aligned}$$

where $(\sigma_u)_{n > \Omega/2}$ is the standard deviation of wind speed fluctuations above the pitch response cut-off frequency (assumed to be half the rotational frequency) and T is the total period of operation in the wind speed band centred on the rated speed at the yaw angle under consideration. For the IEC 61400-1 Edition 3 normal turbulence model the turbulence is given by

$$\sigma_u = I_{15} (0.75 \bar{U} + 5.6) \quad (7.9)$$

For a 12 m/s rated wind speed, with $I_{15} = 0.16$ for a Class A site, $\sigma_u = 2.34$ m/s and $(\sigma_u)_{n > \Omega/2} = 2.34 \cdot (0.4)^{0.5} = 1.48$ m/s. Taking a wind speed band of 2 m/s and yaw angles between -20° and -40° , the expression in square brackets (i.e. the peak factor) comes to 5.5, so that the lifetime extreme value of the wind speed increase without pitch response is about 8 m/s. If the wind speed fluctuations over the outer 8 m of blade are treated as perfectly correlated, this results in a maximum value of 12 m radius flapwise bending moment of 96 KNm (see Figure 7.12), which is over 50% greater than that occurring in a steady 12 m/s wind. Thus, the extreme flapwise bending moment during operation occurs at winds around rated rather than around the upper cut-out speed – a phenomenon which is a normal feature of pitch-regulated machines. Also, the extreme flapwise bending moment is less than for the similarly rated stall regulated machine considered above.

Fatigue loading

The importance of fatigue loading relative to extreme loading is very much a function of material properties. As the vast majority of blades are manufactured from composite materials with similar fatigue properties, discussion in this sub-section will be based on these.

As set out in Sections 7.1.6 and 7.1.7, composite materials are characterised by a very shallow $S-N$ curve – that is, the reciprocal index m in the relation $\sigma \propto N^{-1/m}$ for constant amplitude, reversed loading ($R = -1$) is typically 10 or more. As a result, fatigue damage can be dominated by the small number of high range stress cycles associated with unusual wind conditions, rather than by the routine medium range cycles.

The other significant property of composite materials is the increase in fatigue damage with mean stress level, which is usually accounted for by scaling up the stress amplitude entered in the $R = -1$ $S-N$ curve formulation by the factor $1/(1 - \bar{\sigma}/\sigma_d)$, where σ_d is the design strength in compression for a compression mean or in tension for a tension mean. This increases the relative importance of stress cycles with a high mean.

Behaviour of stall regulated machines in fatigue

For stall-regulated machines, the highest out-of-plane bending moment ranges and means normally occur at high wind speeds and yaw angles. This is illustrated in Figure 7.10, which shows the variation in this moment with wind speed and yaw angle at 60% radius for a 40 m diameter machine, based on the 3D data referred to above. Note that above rated wind speed, the bending moment plots level off, so that a given departure of the lateral wind component from the zero mean, sustained over half a revolution, results in a larger bending moment fluctuation than a change in the longitudinal component of twice this magnitude. For example, if the mean wind speed is 24 m/s, a lateral component of 6 m/s (corresponding to a yaw angle of 14°) causes a bending moment variation of 20 kNm when the blade rotates from 0° to 180° azimuth, compared to a variation of 17 kNm as a result of a ± 6 m/s fluctuation in longitudinal wind speed (which, in any case, could only occur after many blade rotations).

Similar comments apply to vertical wind speed fluctuations, but here there is a built-in initial tilt angle between the air flow and the shaft axis because of shaft angle tilt and updraft. Thus, bending moment plots derived from 3D wind simulations above rated are dominated by fluctuations at blade passing frequency which bloom and decay as the angle between the air flow and the shaft axis rises and falls. Superimposed on these are lower frequency fluctuations caused by changes in the longitudinal wind speed.

Clearly high wind/high yaw cycles will be a major source of fatigue damage, although the contribution of cycles at wind speeds below stall may also be important, because of the more rapid variation of moment with wind speed there, and the much increased number of cycles.

Thomsen (1998) has investigated for blade root out-of-plane bending on a 1.5 MW, 64 m diameter three-bladed machine, taking a constant turbulence intensity of 15% and an $S-N$ curve index of 12. The results, including allowance for mean stress, are plotted in Figure 7.13 (dotted), and indicate that the damage is concentrated at wind speeds of 20 m/s and

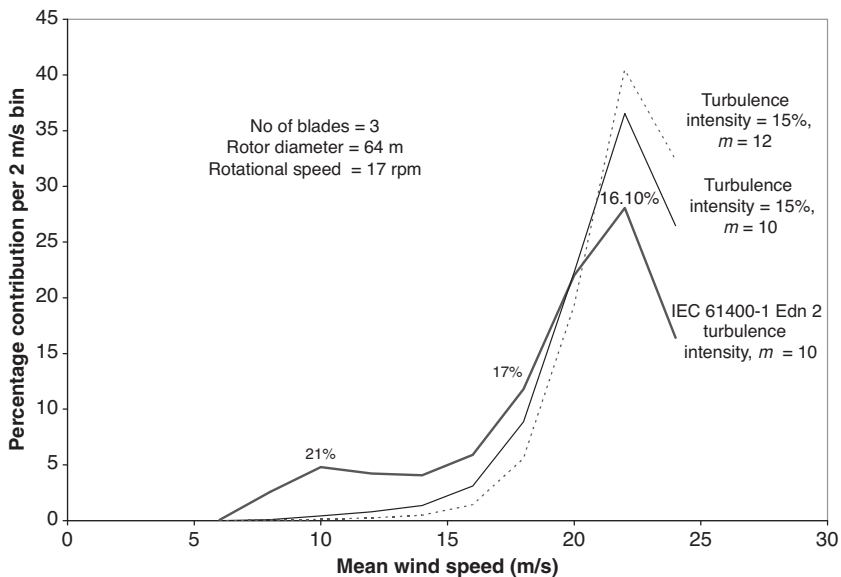


Figure 7.13 Relative contribution to life time fatigue damage for different wind speeds for a 1.5 MW stall regulated machine, including effect of mean load, after Thomsen (1998)

above. Figure 7.13 also shows the effect of adopting a steeper $S-N$ curve (with $m = 10$) and the IEC Class A turbulence distribution (with increasing intensities as mean wind speed decreases). In each case, the relative damage contribution at high wind speeds is reduced, but the switch to the IEC turbulence distribution causes the more significant change.

It should be noted that the relative contributions of different wind speeds to life time fatigue damage are also dependent on the shape of the bending moment/wind speed characteristics. Thus, for the machine with the bending moment/wind speed characteristics at 60% radius presented in Figure 7.10, the peak damage occurs at 10 m/s, if the IEC Class A turbulence intensity distribution is assumed – see Figure 7.15.

Behaviour of pitch regulated machines in fatigue

For pitch-regulated machines, the highest out-of-plane bending moment ranges occur at high wind speeds and yaw angles, but the largest mean values occur around rated wind speed. Moreover, blade pitching results in a rapid fall-off in bending moment with short-term mean wind speed just above rated. This behaviour is illustrated in Figure 7.11, which shows the variation in out-of-plane moment with short-term mean wind speed and yaw angle at 60% radius for a 40 m diameter machine. It transpires that the combination of the steep bending moment/short-term wind speed characteristic, high mean bending moment and large number of loading cycles just above rated wind speed results in more fatigue damage at this wind speed than at higher wind speeds, where the increasing bending moment fluctuations due to yaw offset are mitigated by reducing mean loads and numbers of cycles.

The nature of the bending moment fluctuations at a mean wind speed just above rated is shown in Figure 7.14, which is a time history obtained from a 3D wind speed simulation, for

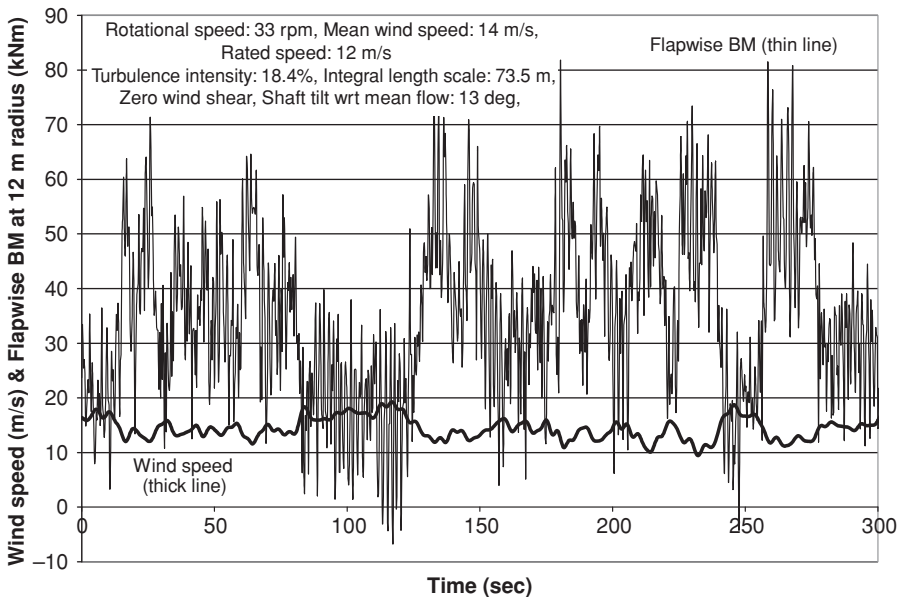


Figure 7.14 Time history of flapwise BM at 12 m radius and short-term mean wind speed for 40 m diameter pitch regulated m/c, based on 3D wind simulation with 14 m/s mean

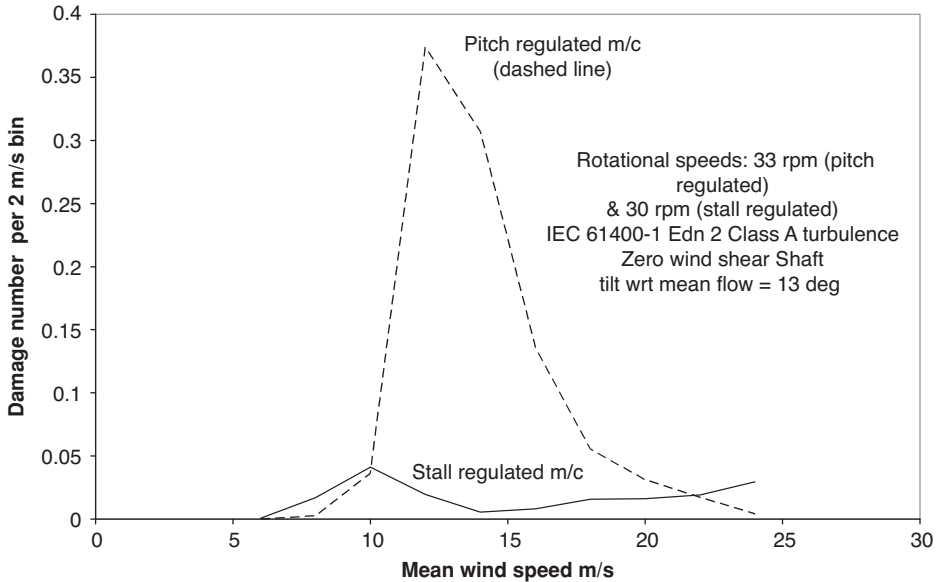


Figure 7.15 Variation of blade fatigue damage in flapwise bending at 12m radius with mean wind speed, for similar 40 m diameter pitch and stall regulated machines, ignoring dynamics

the machine with the bending moment/short-term mean wind speed characteristics presented in Figure 7.11 (with the response to high frequency wind speed fluctuations allowed for separately). As with the case of a stall regulated machine operating at high wind speed discussed above, there are considerable bending moment fluctuations at the rotational speed, but this time they are largely due to spatial variations in longitudinal wind speed across the disc (that is, ‘gust slicing’) rather than due to yaw or tilt offset. In addition, there are large low frequency bending moment fluctuations as a result of short-term mean wind speed changes – indeed, inspection of the bending moment and short-term mean wind speed plots reveals an inverse relationship between the two.

The fatigue damage in flapwise bending at 12 m radius arising from operation of the above machine at different mean wind speeds ignoring dynamics is plotted out in Figure 7.15, and is compared with that for a similar stall regulated machine having the same section modulus. The cross-section is designed to resist the extreme bending moment for the stall regulated machine of 130 KNm. In both cases the $S-N$ curve index is taken as 10, and the IEC 61400-1 Edition 2 Class A turbulence intensity assumed. It is apparent that the pitch regulated machine fatigue damage is concentrated around rated speed, and that the total damage is an order of magnitude greater than the total for the stall regulated machine. As the 12 m radius section modulus to resist the extreme flapwise moment is likely to be less for the pitch regulated machine, fatigue loading is likely to be more critical than indicated by the comparison in the figure.

Factors affecting fatigue criticality

The relative criticality of fatigue and extreme loading is determined by the material properties and safety factors adopted, as well as by the loadings themselves. As an aid to comparison, the

fatigue loading can be described in terms of the notional one cycle equivalent load, $\sigma_{eq(n=1)}$, which is defined as the stress range of the single reverse loading cycle that would cause the same total fatigue damage as the actual fatigue loading on the basis of the design $S-N$ curve. Then fatigue is critical if

$$\frac{\sigma_{eq(n=1)}}{2\sigma_{0d}} > \gamma_L \frac{\sigma_{ext}}{\sigma_{cd}} \quad (7.10)$$

where σ_{0d} is the stress *amplitude* given by the reverse loading fatigue design curve at $N = 1$, σ_{ext} is the stress resulting from the extreme loading case, γ_L is the load factor and σ_{cd} is the design compression stress (which is assumed not to be governed by buckling considerations). The condition may be rewritten in terms of characteristic stress values as follows:

$$\frac{\sigma_{eq(n=1)}}{\sigma_{ext}} > 2\gamma_L \frac{\gamma_{mu}}{\gamma_{mf}} \frac{\sigma_{0k}}{\sigma_{ck}} \quad (7.11)$$

or as $\sigma_{eq(n=1)}/\sigma_{ext} > 2.7((\gamma_{mu}/\gamma_{mf})(\sigma_{0k}/\sigma_{ck}))$ with γ_L set to 1.35.

As is implicit from the survey of GFRP and wood laminate properties in Sections 7.1.6 and 7.1.7, the value of σ_{0k}/σ_{ck} can vary between about 1.0 and 1.4. The IEC rules indicate that the partial materials safety factors for ultimate compression strength and fatigue strength should be taken as 1.3 and 1.2 respectively (assuming the fatigue strength is based on 95% survival probability at 95% confidence level), whereas, as noted in Section 7.1.6, the GL rules for GFRP lay down a value of ca 1.35 for the γ_{mu}/γ_{mf} ratio. Thus, in principal the parameter $2\gamma_L((\gamma_{mu}/\gamma_{mf})(\sigma_{0k}/\sigma_{ck}))$ governing fatigue criticality can take a wide range of values of between about 2.9 and 5.1.

In deriving the fatigue damage plots in Figure 7.15, a mid-range value of 4 has been adopted, resulting in total damages of 0.96 and 0.17 for the pitch and stall regulated machines respectively. However, if, the minimum value of 2.9 were adopted, corresponding to $\sigma_{0k} = \sigma_{ck}$ and $\gamma_{mu} = 1.08 \gamma_{mf}$, the fatigue damages would rise by a factor of about 25 for $m = 10$.

The other important material property governing the criticality of fatigue loading is, of course, the slope index of the log – log $S-N$ curve, m , which affects the value of the notional one cycle equivalent load, $\sigma_{eq(n=1)}$. With the high values applicable to wood laminates, fatigue is much less likely to govern.

Other sources of variability

There are a number of other sources of variability in fatigue damage calculations, apart from uncertainty about the material properties themselves, some of which are detailed below.

1. Two alternative stochastic turbulence models are in common use – those due to Von Karman and Kaimal. The Von Karman model is isotropic, whereas in the Kaimal model, which is more realistic in this respect, the standard deviations of lateral and vertical turbulences are 80% and 50% of the longitudinal turbulence respectively. In the case of stall regulated machines, where wind misalignment at high wind speeds is often the main source of fatigue damage, the choice of turbulence model could clearly have a decisive effect.

2. When the fatigue assessment is based on simulations of limited duration (typically 300–600 seconds), the damage is often dominated by a few extreme cycles, which are subject to significant statistical variation from one simulation to another. Accordingly several simulations at a given mean wind speed are necessary to obtain an accurate result. See ‘The statistical variation of wind turbine fatigue loads’ by Thomsen (1998).
3. In allowing for the reduction in fatigue strength due to mean stress (e.g. according to Equation 7.7), the mean stress can either be calculated over each stress range obtained by rain-flow cycle counting or over the length of the simulation.

Fatigue due to gravity loading

In-plane fatigue loads arise from gravity loading and fluctuations in the in-plane aerodynamic loadings, but gravity loadings dominate for machines large enough to be grid connected, rendering the loading calculation relatively straightforward.

In order to compare the approximately constant amplitude in-plane fatigue loading with the spectrum of out-of-plane fatigue loading, it is convenient to express both as equivalent loads at a specified number of cycles, n_{eq} . Often the 1 Hz equivalent load is used, in which case the number of cycles, n_{eq} , is equal to the number of seconds in the machine lifetime during which the machine operates. For machines of 40 m diameter and above, the fatigue equivalent load for in-plane bending at the blade root is typically greater than that for out-of-plane bending.

Over most of the blade length, the chord dimension is much larger than the blade thickness, so the section modulus for edgewise bending will generally exceed that for flapwise bending. However, for blades attached to the hub or pitch bearing by a circular ring of bolts, which is the normal arrangement, the blade structure adjacent to the root is a cylindrical shell, which will have the same section modulus about both axes if the wall thickness is uniform. As a consequence, the blade root is the first area that should be checked for in-plane fatigue loading.

The procedure can be illustrated for a 20 m tip radius blade in GFRP, designed for an extreme static root moment of 750 kNm at 0.5 m radius. Taking the gravity moment at the root as 124 kNm, and assuming 2.6×10^8 revolutions over the machine lifetime, the notional one cycle in-plane bending equivalent load range, $M_{eq(n=1)}$, becomes $248(2.6 \times 10^8)^{0.1} = 1720$ kNm and $\sigma_{eq(n=1)}/\sigma_{ext} = 1720/750 = 2.3$. This is less than the minimum value of 2.9 of the right hand expression in the inequality (Equation 7.11), so in-plane fatigue loading does not govern. In practice, the cylinder wall thickness would have to be increased significantly to prevent buckling, rendering fatigue less critical still at this diameter.

It may be concluded from the above example that in-plane fatigue loadings are not a significant consideration in the design of stall-regulated blades constructed in GFRP or wood laminates, until much larger diameters are reached. They cannot be ignored entirely, because of blade twist and because they add to the fatigue stress ranges due to flapwise bending at points on the cross-section away from the neutral axis for edgewise bending.

In-plane fatigue loadings are of more significance for pitch-regulated machines because gravity loadings will contribute increasingly to flapwise loadings as the pitch angle increases.

Tip deflection

Under extreme operating conditions, tip deflections of up to about 10% of blade radius can occur, so care is needed to avoid the risk of blade/tower collisions in the case of upwind

Table 7.3 Design strength to stiffness ratios for different wind turbine blade materials

Material	Ultimate compression strength, σ_{cu} (MPa)	Partial safety factor for material strength, γ_{mu}	Compression design strength, σ_{cd} (MPa)	Young's modulus, E (GPa)	Strength to stiffness ratio, $(\sigma_{cd}/E) \times 10^3$
Glass/polyester laminates with 50% fibre volume fraction and 80% UD	580	2.65	219 (ignoring buckling)	32.5	6.7
Carbon fibre/epoxy ply with 60% fibre volume fraction and UD lay-up	1100	2.65	415	142	2.9
Khaya/epoxy laminate	50	1.5	33	10	3.3
Birch/epoxy laminate	81	1.5	54	15	3.6
	Yield strength, σ_y	γ_{my}			
High Yield Steel (Grade Fe 510)	355	1.1	323	210	1.54
Weldable aluminium alloy AA6082	240	1.1	218	69	3.2

machines. GL specify that the quasi-static tip deflection under the extreme unfactored operational loading is not to exceed 50% of the clearance without blade deflection, which implies a safety factor of 2. IEC, on the other hand, require no blade/tower contact when the extreme dynamic tip deflection for each load case is multiplied by the combined partial safety factor for loads and blade elastic modulus.

It is instructive to compare the tip deflections for similar blades designed in different materials. If the skin thickness distributions are chosen so that the design compression strength of each material is fully mobilised under the extreme load case, then the tip deflection will be proportional to the design compression strength to Young's modulus ratio, σ_{cd}/E , of the blade material. These ratios are compared for different materials in Table 7.3.

It is clear from Table 7.3 that a GFRP blade will be about twice as flexible as blades in the other materials (apart from steel), provided that the spar is stocky enough for buckling not to govern the design – for example, as in Figure 7.4. In the case of thin walled cross sections, however, such as that in Figure 7.3, the GFRP compressive design stress has to be reduced significantly to guard against buckling, with the result that blade flexibility is reduced. For example, in the case of the 20 m tip radius blade, the compressive design stress is reduced to about 90 MPa, resulting in a tip deflection of about 2 m under extreme loading, and a strength to stiffness ratio less than that for wood/epoxy. In this connection, it is noteworthy that Hancock *et al.* (1997) record a 130% proof load test on a 31.2 m birch/epoxy blade which resulted in a tip deflection of 3.4 m.

7.1.9 Blade resonance

One of the most important objectives of blade design is the avoidance of resonant oscillations, which, in a mild form, exacerbate fatigue damage and in an extreme form can lead to rapid failure. The excitation of blade resonance can be minimised by maximising the damping and ensuring that the blade flapwise and edgewise natural frequencies are well separated from the exciting frequencies – that is, the rotational frequency and its harmonics, particularly the blade passing frequency – and from the frequencies of other vibration modes with which there is an identifiable risk of coupled oscillations.

Vibrations in stall

On stall regulated machines, the lift curve slope, $dC_l/d\alpha$, goes negative when a section of the blade goes into stall, resulting in local negative aerodynamic damping of blade motion in the lift direction. If the overall aerodynamic damping for a particular mode shape is negative, and exceeds the modal structural damping in magnitude, then divergent oscillations can develop from any initial disturbance, regardless of the relationship between the mode natural frequency and exciting frequencies. The first mode in each direction is most susceptible to such behaviour because the structural damping increases with frequency while the aerodynamic damping diminishes. If conditions favouring first mode oscillations are to be avoided, the factors affecting the aerodynamic damping of both edgewise and flapwise oscillations need to be understood, so these are explored below.

Consider a turbine operating in steady conditions in a perpendicular airflow. If a blade cross section at radius r experiences out-of-plane and in-plane perturbations with velocities $\dot{x}$ in the downwind direction and $\dot{y}$ in the direction opposite to that of blade rotation (assumed clockwise), then the relative velocity triangle is as in Figure 7.16(a).

The lift and drag forces per unit length on a blade element can be resolved into out-of-plane and in-plane forces F_X and F_Y (see Figure 7.16(b)), leading to Equations 5.18 and 5.19. Ignoring the small rotational induction factor, which is very small, these may be rewritten as:

$$F_Y = W [-C_l (U_\infty (1 - a) - \dot{x}) + C_d (\Omega r - \dot{y})] \frac{1}{2} \rho c \quad (7.12)$$

$$F_X = W [C_l (\Omega r - \dot{y}) + C_d (U_\infty (1 - a) - \dot{x})] \frac{1}{2} \rho c \quad (7.13)$$

Here U_∞ is the free stream wind speed and $U_\infty (1 - a)$ the reduced wind speed at the rotor plane as usual. The damping coefficients per unit length for vibrations in the in-plane and out-of-plane directions are then given by

$$\hat{c}_Y(r) = -\frac{\partial F_Y}{\partial \dot{y}} \quad (7.14a)$$

$$\hat{c}_X(r) = -\frac{\partial F_X}{\partial \dot{x}} \quad (7.14b)$$

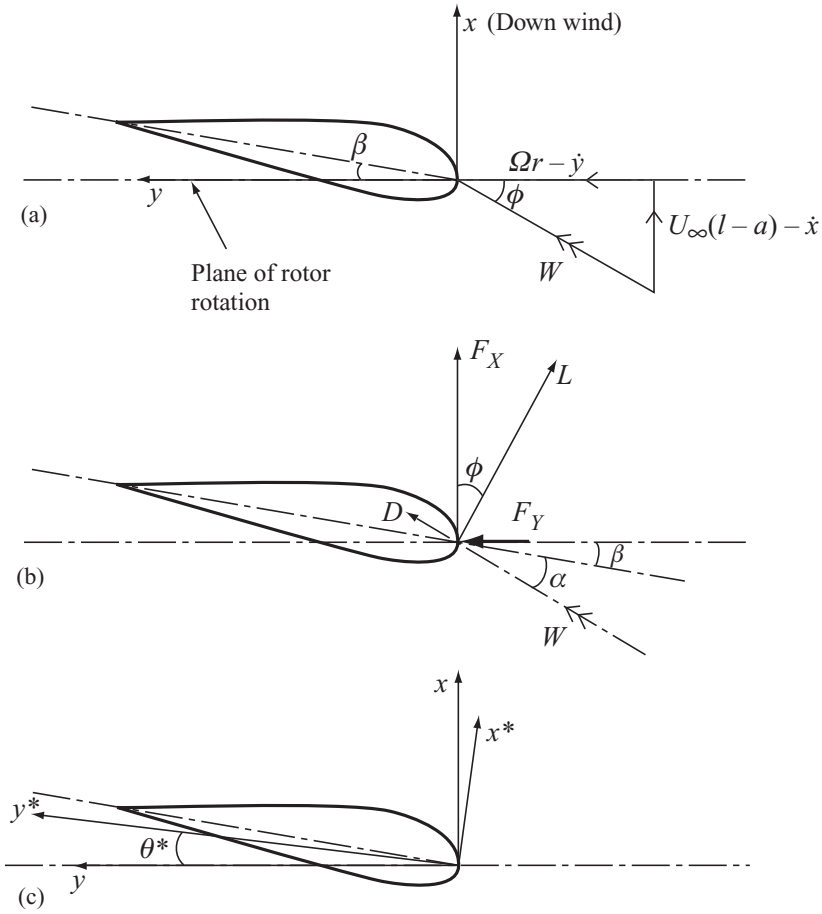


Figure 7.16 (a) Velocity diagram for vibrating blade (looking towards hub); (b) Out-of-plane and in-plane components of lift and drag forces; (c) Directions of vibrations x^* and y^*

Analagous 'cross' coefficients relating the in-plane force to the out-of-plane velocity and vice versa can also be defined as:

$$\hat{c}_{YX}(r) = -\frac{\partial F_Y}{\partial \dot{x}} \quad (7.15a)$$

$$\hat{c}_{XY}(r) = -\frac{\partial F_X}{\partial \dot{y}} \quad (7.15b)$$

Substituting V for $U_\infty(1-a)$ for brevity, the in-plane damping coefficient is derived as follows:

$$\hat{c}_Y(r) = -\frac{\partial F_Y}{\partial \dot{y}} = -\frac{1}{2}\rho c \left\{ \frac{\partial W}{\partial \dot{y}} [-C_l V + C_d \Omega r] + W \left[-\frac{\partial C_l}{\partial \dot{y}} V + \frac{\partial C_d}{\partial \dot{y}} \Omega r - C_d \right] \right\} \quad (7.16)$$

Noting that $\partial W / \partial \dot{y} = -\Omega r / W$ and $\partial C_l / \partial \dot{y} = (\partial C_l / \partial \alpha)(\partial \alpha / \partial \dot{y}) = (\partial C_l / \partial \alpha)(\partial \phi / \partial \dot{y}) = (\partial C_l / \partial \alpha)(V / W^2)$, this equation becomes:

$$\hat{c}_Y(r) = \frac{1}{2} \rho c \frac{\Omega r}{W} \left\{ -V C_l + \frac{V^2}{\Omega r} \frac{\partial C_l}{\partial \alpha} + \frac{2\Omega^2 r^2 + V^2}{\Omega r} C_d - V \frac{\partial C_d}{\partial \alpha} \right\} \quad (7.17)$$

The ‘cross’ coefficients and the out-of-plane damping coefficient are derived by a similar procedure:

$$\hat{c}_{YX}(r) = \frac{1}{2} \rho c \frac{\Omega r}{W} \left\{ -\frac{\Omega^2 r^2 + 2V^2}{\Omega r} C_l - V \frac{\partial C_l}{\partial \alpha} + V C_d + \Omega r \frac{\partial C_d}{\partial \alpha} \right\} \quad (7.18)$$

$$\hat{c}_{XY}(r) = \frac{1}{2} \rho c \frac{\Omega r}{W} \left\{ +\frac{2\Omega^2 r^2 + V^2}{\Omega r} C_l - V \frac{\partial C_l}{\partial \alpha} + V C_d - \frac{V^2}{\Omega r} \frac{\partial C_d}{\partial \alpha} \right\} \quad (7.19)$$

$$\hat{c}_X(r) = \frac{1}{2} \rho c \frac{\Omega r}{W} \left\{ +V C_l + \Omega r \frac{\partial C_l}{\partial \alpha} + \frac{\Omega^2 r^2 + 2V^2}{\Omega r} C_d + V \frac{\partial C_d}{\partial \alpha} \right\} \quad (7.20)$$

It is apparent from inspection of the expressions for the two damping coefficients, $\hat{c}_Y$ and $\hat{c}_X$, that the choice of an aerofoil with a gentler stall – that is, with a smaller lift curve slope after stall onset – will increase the damping coefficient in both cases. Note that the modal damping coefficient is dominated by the damping per unit length over the outboard part of the blade, so it is important to select an aerofoil with a gentle stall in this area only.

The choice of aerofoil also affects performance, so there is merit in expressing the damping coefficients in terms of the power output in order to investigate possible trade-offs between them. It transpires that the damping and ‘cross’ coefficients per unit length can be formulated quite simply in terms of the power output per unit length of blade, $P'(r, V) = \Omega r (-F_Y)$, and the blade thrust per unit length, F_X , as follows:

$$\hat{c}_Y = -\frac{2}{\Omega^2 r^2} P' + \frac{V}{\Omega^2 r^2} \frac{\partial P'}{\partial V} = \frac{1}{\Omega^2 r^2} \left(-2P' + V \frac{\partial P'}{\partial V} \right) \quad (7.21)$$

$$\hat{c}_{YX} = -\frac{\partial F_Y}{\partial \dot{x}} = \frac{\partial F_Y}{\partial V} = \frac{1}{\Omega r} \frac{\partial}{\partial V} (\Omega r F_Y) = -\frac{1}{\Omega r} \frac{\partial P'}{\partial V} \quad (7.22)$$

$$\hat{c}_{XY} = \frac{1}{\Omega r} \left(2F_X - V \frac{\partial F_X}{\partial V} \right) \quad (7.23)$$

$$\hat{c}_X = -\frac{\partial F_X}{\partial \dot{x}} = \frac{\partial F_X}{\partial V} \quad (7.24)$$

Equations 7.21 and 7.23 are derived from the equations $\Omega r \hat{c}_Y + V \hat{c}_{YX} = 2F_Y = -2P' / \Omega r$ and $\Omega r \hat{c}_{XY} + V \hat{c}_X = 2F_X$ which may be verified using Equations 7.17–7.20.

From Equation 7.21 it is clear that the damping coefficient in the in-plane direction, $\hat{c}_Y$, will always be negative unless $\partial P' / \partial V$ exceeds $2(P' / V)$, and that a negative power curve slope should be avoided if the size of the negative damping is to be kept small.

Effect of blade twist

In the discussion so far, damping of vibrations in the out-of-plane and in-plane directions only has been considered. In practice blade twist will result in the flapwise and edgewise vibrations taking place in directions rotated from the out-of-plane and in-plane directions in the same sense as the blade twist, but by a lesser amount (see Chapter 5, Section 5.8.1). If we define x^* and y^* axes in the directions of the flapwise and edgewise displacements, each making an angle of θ^* to the x and y axes respectively, as shown in Figure 7.16(c), then the edgewise damping coefficient per unit length is given by:

$$\hat{c}_Y^* = \hat{c}_Y \cos^2 \theta^* - (\hat{c}_{YX} + \hat{c}_{XY}) \sin \theta^* \cos \theta^* + \hat{c}_X \sin^2 \theta^* \quad (7.25)$$

Substitution of Equations 7.21–7.24 in Equation 7.25 yields:

$$\begin{aligned} \hat{c}_Y^* = \cos^2 \theta^* & \left[\frac{1}{\Omega^2 r^2} \left(-2P' + V \frac{\partial P'}{\partial V} \right) \right] + \cos \theta^* \sin \theta^* \left[\frac{1}{\Omega r} \left(-\frac{\partial P'}{\partial V} + 2F_X - V \frac{\partial F_X}{\partial V} \right) \right] \\ & + \sin^2 \theta^* \left(\frac{\partial F_X}{\partial V} \right) \end{aligned} \quad (7.26)$$

This expression also gives the flapwise damping coefficient per unit length if θ^* is replaced by $\theta^* + 90^\circ$ throughout.

The variation of the damping coefficient $\hat{c}_Y^*$ per unit length at 14 m radius with vibration direction, θ^* , at three different wind speeds is illustrated in Figure 7.17 for a specimen aerofoil section on a 20.5 m tip radius blade rotating at 29 rpm. The data is taken from Petersen

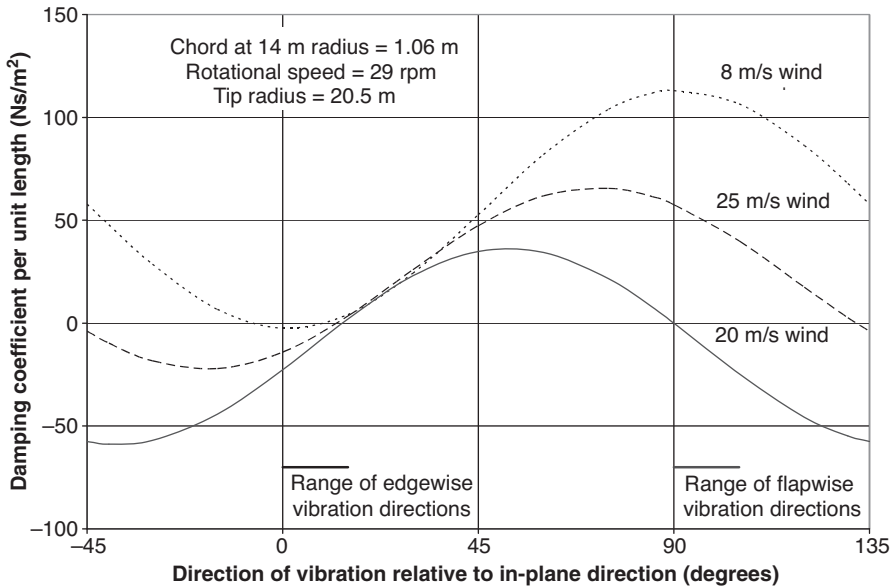


Figure 7.17 Variation in damping coefficient at 14m radius with vibration direction for example aerofoil

et al. (1998) and does not include allowance for the axial induction factor. It can be seen that negative damping is worst at 20 m/s, and that negative edgewise damping is ameliorated by increasing θ^* at the expense of increasing negative flapwise damping.

Although a plot of the local damping coefficient at ca 70% radius can provide a useful indication of trends, the best guide to the likelihood of divergent oscillations is provided by the modal damping coefficient for the mode under consideration. This is obtained by multiplying the right hand side of Equation 7.26 by the square of the local modal amplitude and integrating over the length of the blade.

If comparison of the first mode edgewise and flapwise modal damping coefficients shows there is a benefit to be gained from altering the direction of vibration, small changes can be made by redistributing material within the blade cross-section. Alternatively the blade pitch could be altered in conjunction with a compensatory change in aerofoil camber so that the aerodynamic properties for any given inflow angle are unchanged.

The prediction of edgewise vibrations in stall is examined in detail by Petersen *et al.* (1998), whose work provides the basis of the introductory survey given here. They concluded that the fundamental cause of edgewise blade oscillations that had been observed on some stall-regulated machines of 40 m diameter and over was negative aerodynamic damping, but found that the use of dynamic stall models improved the level of agreement with measurements.

Coupling of edgewise blade mode and rotor whirl modes

A further important finding was that, on one machine subject to stall-induced vibrations which was investigated in detail, there was coupling between the blade first edgewise mode and one of the second rotor whirl modes. The rotor whirl modes arise from the combination of simultaneous nodding and yawing oscillations of the rotor shaft, which occur at the same frequency during operation due to gyroscopic effects. As a result, the rotor hub traces out a circular or elliptical path, running either in the same direction as rotor rotation or in reverse, which explains the existence of two first and second modes.

The explanation for the coupling was as follows. When a pair of blades vibrate in the edgewise direction in anti-phase, they impart a sinusoidally varying in-plane force to the rotor hub even though their edgewise root bending moments cancel out. The direction of this oscillating force rotates with the rotor, so it has horizontal and vertical components of the form $\sin(\omega_1 t + \eta) \cdot \sin \Omega t$ and $\sin(\omega_1 t + \eta) \cdot \cos \Omega t$, where ω_1 is the frequency of the blade first edgewise mod, and Ω is the speed of rotor rotation. With respect to stationary axes the in-plane loads on the hub therefore act at two frequencies – namely $\omega_1 + \Omega$ and $\omega_1 - \Omega$. In the case of the machine investigated by Petersen *et al.*, the upper frequency of $2.9 + 0.5 = 3.4$ Hz coincided with the backward second rotor whirl mode, allowing interaction between this mode and the blade first edgewise mode.

Simulations were carried out on an aeroelastic model of the turbine at various wind speeds and satisfactory agreement obtained between simulated and measured behaviour. In particular, the simulation at 23.2 m/s predicted the build up of large blade root edgewise moment oscillations at the first mode frequency, as observed on the real machine at this wind speed. Significantly, when the latter simulation was repeated with the rotor shaft stiffness increased sufficiently to increase the backward second rotor whirl mode frequency to 3.6 Hz, the predicted blade root edgewise moment oscillations were negligible by comparison.

Mechanical damping

An alternative strategy for preventing damaging edgewise vibrations is the incorporation of a tuned mass damper inside the blade towards the tip. The performance of such a damper on a 22 m tip radius blade is reported by Anderson *et al.* (1998). It was found that the fitting of a damper tuned to the first mode edgewise frequency, and weighing only 0.4% of the total blade weight, effectively suppressed the edgewise vibrations which had previously been observed during high wind speed operation.

7.1.10 Design against buckling

The stress at which a slender plate element without imperfections buckles under compression loading is known as the critical buckling stress. The derivation of the critical buckling stresses for thin walled curved panels bounded by stiffeners, which typically form the blade load-bearing structure, is relatively straightforward when the panel material is isotropic and solutions are provided in Timoshenko and Gere (1961). These do not apply to composite materials such as the GFRP and wood laminates commonly used in blade manufacture, however, as these are anisotropic, but solutions can be derived for a symmetric laminate using the energy method, as outlined below.

Consider a long cylindrical panel of length L , radius r and thickness h , supported along two generators and subtending an angle ψ at the cylinder axis, which is axially loaded in compression. If it deflects to form n half-waves around the circumference between supports and m half-waves along its length, then its out-of-plane deflection can be written as:

$$w = C \sin \frac{n\pi\theta}{\psi} \sin \frac{m\pi x}{L} \quad (7.27)$$

where θ and x are the co-ordinates of the deflected point with respect to one of the long edges and one end respectively. In the absence of in-plane direct strains in the plate, this out-of-plane deflected profile will result in circumferential deflections

$$v_0 = \frac{C\psi}{n\pi} \cos \frac{n\pi\theta}{\psi} \sin \frac{m\pi x}{L} \quad (7.28)$$

These deflections will result in in-plane shear stresses, which reach a maximum at the corners of each rectangular buckled panel. In practice, additional in-plane deflections occur to moderate these shear stresses, as follows:

$$\begin{aligned} u &= A \sin \frac{n\pi\theta}{\psi} \cos \frac{m\pi x}{L} && \text{in the axial direction} \\ v &= B \cos \frac{n\pi\theta}{\psi} \sin \frac{m\pi x}{L} && \text{in the circumferential direction} \end{aligned} \quad (7.29)$$

The in-plane strain energy is calculated as

$$U_2 = \frac{1}{2}h \iint (\sigma_1 \varepsilon_1 + \sigma_2 \varepsilon_2 + \tau \gamma) r d\theta dx \quad (7.30)$$

with the suffices 1 and 2 denoting the axial and circumferential directions respectively, so that

$$\varepsilon_1 = \frac{\partial u}{\partial x}, \quad \varepsilon_2 = \frac{\partial v}{r \partial \theta}, \quad \gamma = \frac{\partial u}{r \partial \theta} + \frac{\partial (v_0 + v)}{\partial x} \quad (7.31)$$

Substituting $\sigma_1 = E_x(\varepsilon_1 + \nu_y \varepsilon_2)/(1 - \nu_x \nu_y)$, $\sigma_2 = E_y(\varepsilon_2 + \nu_x \varepsilon_1)/(1 - \nu_x \nu_y)$ and $\tau = G_{xy} \gamma$, where E_x , E_y and G_{xy} are the longitudinal, transverse and shear moduli of the laminate respectively (obtained by averaging the corresponding moduli of the individual plies) and ν_x and ν_y are the effective Poisson's ratios, the in-plane strain energy becomes:

$$U_2 = \frac{h}{2(1 - \nu_x \nu_y)} \iint [E_x \varepsilon_1^2 + E_y \varepsilon_2^2 + 2E_x \nu_y \varepsilon_1 \varepsilon_2 + (1 - \nu_x \nu_y) \gamma^2 G_{xy}] r d\theta dx \quad (7.32)$$

Substituting the expressions for ε_1 , ε_2 , and γ from Equation 7.31 and integrating over the width of the panel, ψr , and the length of one half wave, L/m , we obtain

$$U_2 = \frac{E_x h}{1 - \nu_x \nu_y} \psi r \frac{L}{m} \left(\frac{m\pi}{L} \right)^2 \frac{C^2}{8} \times \left[\alpha^2 + \beta^2 \frac{E_y}{E_x} \left(\frac{n}{\lambda} \right)^2 + 2\nu_y \alpha \beta \left(\frac{n}{\lambda} \right) + (1 - \nu_x \nu_y) \frac{G_{xy}}{E_1} \left\{ \alpha \left(\frac{n}{\lambda} \right) + \beta + \frac{\psi}{n\pi} \right\}^2 \right] \quad (7.33)$$

where $\lambda = m\psi r/L$ and the ratios $\alpha = A/C$ and $\beta = B/C$ are yet to be determined.

The expression for the strain energy of curvature is derived as follows. Replacing the angular coordinate θ by the linear coordinate y ($= r\theta$), the bending energy absorbed in an area $dx \cdot dy$ is:

$$dU_b = -\frac{1}{2} \left(M_x \frac{\partial^2 w}{\partial x^2} + M_y \frac{\partial^2 w}{\partial y^2} \right) dx \cdot dy$$

where $M_x = -D_x(\partial^2 w/\partial x^2) - D_{xy}(\partial^2 w/\partial y^2)$ and $M_y = -D_y(\partial^2 w/\partial y^2) - D_{xy}(\partial^2 w/\partial x^2)$ for a specially orthotropic laminate – that is, one in which the reinforcement in each layer is either oriented at 0° or 90° , or is bi-directional with the same amount of fibres at $+\theta^\circ$ and $-\theta^\circ$. D_x and D_y are the flexural rigidities of the laminate when flat, for bending about the y axis and x axis respectively, and D_{xy} is the ‘cross flexural rigidity’ – that is, the moment per unit width about one axis generated by unit curvature about the other. Hence,

$$dU_b = \frac{1}{2} \left(D_x \left(\frac{\partial^2 w}{\partial x^2} \right)^2 + 2D_{xy} \frac{\partial^2 w}{\partial x^2} \frac{\partial^2 w}{\partial y^2} + D_y \left(\frac{\partial^2 w}{\partial y^2} \right)^2 \right) dx dy \quad (7.34)$$

The twisting energy absorbed in an area $dx dy$ is:

$$dU_t = \frac{1}{2} (M_{xy} + M_{yx}) \frac{\partial^2 w}{\partial x \partial y} dx dy$$

where

$$M_{xy} = 2 \left[\int_{-h/2}^{h/2} G_{xy}(z) \cdot z^2 dz \right] \frac{\partial^2 w}{\partial x \partial y}$$

in which z is the distance measured from the mid-plane of the laminate, $G_{xy}(z)$ is the in-plane shear modulus at that distance and h is the laminate thickness. Denoting the torsional rigidity, $\left[\int_{-h/2}^{h/2} G_{xy}(z) \cdot z^2 dz \right]$, by D_T , then

$$dU_t = \frac{1}{2} \cdot 4D_T \left(\frac{\partial^2 w}{\partial x \partial y} \right)^2 dx dy \quad (7.35)$$

The total strain energy of curvature over the width of the panel and the length of one half wave is found by substituting the out-of-plane deflection given by Equation 7.27 in Equations 7.34 and 7.35 and integrating over this area, which gives:

$$U_1 = U_b + U_t = \frac{C^2 \psi r L}{8} D_x \left(\frac{m\pi}{L} \right)^4 \left[1 + \left(\frac{n}{\lambda} \right)^4 \frac{D_y}{D_x} + \left(\frac{n}{\lambda} \right)^2 \left\{ 2 \frac{D_{xy}}{D_x} + 4 \frac{D_T}{D_x} \right\} \right] \quad (7.36)$$

The energy absorbed by the panel during buckling as a result of in-plane strains and out-of-plane curvature is equal to the work done by the critical axial load as the panel shortens. The shortening of the panel over one half wave length is given by

$$\int_0^{L/m} \frac{1}{2} \left(\frac{\partial w}{\partial x} \right)^2 dx = \frac{\pi^2}{4} C^2 \frac{m}{L} \sin^2 \frac{n\pi\theta}{\psi} \quad (7.37)$$

so the work done by the axial force of N_x per unit width over the panel width is

$$T_1 = \frac{\pi^2}{8} C^2 \frac{m}{L} \psi r N_x \quad (7.38)$$

The equality $T_1 = U_1 + U_2$ yields the critical value of the axial force as follows:

$$\begin{aligned} (N_x)_{cr} = D_x \left(\frac{m\pi}{L} \right)^2 & \left[1 + \left(\frac{n}{\lambda} \right)^4 \frac{D_y}{D_x} + \left(\frac{n}{\lambda} \right)^2 \left\{ 2 \frac{D_{xy}}{D_x} + 4 \frac{D_T}{D_x} \right\} \right] + \frac{E_x h}{1 - \nu_x \nu_y} \\ & \times \left[\alpha^2 + \beta^2 \frac{E_y}{E_x} \left(\frac{n}{\lambda} \right)^2 + 2\alpha\beta \nu_y \frac{n}{\lambda} + (1 - \nu_x \nu_y) \frac{G_{xy}}{E_x} \left\{ \alpha \frac{n}{\lambda} + \beta + \frac{\psi}{n\pi} \right\}^2 \right] \end{aligned} \quad (7.39)$$

Noting that $m\pi/L = (m\psi r/nL)(n\pi/\psi r) = (\lambda/n)(n\pi/\psi r)$, this equation becomes

$$\begin{aligned} (\sigma_x)_{cr} = \frac{D_x}{h} \left(\frac{\lambda n\pi}{n\psi r} \right)^2 & \left[1 + \left(\frac{n}{\lambda} \right)^4 \frac{D_y}{D_x} + \left(\frac{n}{\lambda} \right)^2 \left\{ 2 \frac{D_{xy}}{D_x} + 4 \frac{D_T}{D_x} \right\} \right] + \frac{E_x}{1 - \nu_x \nu_y} \\ & \times \left[\alpha^2 + \beta^2 \frac{E_y}{E_x} \left(\frac{n}{\lambda} \right)^2 + 2\alpha\beta \nu_y \frac{n}{\lambda} + (1 - \nu_x \nu_y) \frac{G_{xy}}{E_x} \left\{ \alpha \frac{n}{\lambda} + \beta + \frac{\psi}{n\pi} \right\}^2 \right] \end{aligned} \quad (7.40)$$

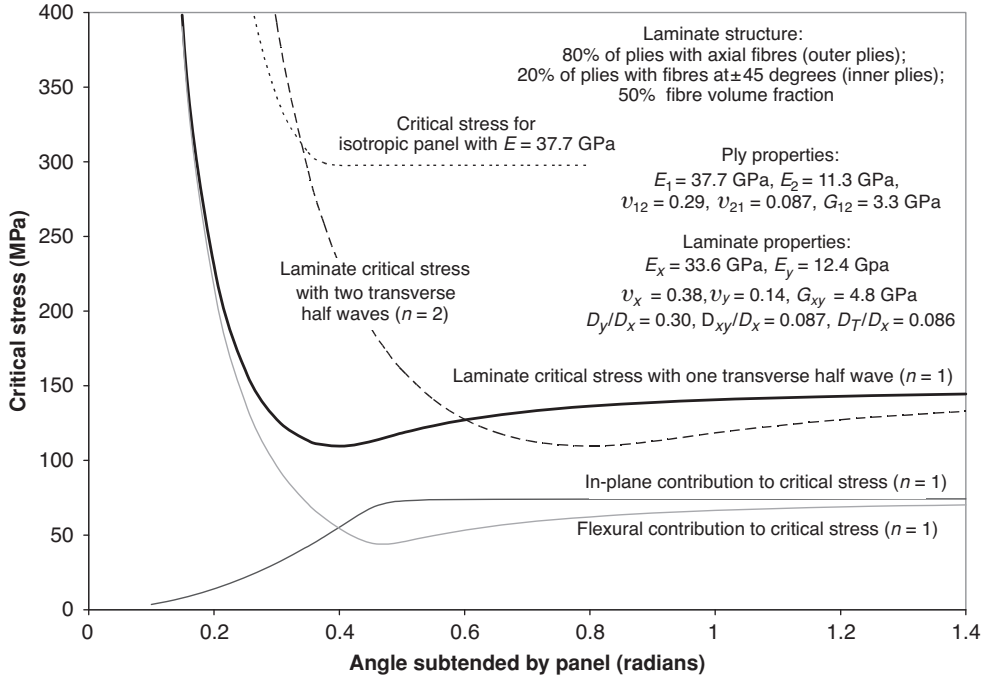


Figure 7.18 Variation of axial critical buckling stress with panel width for specimen curved anisotropic panel with radius 1150 mm and thickness 15 mm

The right hand side of Equation 7.40 contains four unknowns, the number of transverse half waves, n , the ratio of longitudinal to transverse half wave length, n/λ , and the factors α and β . Assuming that there is only one transverse half wave, as is normally the case, the expression is minimised with respect to α , and β for each value of n/λ , and then with respect to n/λ to obtain the critical stress.

The results of this exercise are illustrated for a particular curved laminate panel in Figure 7.18. The radius of curvature, r , of 1150 mm and thickness, h , of 15 mm are chosen to be representative of the values likely to obtain at 70% radius on a blade with 20 m tip radius. The laminate has 80% of its plies with fibres oriented axially and 20% with fibres at $\pm 45^\circ$ to resist shear loads. In each case, the fibre volume fraction is 50%. The $\pm 45^\circ$ plies are concentrated about the laminate mid-plane so that they do not detract significantly from the longitudinal flexural rigidity. Thus, if the longitudinal modulus of the UD plies is denoted by E_1 , and the Poisson's ratios by ν_{12} and ν_{21} , the longitudinal flexural rigidity is given approximately by:

$$D_x = \frac{E_1 h^3}{12(1 - \nu_{12}\nu_{21})} \quad (7.41)$$

The other ply and laminate properties required for evaluation of the critical stress are detailed on the figure. Note that in the derivation of the laminate in-plane stiffness properties it is necessary to transform the in-plane stiffness properties of the $\pm 45^\circ$ plies obtained initially

in relation to the ply axes (which are parallel to the fibre directions) to the set of properties in relation to the global x and y axes of the laminate as a whole. See, for example, Barbero (1998) for the requisite formulae.

The heavy curve in Figure 7.18 shows the variation in axial critical stress with panel width (in terms of subtended angle) when the buckled shape has only a single half wave in the transverse direction and the fine lines below show the separate in-plane and flexural contributions. The minimum stress of 110 MPa occurs when the angle subtended by the panel is about 20° , but there is only a gradual increase in critical stress as the angle increases above this. When the subtended angle exceeds about 35° , buckling with two half waves in the transverse direction takes over as the critical mode – see dashed line.

Also shown for comparison is the critical buckling stress variation for an isotropic plate with Young's modulus equal to the longitudinal modulus of the UD plies of the laminate – see dotted line. The minimum critical stress in this case is 298 MPa – about 2.7 times as big.

7.1.11 Blade root fixings

The fixing of the blade root to the hub is one of the most critical areas of blade design, because the order of magnitude difference between the relative stiffnesses of the steel hub and the blade material – usually GFRP or wood – militates against a smooth load transfer. The connection is usually made by steel bolts, which can either be embedded in the blade material in the axial direction or aligned radially to pass through the blade skin, but in either case stress concentrations are inevitable.

Figure 7.19 illustrates four different types of blade root fixings in section. The blade structure is usually a cylindrical shell at the root, in which case the stud or bolt fixings are arranged in a circle. Figure 7.19(a) shows the carrot connector, which is the standard fixing for laminated wood blades. The connector consists of a tapered portion carbon-epoxy grouted into a stepped hole drilled into the end of the blade, together with a projecting threaded stud for attachment to the hub or pitch bearing. Connectors are either machined from high strength steel or cast from spheroidal graphite iron (SGI). They are normally preloaded to reduce fatigue loading. A similar connector, in which the embedded portion is cylindrical rather than tapered, is in common use on GFRP blades.

Figures 7.19(b)–(d) show three further fixing arrangements used on GFRP blades. The T-bolt connector, shown in Figure 7.19(b), consists of a steel stud inserted into a longitudinal hole in the blade skin, which engages with a cylindrical nut held in a transverse hole. The stud is preloaded to reduce fatigue loading.

The 'pin-hole flange' arrangement in Figure 7.19(c) uses the same method of load transfer between the GRP and the steel – that is, bearing on a transverse rod – but the interface does not lend itself to preloading. Moreover the bolts attaching the flange to the hub are eccentric to the blade skin, so the flange has to resist the resultant bending moment as well.

In the trumpet flange detailed in Figure 7.19(d), the blade root is splayed out in the form of a trumpet mouth and clamped between inner and outer flanges by the ring of bolts which attach the flange to the hub. These bolts also pass through the GFRP skin to provide positive anchorage. Again the flange has to resist bending moments arising from the eccentricity of the fixing bolts to the blade skin where it emerges from the flange. The pin-hole and trumpet flange arrangements are rarely used for larger blades.

The stress distributions calculated for blade root fixings are subject to significant levels of uncertainty, so it is normal to conduct both static and fatigue tests on them to verify

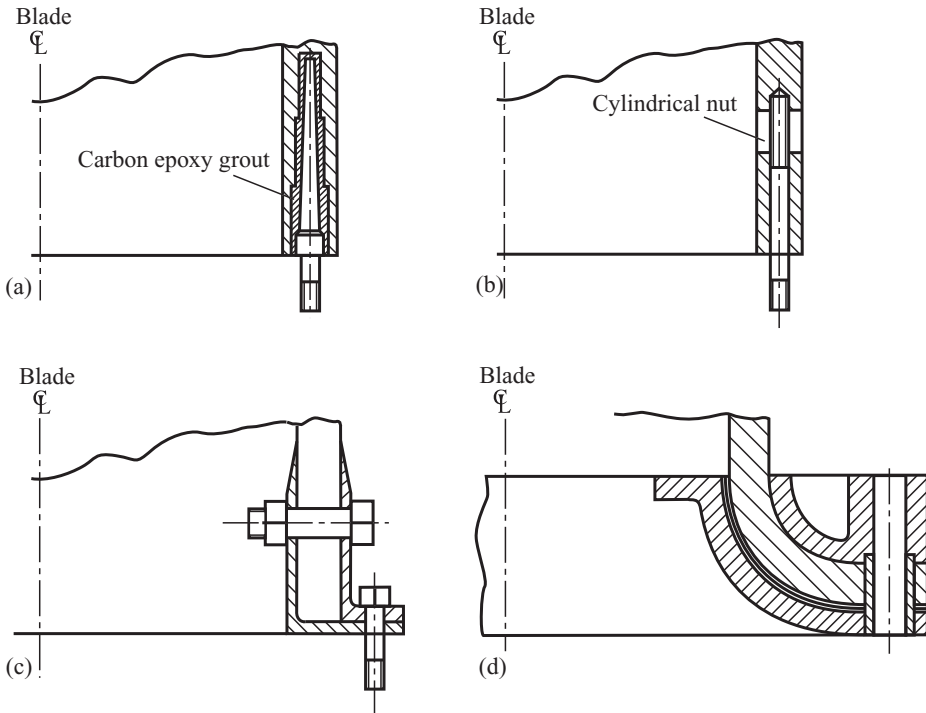


Figure 7.19 (a) Carrot connector; (b) T-bolt connector; (c) Pin-hole flange; (d) Trumpet flange

the suitability of the design. Static pull-out failures of carrot connectors occur as a result of shearing of the wood surrounding the grout, but fatigue failures can also occur in the connector itself or the grout. However, SGI studs subjected to $R = 0.1$ fatigue loading at over 60% of the UTS have survived for approximately 10^6 cycles.

Mayer (1996) records the results of fatigue tests on the other blade root fixings featured in Figure 7.19, but in no case did failure occur as a result of fatigue of the GFRP in the region of the root fixing. In the case of the T-bolt fixing arrangement, failure occurred in the studs rather than in the GFRP. The pin-hole flange specimens developed fatigue cracks in the GFRP in areas remote from the root fixings and the trumpet flange specimens developed cracks in the flanges themselves.

7.2 Pitch bearings

On pitch-regulated machines a bearing similar to a crane slewing ring is interposed between each blade and the hub to allow the blade to be rotated or ‘pitched’ about its axis. A typical arrangement is as shown in Figure 7.20, in which the inner and outer rings of the bearing are bolted to the blade and hub respectively.

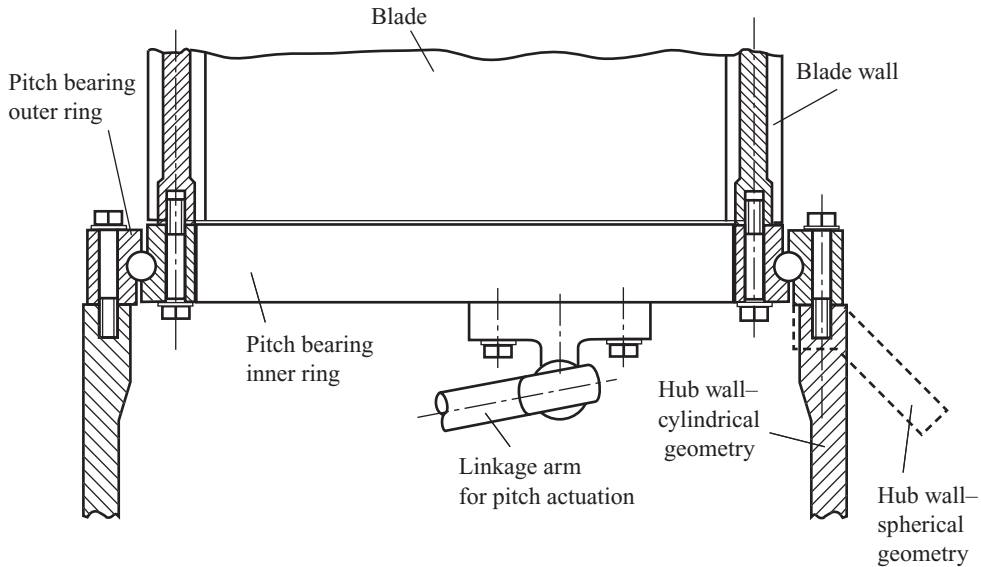


Figure 7.20 Typical pitch-bearing arrangement

The different types of bearings available can be classified according to the rolling elements used and their arrangement, in order of increasing moment capacity, as follows:

- (a) Single-row roller bearings, with alternate rollers inclined at $+45^\circ$ and -45° to the plane of the bearing.
- (b) Single-row ball bearings.
- (c) Double-row ball bearings.
- (d) Three-row roller bearings.

These are shown in cross-section in Figure 7.21. The single-row ball bearing slewing rings are normally designed to transmit axial loads in both directions and are, therefore, known as four point contact bearings. Low contact stresses are achieved by making the radii on each side of the grooves only slightly larger than that of the balls.

At low wind speeds, the cyclic in-plane bending moment at the blade root due to gravity is of similar magnitude to the out-of-plane moment due to blade thrust, so bearing loads will alternate in direction over portions of the bearing circumference. Accordingly it is desirable to avoid the risk of play by preloading the bearing. This can be achieved relatively easily on bearings in which one of the rings is split on a plane normal to the axis, such as types (c) and (d), but is more difficult when both rings are solid. In this case it is necessary to force the rolling elements into the races one by one during manufacture.

The bearing selected for a particular application needs to have sufficient moment capacity to both resist the extreme blade root bending moments and provide adequate fatigue life. Manufacturers catalogues typically specify both the extreme moment capacity and the steady moment loading that will give a life of, say, 30,000 bearing revolutions, so the wind turbine designer's chief task is to convert the anticipated pitch bearing duty into the equivalent constant

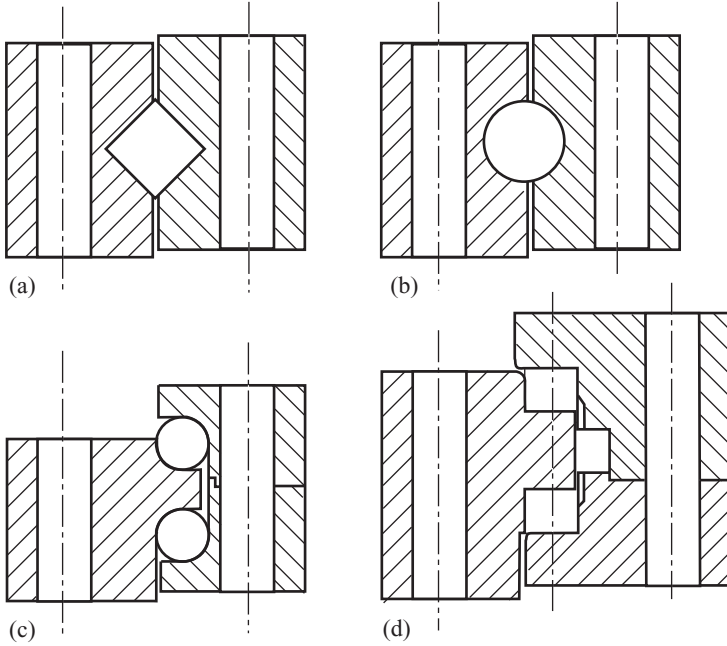


Figure 7.21 (a) Single-row crossed roller bearings; (b) Single-row ball bearings; (c) Double-row ball bearings; (d) Three-row roller bearings

loading at the appropriate number of revolutions. If the rolling elements are ball bearings, the bearing life is inversely proportional to the cube of the bearing loading, so the equivalent loading at N revolutions of the pitch bearing can be calculated according to the formula

$$M_{eqt} = \left[\frac{\sum_i n_i M_i^3}{N} \right]^{1/3} \quad (7.42)$$

where n_i is the total pitch bearing movement anticipated over the design life at moment loading M_i , expressed as a number of revolutions. In the case of roller bearings, the index of the S – N curve is $10/3$ instead of 3 , so the above formula should be modified accordingly. As the blade root out-of-plane moment drops as the wind speed increases above rated, the fatigue damage will be concentrated at wind speeds near rated.

The total pitch bearing movement over a period of operation at a particular wind speed is a function of the turbulence intensity and the pitch control algorithm, and is best predicted by means of a wind simulation. The mean blade pitching rate during operation above rated wind speed is found to be of the order of $1^\circ/\text{s}$, assuming the pitch system only responds to wind speed fluctuations at a frequency less than the speed of rotation.

The performance of slewing ring bearings such as those employed as pitch bearings is critically dependent on the extent of bearing distortion under load, so manufacturers normally specify a maximum axial deflection and tilt of the bolted contact surfaces. For example, the

limiting values given by Rothe-Erde for a single-row ball bearing slewing ring with a 1000 mm track diameter were 0.6 mm and 0.17° respectively. Local tilting of the bearing rings could clearly be minimised if the blade wall, bearing track and hub wall were all positioned in the same plane. However, this would necessitate the provision of flanges, so the simpler arrangement shown in Figure 7.20, in which the fixing bolts are inserted centrally into the blade and hub walls, is generally preferred. The designer must then ensure that the blade and hub structures are of sufficient stiffness to limit the bearing distortion due to the eccentric loading to acceptable values.

It is standard practice to preload the bearing fixing bolts in order to minimise bolt fatigue loading. Grade 10.9 bolts are commonly used so that the preload can be maximised.

7.3 Rotor hub

The relatively complex three dimensional geometry of rotor hubs favours the use of casting in their manufacture, with spheroidal graphite iron being the material generally chosen.

Two distinct shapes of hub for three-bladed machines can be identified: tri-cylindrical or spherical. The former consists of three cylindrical shells concentric with the blade axes, which flare into each other where they meet, while the latter consists simply of a spherical shell with cut-outs at the three blade mounting positions. Diagrams of both types are shown in Figure 7.22, while an actual spherical hub is illustrated in Figure 7.23. The structural action of the hub in resisting three loadings is discussed in the following paragraphs.

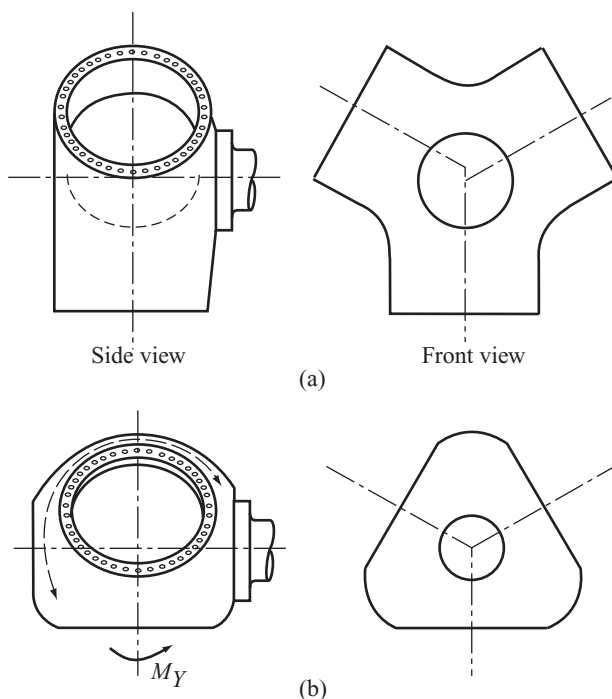


Figure 7.22 (a) Tri-cylindrical hub; (b) Spherical hub



Figure 7.23 Rotor Hub. View of spherical-shaped rotor hub for the 1.5 MW NEG Micon turbine awaiting installation. The hub and spinner are temporarily oriented with the rotor shaft axis vertical. The turbine is stall-regulated, so slotted blade fixing holes are provided to allow for fine adjustment of blade pitch to suit the site. Reproduced by permission of NEG-Micon

1. *Symmetric rotor thrust loading*: The blade root bending moments due to symmetric rotor thrust loading put the front of the hub in bi-axial tension near the rotor axis and the rear in bi-axial compression, while the thrust itself generates out-of-plane bending stresses in the hub shell adjacent to the low speed shaft flange connection. The load paths are easy to visualise in this case.
2. *Thrust loading on a single blade*: This generates out-of-plane bending stresses in the hub shell at the rear, and in-plane tensile stresses around a curved load path between the upwind side of the blade bearing and the portion of the low speed shaft flange connection remote from the blade (see dashed line in Figure 7.22b). The resultant lateral loads will result in out-of-plane bending.
3. *Blade gravity moments*: On the tri-cylindrical hub, equal and opposite blade gravity moments are communicated via the cylindrical shells to areas near the rotor axis at front and rear where they cancel each other out. It is less straightforward to visualise the corresponding load paths on the spherical hub, as out-of-plane bending is likely to be mobilised.

The complexity of the stress states arising from the latter two types of loading renders finite element analysis of rotor hubs more or less mandatory. At the most, six load cases need to be analysed, corresponding to the separate application of moments about the three axes and forces along the three axes at a single hub/blade interface. Then the distribution of hub stresses due to combinations of loadings on different blades can be obtained by superposition. Similarly the fluctuation of hub stresses over time can be derived by inputting the time histories of the blade loads obtained from a wind simulation.

The critical stresses for hub design are the in-plane stresses at the inner or outer surface, where they reach a maximum because of shell bending. For any one location on the hub, these are defined by three quantities at each surface – the in-plane direct stresses in two directions at right angles and the in-plane shear stress. In general, these stresses will not vary in-phase with each other over time, so the principal stress directions will change, complicating the fatigue assessment.

There is, as yet, no generally recognised procedure for calculating the fatigue damage accumulation due to multi-axial stress fluctuations, although the following methods have been used, despite their acknowledged imperfections. They all cater for one or more series of repeated stress cycles rather than the random stress fluctuations resulting from turbulent loading.

1. *Maximum shear method*: Here the fatigue evaluation is based on the maximum shear stress ranges, calculated from either the $(\sigma_1 - \sigma_2)/2$, $\sigma_1/2$ or $\sigma_2/2$ time histories. The effect of mean stress is allowed for using the Goodman relationship:

$$\frac{\tau_a}{S_{SN}} + \frac{\tau_m}{S_{Su}} = \frac{1}{\gamma} \quad (7.43)$$

where

τ_a is the alternating shear stress

τ_m is the mean shear stress

S_{SN} is the alternating shear stress for N loading cycles from the material S – N curve

S_{Su} is the ultimate shear strength

γ is the safety factor

Having used Equation 7.43 to determine S_{SN} , the permitted number of cycles for this loading range can be derived from the S – N curve, enabling the corresponding fatigue damage to be calculated.

2. *ASME Boiler and pressure vessel code method*: This is similar to the maximum shear method, but the shear stress ranges are based on notional principal stresses calculated from the *changes* in the values of σ_x , σ_y , σ_z , τ_{xy} , τ_{yz} and τ_{zx} from datum values occurring at one of the extremes of the stress cycle. Mean stress effects are not included.
3. *Distortion energy method*: In this method, the fatigue evaluation is based on the fluctuations of the effective or Von Mises stress. In the case of the hub shell, the stress perpendicular to the hub surface (and hence the third principal stress) is zero, so the effective stress is given by:

$$\sigma' = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + \sigma_1^2 + \sigma_2^2}{2}} \quad (7.44)$$

As the effective stress is based on the distortion energy, it is a scalar quantity, so it needs to be assigned a sign corresponding to that of the dominant principal stress. The effect of mean stress is allowed for in the same way as for the maximum shear method, except that the stresses in Equation 7.43 are now direct stresses instead of shear stresses.

S – N curves for spheroidal graphite iron are given in Hück (1983).

7.4 Gearbox

7.4.1 Introduction

The function of the gearbox is to step up the speed of rotor rotation to a value suitable for standard induction generators, which, in the case of fixed speed machines or two speed machines operating at the higher speed, is usually 1500 rpm plus the requisite slip. For machines rated between 300 kW and 5 MW, with upper rotational speeds between 48 and 12 rpm, overall gear ratios of between about 1:31 and 1:125 are therefore required. Normally these large step-ups are achieved by three separate stages with ratios of between 1:3 and 1:5 each.

The design of industrial fixed ratio gearboxes is a large subject in itself and well beyond the scope of the present work. However, it is important to recognise that the use of such gearboxes in wind turbines is a special application, because of the unusual environment and load characteristics, and the sections which follow focus on these aspects. Sections 7.4.2–7.4.6 consider variable loading, including drive train dynamics and the impact of emergency braking loads, and examine how gear fatigue design is adapted to take account of it. The relative benefits of parallel and epicyclic shaft arrangements are discussed in Section 7.4.7, while subsequent sections deal with noise reduction measures and lubrication and cooling. A useful reference is the American Gear Manufacturers Association Information Sheet entitled ‘Recommended practices for design and specification of gearboxes for wind turbine generator system’ published in 1996 in conjunction with the American Wind Energy Association, which covers the special requirements of wind turbine gearboxes in some detail. This has now been expanded into the standard ‘Design and specification of gearboxes for wind turbines’, ANSI/AGMA/AWEA 6006-A03.

7.4.2 Variable loading during operation

The torque level in a wind turbine gearbox will vary between zero and rated torque according to the wind speed, with excursions above rated on fixed speed pitch regulated machines due to slow pitch response. The short-term torque fluctuations will be subject to dynamic magnification to the extent that they excite drive train resonances (see Section 7.4.3 below). In addition there will be occasional much larger torques of short duration due to braking events, unless the brake is fitted to the low speed shaft. Figure 7.24 shows example load-duration curves (excluding dynamic effects and braking) for two 500 kW, two-bladed fixed speed machines – one stall and the other pitch regulated. The curve for the former is calculated by simply combining the power curve with the distribution of instantaneous wind speeds, which is obtained by superposing the turbulent variations about each mean wind speed on the Weibull distribution of hourly means. Excursions above rated power are not included.

In the case of a pitch regulated machine, the pitch control system is not normally designed to respond to wind speed fluctuations at blade passing frequency or above, as this would impose excessive loads on the control mechanism. Thus, there is no attenuation of the significant power fluctuations that occur at blade passing frequency due to turbulence, which are illustrated for the example two-bladed, 500 kW machine operating in a 20 m/s mean wind with 16.5% turbulence intensity in Figure 7.25.

The load duration curve for a fixed speed pitch regulated machine can be derived approximately from the distribution of instantaneous wind speeds below rated wind speed, and the

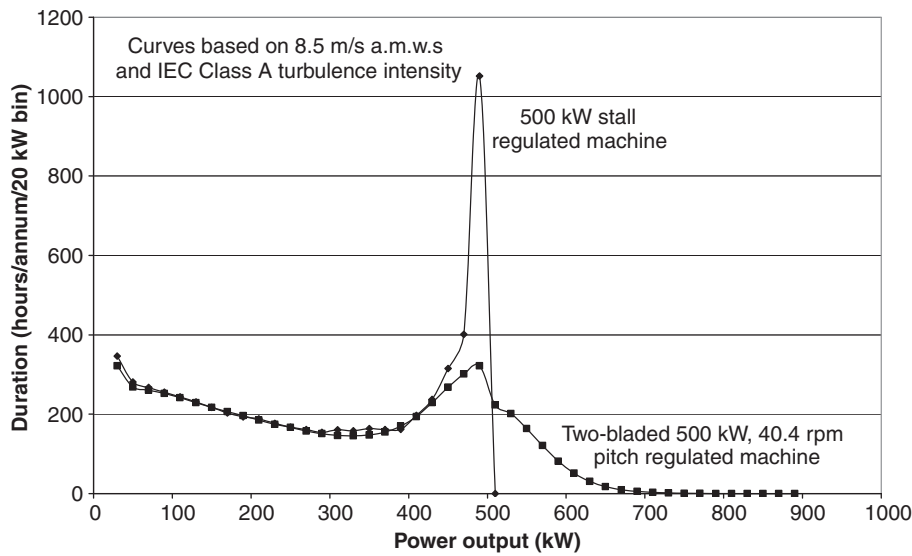


Figure 7.24 Load duration curves for 500 kW, 2 bladed pitch regulated and 500 kW, stall regulated fixed speed machines

distribution of short-term mean wind speeds (i.e. those to which the pitch system can respond) above. The former can be combined with the power curve to give the power distribution due to instantaneous winds below rated directly, while the winds above rated are assumed to produce Gaussian spreads of power outputs about the rated value, with the standard deviation depending on the short-term mean wind. The standard deviation of power fluctuations when

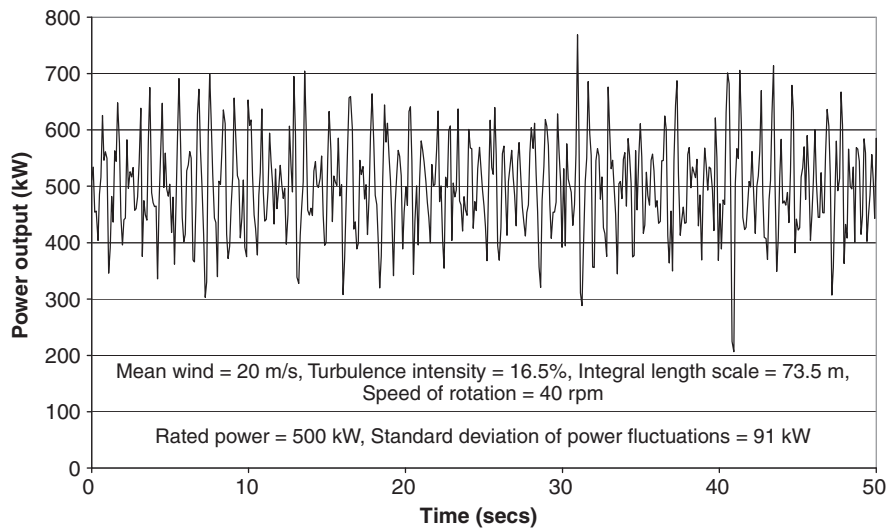


Figure 7.25 Simulated power output for two-bladed, 40 m diameter pitch regulated m/c operating in above rated wind speed

the pitch control system is operational can be related to that portion of the wind fluctuations above the pitch control system cut off frequency as follows:

$$\sigma_P^2 = \frac{1}{N^2} \sum_j \sum_k \left[\int_{\Omega}^{\infty} S_u^o(r_j, r_k, n) dn \right] \left(\frac{dp}{du} \right)_j \left(\frac{dp}{du} \right)_k \quad (7.45)$$

where $S_u^o(r_i, r_k, n)$ is the rotationally sampled cross spectrum of the wind speed fluctuations at a pair of points, j and k , on the rotor (see Section 5.7.5) and, $(dp/du)_j$ is the rate of change with wind speed of the power generated by the blade elements at r_j on all N blades if the pitch does not change. The summations are carried out over the whole rotor, and give $\sigma_P = 0.213 (dP/du)\sigma_u = 91$ kW for the example two-bladed machine operating at 40.4 rpm in a 20 m/s mean wind with 16.5% turbulence intensity. Here dP/du is the rate of change of turbine power with wind if the pitch does not change. The standard deviation of the power fluctuations for a three-bladed machine of similar size would be about one third less.

7.4.3 Drive train dynamics

All wind turbines experience aerodynamic torque fluctuations at blade passing frequency and multiples thereof because of the ‘gust slicing’ phenomenon, and these fluctuations will inevitably interact with the dynamics of the drive train, modifying the torques transmitted. In the case of fixed speed wind turbine with an induction generator, the resulting drive train torque fluctuations can be assessed by dynamic analysis of a drive train model consisting of the following elements connected in series:

- a body with rotational inertia and damping (representing the turbine rotor);
- a torsional spring (representing the gearbox);
- a body with rotational inertia (representing the generator rotor);
- a torsional damper (modelling the resistance produced by slip on the induction generator);
- a body of infinite rotational inertia rotating at constant speed (the mechanical equivalent of the electrical grid).

The inertias, spring stiffness and damping must all be referred to the same shaft.

7.4.4 Braking loads

Most turbines have the mechanical brake located on the high speed shaft, with the result that braking loads are transmitted through the gearbox. If, as is sometimes the case, the mechanical brake is one of the two independent braking systems required, then it must be capable of decelerating the rotor to a standstill from an overspeed – for example, after a grid loss. This typically requires a torque of about three times rated torque.

The mechanical brake is only required to act alone during emergency shutdowns, which are comparatively rare. During normal shutdowns the rotor is decelerated to a much lower speed by aerodynamic braking, so the duration of mechanical braking is much less,

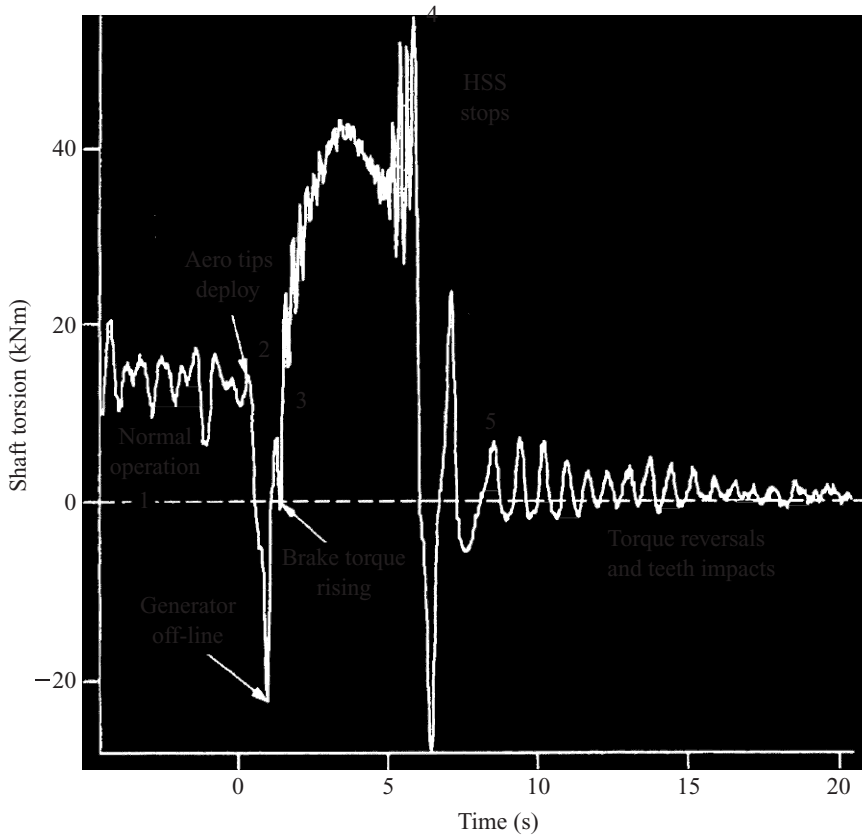


Figure 7.26 Low-speed shaft torque during braking at normal shut-down. Extracted from AGMA/AWEA 921–A97, Recommended practices for design and specification of gearboxes for wind-turbine generator systems, with permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314, USA

but the braking torque is the same, unless there is provision for two different braking torque levels.

Figure 7.26 is a typical record of low speed shaft torque during a normal shutdown, in which the mechanical brake is applied as soon as the generator has been taken off-line. It is apparent that the braking torque is far from constant, taking a couple of seconds to reach its first maximum and then falling off slightly before reaching a higher maximum just before the high speed shaft stops. Following this, there are significant torque oscillations due to the release of wind-up in the drive train. These result in torque reversals accompanied by tooth impacts and take some time to decay.

Although braking loads are infrequent and of short duration, their magnitude means that they can have a decisive effect on fatigue damage. The AGMA/AWEA document recommends that the time histories of braking and other transient events are simulated with the aid of a dynamic model of the drive train for input into both the gear extreme load design calculations and the fatigue load spectrum.

7.4.5 Effect of variable loading on fatigue design of gear teeth

Gear teeth must be designed in fatigue to achieve both acceptable contact stresses on the flanks and acceptable bending stresses at the roots. In non wind turbine applications, gearboxes typically operate at rated torque throughout their lives, so the gear strengths are traditionally modified by 'life factors' which are derived from the material $S-N$ curves on the basis of the predicted number of tooth load cycles for the gear in question. The British code for determining permissible gear contact stresses, BS 436: Part 3:1986 (since replaced by BS ISO 6336), recognises an endurance limit for both contact stress and bending stress, so that the life factors are unity when the number of tooth load cycles exceeds 10^9 and 3×10^6 respectively, but increase for lesser numbers of cycles.

The Hertzian compression stress between a pair of spur gear teeth in contact at the pitch point (i.e. at the point on the line joining the gear centres) is given by

$$\sigma_C = \sqrt{\frac{F_t}{bd_1} \frac{E}{\pi(1-\nu^2)} \frac{u+1}{u} \frac{1}{\sin \alpha \cos \alpha}} \quad (7.46)$$

where

F_t is the force between the gear teeth at right angles to the line joining the gear centres

b is the gear face width

d_1 is the pinion pitch diameter

u is the gear ratio (greater than unity)

α is the pressure angle – that is, the angle at which the force acts between the gears – usually 20° – 25°

Note that the contact stress increases only as the square root of the force between the teeth because the area in contact increases with the force as well.

The maximum bending stress at the tooth root is given by

$$\sigma_B = \frac{F_t h}{\frac{1}{6} b t^2} K_S \quad (7.47)$$

where

h is the maximum height of single tooth contact above the critical root section

t is the tooth thickness at the critical root section

K_S is a factor to allow for stress concentration at the root

For gearing operating at rated torque only, the designer needs to show that the resultant bending stress multiplied by an appropriate safety factor is less than the endurance limit multiplied by the life factor, Y_N , and a number of stress modifying factors, as follows:

$$\sigma_B \cdot \gamma \leq \sigma_{B \text{ lim}} \cdot Y_N \cdot Y_R \cdot Y_X \cdot \dots \quad (7.48)$$

A similar calculation is required in relation to the contact stress.

Given the predicted turbine load spectrum (Section 7.4.2), which should include dynamic effects (see Section 7.4.3), it is then necessary to establish the required design torque at the endurance limit. Normally this is done by invoking Miner's rule and determining the infinite

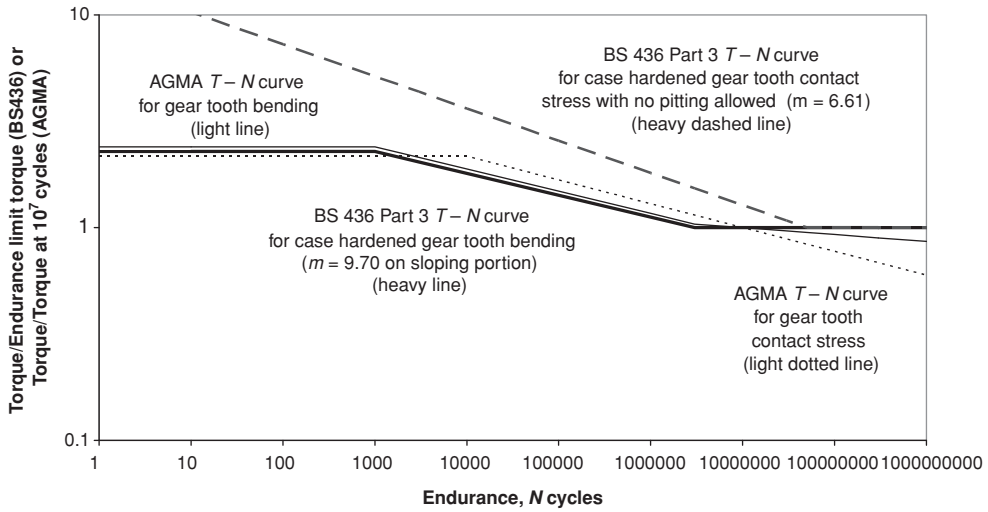


Figure 7.27 Specimen torque – endurance curves for gear tooth design

life torque for which the design torque spectrum yields unity fatigue damage in conjunction with the prescribed $S-N$ curve. Y_N in Equation 7.48 can then be set to unity, as the life factor has been accounted for in the derivation of the required infinite life torque.

Figure 7.27 shows specimen torque – endurance curves laid down by BS 436 for case hardened gears for tooth bending and tooth contact stress (with no pitting allowed) plotted in terms of the torque at the endurance limit. Hence in each case the design infinite life torque, T_∞ , is calculated according to:

$$T_\infty = \left[\sum_i \left(\frac{N_i}{N_\infty} T_i^m \right) \right]^{1/m} \quad (7.49)$$

where N_i is the number of cycles at torque level T_i , and torques less than T_∞ are omitted from the summation. The number of cycles at the lower knee of the torque – endurance curve, N_∞ , is always 3×10^6 cycles for tooth bending but is generally higher for contact stress, varying according to the material. Note that the slope index, m , of the torque – endurance curve for contact stress is half that of the contact stress – endurance curve because contact stress only increases as the square root of torque (Equation 7.46).

Leaving braking loads out of consideration to begin with, the design infinite life torque will be equal to the rated torque if there are no power fluctuations above rated, because the number of gear tooth loading cycles at rated torque will be well above N_∞ . For example, in the case of the 500 kW stall regulated machine featured in Figure 7.24, the teeth on the critical pinion driven by the 30 rpm low speed shaft will experience $3 \times 30 \times 60 \times 1050 \times 20 = 1.13 \times 10^8$ load cycles at rated torque over 20 years, assuming a first stage gear ratio of 3. On the other hand, for the 500 kW, two-bladed pitch regulated machine, the power fluctuations above rated detailed in the Figure 7.24 load-duration curve result in a design infinite life torque for the first stage pinion tooth bending stress of 1.36 times the rated torque, with most of

the damage coming from torques just above this value. (The first stage gear ratio is assumed to be three as before and the turbine rotational speed is taken as 40.4 rpm). The design infinite life torque for tooth contact stress is only $1.17 \times$ rated torque – significantly less than for bending, as expected from comparison of the BS 436 Part 3 torque – endurance curves in Figure 7.27.

Figure 7.27 also shows specimen torque – endurance curves derived from $S-N$ curves in the ANSI/AGMA standard 2001-C95 ‘Fundamental rating factors and calculation methods for involute spur and helical gear teeth’ plotted in terms of the torque at 10^7 cycles. The torque – endurance curve for tooth bending stress, which is based on a middle of the range Brinell Hardness value of 250 HB, closely parallels the selected BS 436 Part 3 curve, except that the curve continues with a very shallow slope beyond 3×10^6 cycles instead of displaying an endurance limit. The design torques at 10^7 cycles for tooth bending for the example 500 kW machines featured in Figure 7.24 are similar to the design infinite life torques obtained using the BS 436 Part 3 torque – endurance curves.

The ANSI/AGMA 2001-C95 torque – endurance curves for tooth contact stress are significantly more conservative than the selected BS 436 Part 3 curve. This is particularly so in the case of the ANSI/AGMA curve selected, which is the one recommended for wind turbine applications, in view of the elimination of the lower knee. The absence of the lower knee increases the design torque at 10^7 cycles for tooth contact to 1.4 times the rated torque for the stall regulated machine, but the figure for the pitch regulated machine is only about 10% higher.

From the above discussion, the general conclusion can be drawn that tooth bending fatigue usually governs the increased gearbox rating required to take care of load excursions above rated.

The effect of braking loads on the design infinite life torque according to BS 436 Part 3 can be illustrated with respect to the example machines discussed in Section 7.4.2. Although the mechanical brake must be capable of decelerating an overspeeding rotor unassisted, a shut-down under these conditions will be a very rare event. Accordingly the typical emergency shut-down considered for fatigue design purposes is deceleration from normal rotational speed under the action of mechanical and aerodynamic braking combined, with an assumed stopping time of three seconds. An emergency shut-down frequency of 20 per annum is assumed. Normal shut-downs are assumed to occur on average twice a day, with a stopping time of 1.5 seconds, because of the reduced rotational speed at which mechanical braking is initiated for parking. In each case the braking torque is assumed to remain constant at three times rated torque throughout the brake application for simplicity. Based on these assumptions, the percentage increases in design infinite life torque for gear tooth bending in fatigue, due to the inclusion of braking loads in the load spectrum, are shown in Table 7.4 for emergency braking alone on the one hand and normal plus emergency shut-downs on the other.

Also shown in Table 7.4 are the percentage increases in the AGMA design infinite life torque for gear tooth bending at 10^7 cycles due to the inclusion of braking loads. It is seen that the inclusion of emergency braking loads alone makes very little difference to design torques in the case of the pitch regulated machine, but is significant in the case of the stall regulated machine. The addition of braking loads at normal shut-downs incurs a much greater penalty in both cases because of the large number of stops involved, indicating that provision for brake application at reduced torque on these occasions would probably be worthwhile. Note that the larger percentage increases in design torques due to braking indicated by BS 436 Part 3 are a consequence of the assumption that there is an endurance limit.

Table 7.4 Illustrative increases in design torques for gear tooth bending due to inclusion of braking loads in fatigue load spectrum, according to BS 436 and AGMA rules

	500 kW stall regulated machine		500 kW two-bladed pitch regulated machine	
	Percentage increase in BS 436 design infinite life torque for tooth bending	Percentage increase in ANSI/AGMA 250 HB design torque at 10^7 cycles for tooth bending	Percentage increase in BS 436 design infinite life torque for tooth bending	Percentage increase in ANSI/AGMA 250 HB design torque at 10^7 cycles for tooth bending
Emergency braking at $3 \times$ FLT	30%	16%	4%	3%
Emergency plus normal braking, each at $3 \times$ FLT	65%	47%	25%	21%

7.4.6 Effect of variable loading on fatigue design of bearings and shafts

Bearing lives are approximately inversely proportional to the cube of the bearing loading. Applying Miner’s rule, the equivalent steady bearing loading over the gearbox design life can thus be calculated from the load duration spectrum according to the formula:

$$F_{eqt} = \left[\frac{\sum_i N_i F_i^3}{\sum_i N_i} \right]^{1/3} \tag{7.50}$$

where N_i is the number of revolutions at bearing load level F_i . Gravity often dominates the loading on the low speed shaft bearings, but on the other shafts the bearing loads result from drive torque only, so the bearing load duration spectrum can be scaled directly from the torque duration spectrum. Note that the S – N curve for bearings is much steeper than those for gear tooth design, so that occasional large braking loads will be of less significance.

The nature of the fatigue loading of intermediate shafts is essentially different from that of gear teeth, as the former is governed by the torque *fluctuations* as opposed to the absolute torque magnitude. Consequently the fatigue load spectrum for shaft design should be derived from rain-flow cycle counts on simulated torque time histories rather than on the load duration curve used for gear tooth design.

7.4.7 Gear arrangements

Parallel axis gears may be arranged in one of two ways in each gear stage. The simplest arrangement within a stage consists of two external gears meshing with each other and is commonly referred to as 'parallel shaft'. The alternative 'epicyclic' arrangement consists of a ring of planet gears mounted on a planet carrier and meshing with a sun gear on the inside and an annulus gear on the outside. The sun and planets are external gears and the annulus is an internal gear as its teeth are on the inside. Usually either the annulus or planet carrier are held fixed, but the gear ratio is larger if the annulus is fixed.

The epicyclic arrangement allows the load to be shared out between the planets, reducing the load at any one gear interface. Consequently the gears and gearbox can be made smaller and lighter, at the cost of increased complexity. The scope for material savings are greatest in the input stages of the gear train, so it is common to use the epicyclic arrangement for the first two stages and the parallel shaft arrangement for the output stage. A further advantage of epicyclic gearboxes is greater efficiency as a result of the reduced sliding that takes place between the annulus and planet teeth.

The derivation of the optimum gear ratio in a series of parallel shaft stages is fairly straightforward and is described below. Equation 7.47 for tooth bending stress can be modified as follows:

$$\sigma_B = \frac{F_t h}{\frac{1}{6} b t^2} K_S = F_t \frac{6(h/m)}{b m (t/m)^2} \cdot K_S = F_t \frac{6z_1(h/m)}{b d_1 (t/m)^2} \cdot K_S \quad (7.47a)$$

where m is the module, defined as d_1/z_1 for spur gears and z_1 is the number of pinion teeth. If the ratios h/m and t/m are treated as constants, then the bending stress is proportional to the number of teeth for a given size of gear. Hence the design of the gears is governed by contact stress because, in principle, the bending stress can always be reduced by reducing the number of pinion teeth. Thus, based on Equation 7.46, the permitted tangential force, F_t , is proportional to $b d_1 u / (u + 1)$ so that the permitted low speed shaft torque, $T_{LSS} = F_t d_2 / 2$ is given by

$$T_{LSS} \propto d_2 b d_1 u / (u + 1) = b d_2^2 / (u + 1) \quad (7.51)$$

Hence the volumes of the low speed shaft gear wheel and the meshing pinion can be expressed as $V_2 = k T_{LSS} (u + 1)$, where k is a constant, and, $V_1 = V_2 / u^2$ respectively. These can be used to derive an expression for the volume of gears in a drive train with an infinite number of stages each with the same ratio. It is found that the total gear volume is a minimum for a gear stage ratio of 2.9, but increases by only 10% when the ratio drops to 2.1 or rises to 4.3.

The gear teeth of parallel shaft gear stages are only loaded in one direction, so the permitted alternating bending stress amplitude in fatigue, σ_{alt} , is modified to account for the non-zero mean value in accordance with the Goodman relation:

$$\frac{\sigma_{alt}}{\sigma_{lim}} = 1 - \frac{\bar{\sigma}}{\sigma_{ult}} \quad (7.52)$$

where σ_{lim} is the permitted alternating bending stress amplitude with zero mean, $\bar{\sigma}$ is the mean bending stress and σ_{ult} is the ultimate tensile strength. Setting $\bar{\sigma} = \sigma_{alt}$ results in:

$$\sigma_{alt} = \frac{\sigma_{lim}\sigma_{ult}}{(\sigma_{ult} + \sigma_{lim})} \quad (7.53)$$

If the $\sigma_{lim}/\sigma_{ult}$ ratio is 0.2, then $\sigma_{alt} = 0.833\sigma_{lim}$ and the permitted peak bending stress at the endurance limit is $1.667\sigma_{lim}$. In epicyclic gearboxes, by contrast, the gear teeth on the planets' wheels are loaded in both directions, so the permitted peak bending stress at the endurance limit is only σ_{lim} . As the number of teeth on the smallest gear cannot be reduced indefinitely, this means that tooth bending is more likely to govern in the case of epicyclic gearing.

The minimum total gear volume for an infinite series of epicyclic gear stages with fixed annuli is obtained for a gear stage ratio of two, which implies that the radius of the sun gear is the same as that of the annulus gear and that there are an infinite number of planets! This is not realistic, and the annulus radius is in practice typically double the sun radius, giving a gear ratio of three. It is instructive to compare the volume of gears for an epicyclic and parallel gear stages with this ratio, assuming that tooth bending stress governs in each case.

For the parallel stage, it can be shown using Equation 7.47a that the volume of the pinion is:

$$\frac{\pi}{4}bd_1^2 = k_B F_t d_1 z_1 / \sigma_B = k_B \frac{F_t d_2 z_1}{2\sigma_{alt}u} = k_B \frac{2T_{LSS}z_1}{1.667\sigma_{lim}u} \quad (7.54)$$

where k_B is a constant. This gives a volume for gear wheel and pinion of $1.2k_B T_{LSS}z_1 (1 + 1/u^2)u / \sigma_{lim} = 4k_B T_{LSS}z_1 / \sigma_{lim}$ for $u = 3$.

For the epicyclic stage, the volume of the planet, which is assumed to have the same number of teeth as the pinion of the parallel stage – that is, the minimum permissible – is:

$$\frac{\pi}{4}bd_{p1}^2 = k_B F_t d_{p1} z_1 / \sigma_B = k_B F_t d_{p1} z_1 / \sigma_{lim} \quad (7.55)$$

If the low speed shaft drives the planet carrier and the N planets are spaced at 1.15 diameters, then the low speed shaft torque is:

$T_{LSS} = F_t N (r_A + r_S)$ where $N = \pi (r_A + r_S) / (1.15 (r_A - r_S))$, r_A is the annulus radius and r_S is the sun radius.

Hence, putting $a = r_A / r_S$, the volume of a planet is $k_B T_{LSS} 1.15(a - 1) / (\pi (a + 1)^2 r_S) \times (d_{p1} z_1 / \sigma_{lim})$ and the volume of the sun is $4 / (a - 1)^2$ times as big. The total volume of planets and sun becomes:

$$V = k_B T_{LSS} \frac{1}{a + 1} \frac{d_{p1} z_1}{r_S \sigma_{lim}} \left(1 + \frac{4}{(a - 1)^2 N} \right) \quad (7.56)$$

Substituting $a = 2$, we obtain:

$d_{p1} = r_S$ and $N = 3\pi / 1.15 = 8.195$ which is rounded down to 8, giving

$$V = k_B T_{LSS} \frac{z_1}{3\sigma_{lim}} \left(1 + \frac{4}{8} \right) = 0.5k_B T_{LSS} z_1 / \sigma_{lim}$$

Hence the volume of the sun and planets of the epicyclic stage is only one eighth of the volume of the gearwheel and pinion of the equivalent parallel stage, assuming the designs are governed by gear tooth bending stress. If contact stress were to govern, the relative volume of the epicyclic stage would be even less.

The dramatic materials savings obtainable with epicyclic gearboxes depend on equal sharing of loads between the planets. Although this is theoretically achievable through accuracy of manufacture, it is in practice desirable to introduce some flexibility in the planet mountings to take up any planet position errors – for example, by supporting the planets on slender pins cantilevered out from the planet carrier. Note that the fatigue design of such pins is, like the design of intermediate shafts, governed by torque *fluctuations* rather than by torque absolute magnitude.

7.4.8 Gearbox noise

The main source of gearbox noise arises from the meshing of individual teeth. Loaded teeth deflect slightly, so that if no tooth profile correction is made, unloaded teeth are misaligned when they come into contact, resulting in a series of impacts at the meshing frequency. It is, therefore, standard practice to adjust the tooth profile – usually by removing material from the tip area of both gears, referred to as ‘tip relief’ – to bring the unloaded teeth back into alignment at the rated gear loading. In the case of wind turbines, the gear loading is variable, so it is necessary to select the load level at which the tip relief provides the correct compensation. If the tip relief load level is too high, there will be excessive loss of tooth contact near the tips at low powers, while if it set too low the noise level at rated power will be too high. However, if gearbox noise is expected to be more intrusive at low wind speeds, when it is less likely to be masked by aerodynamic noise, then a low compensation load level should be selected.

Helical gears are usually quieter than spur gears (with teeth parallel to the gear axis) because the width of the tooth comes into mesh over a finite time interval rather than all at once. Moreover, the peak tooth deflections of helical gears are less than those of spur gears because there are always at least two teeth in contact rather than one, and because the varying bending moment across the tooth width means that the less heavily loaded portions of the tooth can provide restraint to the part that is most heavily loaded. As a result, the tooth misalignments due to insufficient/excessive tip relief at a particular load level will be reduced.

Epicyclic gears are normally quieter than parallel shaft gears because the reduced gear size results in lower pitch line velocities. However, this benefit is lost if spur gears are used rather than helical gears, in order to avoid problems with planet alignment. One way of maintaining the alignment of helical planet gears is to provide thrust collars on the sun and annulus.

As the annulus of an epicyclic gear stage is often fixed, it would be convenient to integrate it with the gearbox casing. However, this would enable annulus gear meshing noise to be radiated directly from the casing, so it is preferable to make the annulus a separate element, supported on resilient mountings. Similarly, resilient gearbox mountings should be used to attenuate the transmission of gearbox noise to the nacelle structure and tower.

The noise produced by gear tooth meshing can reach the environment outside the wind turbine by a variety of routes, as follows:

- through the shaft directly to the blades, which may radiate efficiently;
- through the resilient mounts of the gearbox to the support structure and thereby to the tower, which can radiate efficiently under some circumstances;

- through the resilient mounts of the gearbox to the support structure and thereby to the nacelle structure, which can also radiate;
- through the casing wall to the nacelle air and then via air intake and exhaust ducts;
- through the casing wall to the nacelle air and then via the nacelle structure.

All these paths are modally dense and it is virtually impossible to design out a selected frequency. If noise is a problem then the options are to reduce the source sound level, perhaps by improving the tip relief as described above, or to modify the major path to reduce transmission. Identification of the major path is not straightforward, but one way of doing so is to use Statistical Energy Analysis (SEA), which combines a theoretical model with extensive field measurements. The path may not be simple, as non-linearity in the system can make one path the predominant one at low wind speeds and another path critical at higher wind speeds. Treatment of a radiating path can involve damping treatment such as shear layer damping or even just sand or bitumen layers added to the tower wall, for instance. In some cases the treatment can have more than one effect. When blades are the major source of radiation and damping material is added inside the blades then this material can act as a stiffening material as well as a damping mechanism. Sometimes it is useful to add tuned absorbers to parts of the structure to damp out one particular frequency. An alternative use of such tuned absorbers is to design them to raise the impedance at the tuned frequency so that the offending vibration does not pass that point on the structure.

7.4.9 Integrated gearboxes

As noted in Section 6.11.1, Chapter 6, the cases of integrated gearboxes must be very robust, in order to transmit the rotor loads to the nacelle structure without experiencing deflections which would impair the proper functioning of the gears. In view of the complex shape of the casing, stress distributions due to each load vector usually have to be determined using Finite Element (FE) analysis – these can then be superposed in line with the different extreme load combinations. The fatigue analysis will require the superposition of stress histories resulting from simultaneous time histories of rotor thrust, yaw moment and tilt moment derived from simulations at different wind speeds.

7.4.10 Lubrication and cooling

The function of the lubrication system is to maintain an oil film on gear teeth and the rolling elements of bearings, in order to minimise surface pitting and wear (abrasion, adhesion and scuffing). Varying levels of the elastohydrodynamic lubrication provided by the oil film can be identified, depending on oil film thickness. These range from full hydrodynamic lubrication, which exists when the metal surfaces are separated by a relatively thick oil film, to boundary lubrication when the asperities of the metal surfaces may be separated by lubricant films only a few molecular dimensions in thickness. Scuffing, which is a severe form of adhesive wear involving localized welding and particle transfer from one gear to the other, can occur under boundary lubrication conditions, which are promoted by high loading and low pitch line velocity and oil viscosity.

Two alternative methods of lubrication are available – splash lubrication and pressure fed. In the former, the low speed gear dips into an oil bath and the oil thrown up against the

inside of the casing is channelled down to the bearings. In the latter, oil is circulated by a shaft driven pump, filtered and delivered under pressure to the gears and bearings. The advantage of splash lubrication is its simplicity and hence reliability, but pressure fed lubrication is usually preferred for the following reasons:

- oil can be positively directed to the locations where it is required by jets;
- wear particles are removed by filtration;
- the churning of oil in the bath, which can result in a net efficiency loss, is avoided;
- the oil circulation system enables heat to be removed much more effectively from the gearbox by passing the oil through a cooler mounted outside the nacelle;
- it allows for intermittent lubrication when the machine is shutdown if a standby electric pump is incorporated.

With a pressure fed system, it is normal practice to fit temperature and pressure switches downstream of the filter to trip the machine for excessive temperature or insufficient pressure.

Guidance on the selection of lubricant, which has to take into account the ambient temperatures at the site in question, is given in the AGMA/AWEA document. Sump heaters may be needed to enable oil to be circulated when the turbine starts up at low temperatures.

7.4.11 Gearbox efficiency

Gearbox efficiency can vary between about 95% and 98%, depending on the relative number of epicyclic and parallel shaft stages and on the type of lubrication.

7.5 Generator

7.5.1 Fixed-speed induction generators

The induction generators commonly used in fixed-speed wind turbines are very similar to conventional industrial induction motors. In principle the only differences between an induction machine operating as a generator and as a motor are the direction of power flow in the connecting wires, whether torque is applied to or taken from the shaft and if the rotor speed is slightly above or below synchronous. The size of the market for induction motors is very large and so, in many cases, an induction generator design will be based on the same stator and rotor laminations as a range of induction motors in order to take advantage of high manufacturing volumes. Some detailed design modifications, for example, changes in rotor bar material, may be made by the machine manufacturers to reflect the different operating regime of wind turbine generator, particularly the need for high efficiency at part load, but the principles of operation are those of conventional induction machines. The synchronous speed, which is determined by the number of magnetic poles, and network frequency will be 1500 rpm (4 pole), 1000 rpm (6 pole) or 750 rpm (8 pole) for connection to a 50 Hz network. For commercial and safety reasons it is common to use a voltage of less than 1000 V (usually 690 V in Europe or 575 V in the USA) even for large generators and in some large wind turbines the resulting high currents have led to the decision to locate the turbine transformer

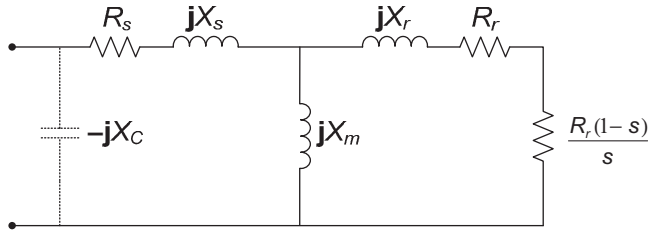


Figure 7.28 Steady state equivalent circuit of an induction machine with power factor correction capacitors. R_s : stator resistance, X_s : stator reactance, R_r : rotor resistance, X_r : rotor reactance, X_m : magnetising reactance, X_c : power factor correction reactance. j is the imaginary operator

in the nacelle or at the top of the tower. The physical protection of the generator windings is arranged to avoid the ingress of moisture, that is, a totally enclosed design, and in some wind turbines liquid cooling is used to reduce air-borne noise. A high slip (in some cases up to 2% at rated output power) is often requested by the wind turbine designer as this increases torsional compliance and damping in the wind turbine drive train and helps limit torsional oscillations in the drive train induced by the periodic torque variations of the aerodynamic rotor. However, this is at the expense of electrical losses in the rotor and the consequent generation of heat.

Figure 7.28 shows the conventional equivalent circuit of an induction machine that may be used to analyse its steady-state behaviour (Anaya Lara, 2009; Hindmarsh, 1984; McPherson, 1990). The rotor loss term is shown separated from the term representing mechanical power, which is a function of slip. The slip (s) is the difference between the angular velocity of the stator field and rotor:

$$s = \frac{\omega_s - \omega_r}{\omega_s} \quad (7.57)$$

For motor operation, the rotor runs slightly slower than the stator field and the slip is positive. For generator operation the rotor runs slightly faster than the stator field and the slip is negative.

Figure 7.29 shows how the active power varies with slip for a 1 MW induction machine. A convention has been chosen with the current flowing into the induction machine and so the normal operating region for a generator is between 0 and -1 MW. In this example, at 1 MW generation the slip is around -1% (-0.01 per unit) with the rotor rotating faster than the synchronous speed of the stator field. It may be seen that the maximum power that can be generated before the peak of the curve is reached is 2.6 MW. If the generator is connected to a distribution network with a low short-circuit level (and hence a high source impedance) the maximum power which may be exported before the peak of the curve is reached is reduced.

Figure 7.30 shows how the reactive power drawn by the generator varies with slip. The normal generating operating region is again shown. At 1 MW output the generator draws some 500 kVar. It may be seen that the reactive power requirement increases very rapidly if the output power, and hence slip, rises above its rated value.

Figures 7.29 and 7.30 may be combined to give the conventional circle diagram representation of an induction machine shown in Figure 7.31. Again the normal generating region

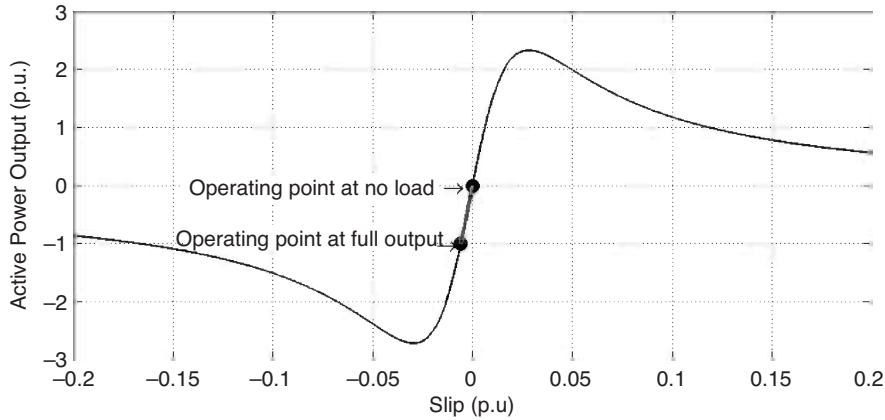


Figure 7.29 Variation of active power with slip for an induction machine

is shown. Fixed power factor correction capacitors (X_c) are useful to reduce the requirement for reactive power and so translate the circle diagram along the y-axis towards the origin (but not all the way otherwise there is a danger of self-excitation, see Chapter 10, Section 10.6.2). The equations used to describe the steady-state performance of induction generators are given in any standard undergraduate textbook (e.g. Hindmarsh, 1984; McPherson, 1990). Dynamic analysis is more complex but is dealt with by Krause (1986).

7.5.2 Variable slip induction generators

Variable slip operation is achieved by introducing an external resistance into the rotor circuit, as shown in Figure 7.32.

The external resistance is controlled by a semi-conductor switch. Below rated torque the switch short-circuits the external resistor to give no effect on the generator. Above rated torque,

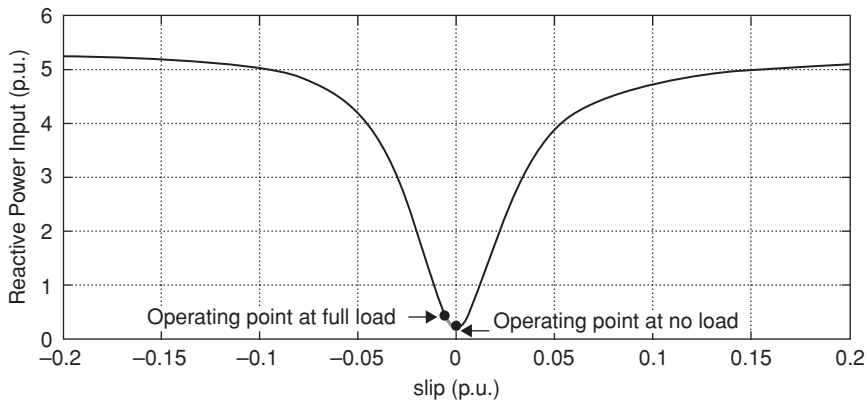


Figure 7.30 Variation of reactive power with slip for a 1 MW induction machine equipped with no-load power factor correction capacitors

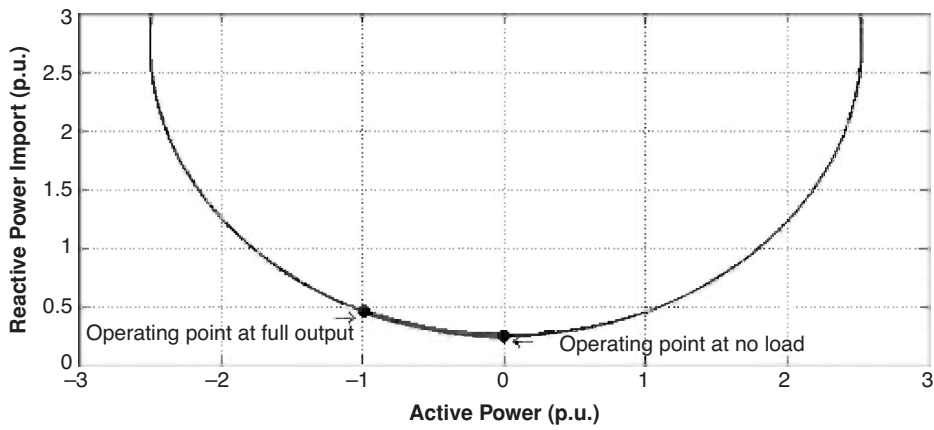


Figure 7.31 Circle diagram of 1 MW induction machine

Pulse Width Modulation control is used to introduce the external resistance progressively into the rotor circuit. The alteration of the torque-slip curve is shown in Figure 7.33. As more external resistance is added, the slope of the torque slip curve is reduced, for example to O-B. Below-rated operation along O-A is just like a fixed speed generator, but above rated the external resistor is varied continuously to maintain constant reaction torque, resulting in variable speed operation along A-B. Operation at point B, -2.8% slip (1542 rpm for a four-pole, 50 Hz generator) would result in losses of approximately 28 kW in a 1 MW generator. Increasing the rotor resistance R_r still further gives a greater speed range but beyond about -10% slip it becomes more difficult to dissipate the heat caused by the losses.

As in the fixed speed case, power factor correction capacitors are used to reduce the reactive power requirement.

7.5.3 Variable speed operation

There are two main approaches to electrical variable-speed operation. Either all the output power of the wind turbine is passed through two back-to-back frequency converters to give a wide range of variable speed operation, or a restricted speed range is achieved by converting only that fraction of the output power flowing in the rotor of a wound rotor induction machine.

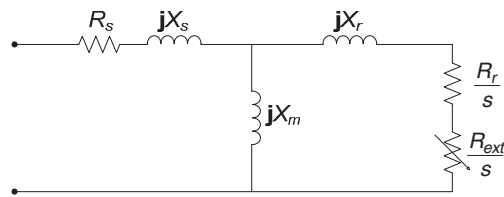


Figure 7.32 Steady state equivalent circuit of variable slip induction generator showing addition of external resistor R_{ext}

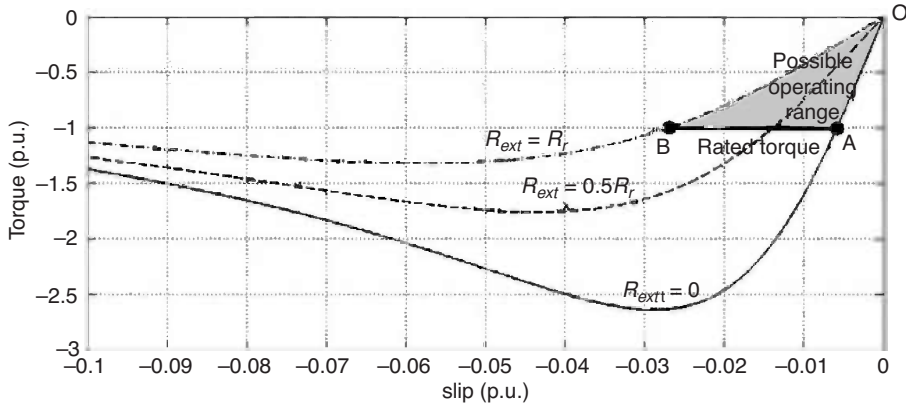


Figure 7.33 Effect of external resistance on the torque slip curve of an induction generator

In both cases, Graetz Bridge, voltage source converters are used with Insulated Gate Bipolar Transistors (IGBT) as the switching devices (Figure 7.34). The IGBTs are switched rapidly, typically at between 2–6 kHz with Pulse Width Modulation (PWM), to produce a close approximation to a sine wave voltage. Common techniques used to synthesize the sine wave voltage include carrier modulated (sine-triangular) PWM, hysteresis control and space vector modulation. All these modulation techniques produce quite similar results but space vector control is easier to implement in a digital control system. With rapid switching the voltage wave form approximates closely to a sine wave but at the expense of increased switching losses. The generator side converter rectifies all the power to DC, which is then inverted by the network side converter. Operation of this type of voltage source converter is described in Mohan, Undeland and Williams (1995) and Anaya Lara *et al.* (2009).

The IGBTs are switched to produce an approximation to a sine wave (Figure 7.35). Hence the fundamental function of the voltage source converters is as shown in Figure 7.36. Within their operating limits, the voltage source converters can create a voltage of any frequency,

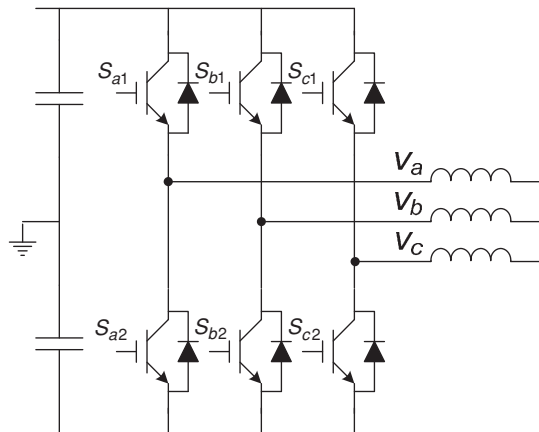


Figure 7.34 Voltage source converter

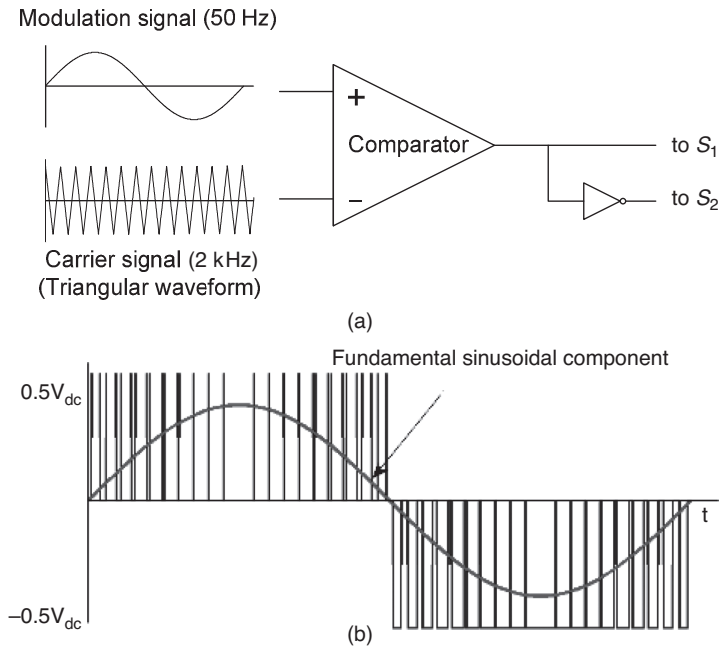


Figure 7.35 Carrier modulated (sine-triangular) Pulse Width Modulation (a) Sine-triangular modulation circuit (b) Pulse Width Modulation (PWM) output of sine-triangular modulation

phase or magnitude. They can be used to interface to the 50 or 60 Hz power system as well as the variable speed generator. In addition they can be used to apply a voltage at the slip frequency of a wound rotor induction machine in the Doubly Fed Induction Generator.

7.5.4 Variable speed operation using a Doubly Fed Induction Generator (DFIG)

In a variable slip generator, a speed increase, of the wound rotor induction generator, is achieved by adding resistance into the rotor circuit using an external resistor. The power consumed in the external resistor is directly proportional to slip speed. Thus, a 10% speed increase leads to losses in the external resistor of approximately 10% of the generator stator output power.

These additional losses do not compromise energy production because they occur only above rated, where surplus wind energy is being discarded anyway. However, this high level of losses is undesirable in large wind turbines because of the high cooling requirement to dispose of the resulting heat. Hence a development of the variable slip system has been to

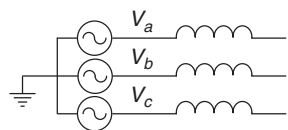


Figure 7.36 Ideal voltage sources from voltage source converters

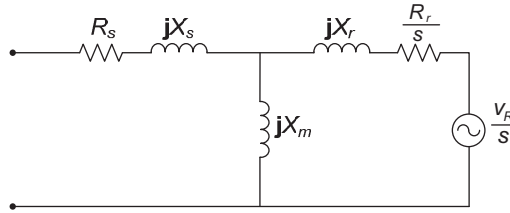


Figure 7.37 Steady state equivalent circuit of the DFIG

replace the external controlled resistance with a pair of back-to-back voltage source converters. These apply a variable voltage (and hence inject current) at the rotor slip frequency and so allow operation above and below the synchronous speed of the stator field. The synchronous speed of the stator field is determined by the network frequency and number of poles of the stator winding (e.g. 1500 rpm for a four-pole winding on a 50 Hz system) and a $\pm 30\%$ speed variation around this will require the power rating of rotor circuit converters to be approximately 30% of rated power.

The steady state equivalent circuit of the DFIG is shown in Figure 7.37. The external resistor of the variable slip generator is replaced by a voltage source. This applies a voltage to the slip rings of the wound rotor at slip frequency. The equivalent circuit has the rotor circuit referred to the stator and so the injected rotor voltage is divided by slip in the equivalent circuit.

The effect of injecting a voltage into the rotor is shown in Figure 7.38. The applied rotor voltages are quite small because, as shown in the equivalent circuit, the rotor voltage is divided by the slip.

Thus, with rated applied torque (-1 per unit), the speed may be varied by adjusting the injected voltage into the rotor circuit. Point B gives super-synchronous operation with power flowing out of the generator rotor. Point A gives sub-synchronous operation with power flowing into the generator rotor.

These power flows are shown in Figure 7.39.

The power relationships of the DFIG illustrate how the power flows in the rotor circuit vary with slip.

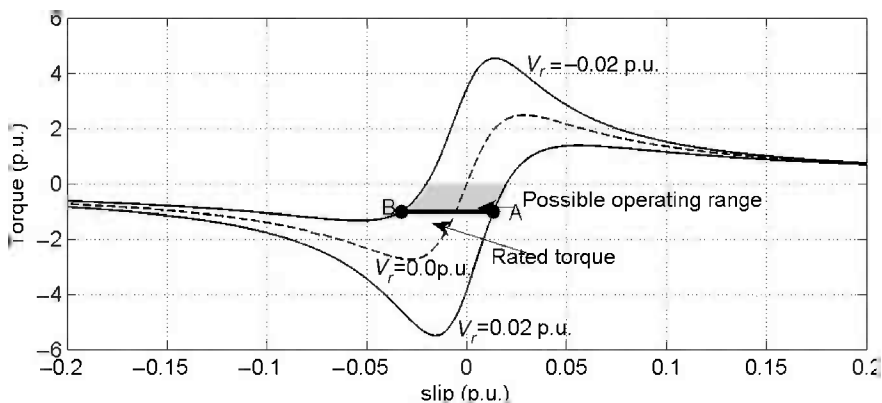


Figure 7.38 Steady-state torque slip curves of a DFIG

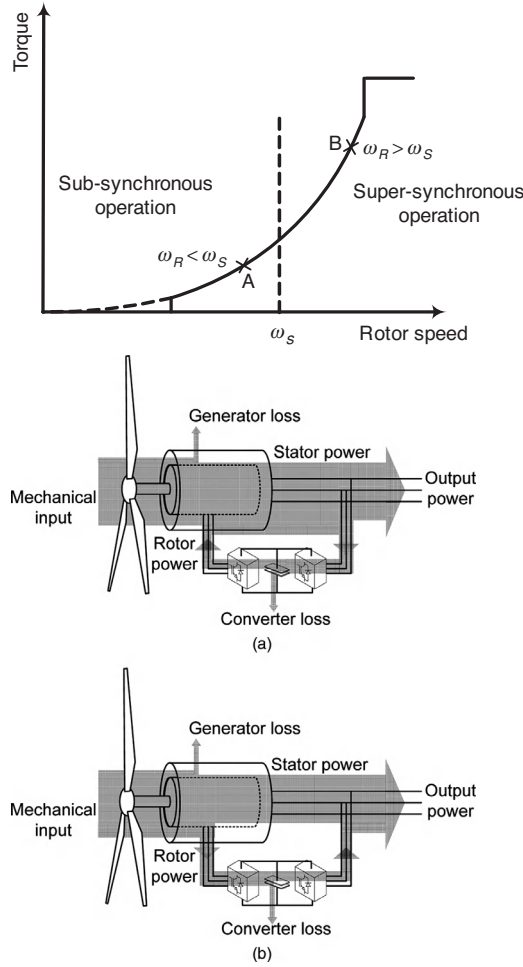


Figure 7.39 Power flows in a DFIG (a) Sub-synchronous operation (b) Super-synchronous operation

If the stator and rotor generator losses are neglected, the power transferred across the air gap of the generator is the same as the power in the stator. This is the mechanical input power minus the power flowing in the rotor circuit.

$$P_{air\_gap} = P_{stator} = P_{mech} - P_{rotor}$$

$$T\omega_s = T\omega_r - P_{rotor}$$

$$P_{rotor} = -T(\omega_s - \omega_r)$$

$$= -Ts\omega_s = -sP_{air\_gap}$$

$$= -sP_{stator}$$

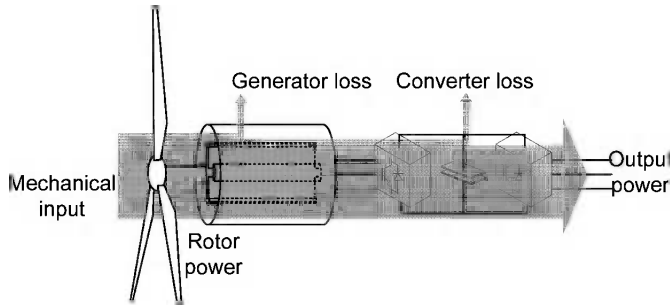


Figure 7.40 Power flows in a Full Power Converter

where T is torque on the generator shaft, ω_s is synchronous speed and ω_r is rotor speed, s : slip.

Thus, for a negative slip (super-synchronous operation) power flows out of the generator rotor while for positive slip (sub-synchronous operation) power flows into the generator rotor.

This ‘doubly fed’ concept was used in early large prototype wind turbines, for example, the 3 MW Growian constructed in Germany in the early 1980s and the Boeing MOD 5B in the USA at the same time. Then, cyclo-converters were used to change the frequency of the rotor circuit, but modern practice is to use two back-to-back voltage source converters. Control techniques vary but one approach is to use vector control techniques on the machine side converter to adjust torque and the excitation of the generator independently. The network side bridge maintains the voltage of the DC link (Pena, Clare and Asher, 1996; Muller *et al.*, 2002).

7.5.5 Variable speed operation using a Full Power Converter

Figure 7.40 shows the power flows in a full power converter, variable-speed generation system. All the power from the generator is rectified to DC and then inverted to the network voltage. This arrangement can be used with a range of generators. Induction generators with a gearbox mechanical transmission may be used in a configuration that is the inverse of the variable speed drives used for large mechanical loads, for example, pumps and fans. Wound rotor or permanent magnet synchronous generators may be used, either with a high speed generator coupled to the aerodynamic rotor through a gearbox, or with a slow speed multi-pole direct drive generator which avoids the need for a gearbox.

Early broad range variable-speed wind turbines used a diode rectifier bridge in the generator converter and a naturally commutated, thyristor, current source, converter on the network side (Freris, 1990). However, naturally commutated thyristor converters always consume reactive power and generate considerable characteristic harmonic currents. On weak distribution systems it is difficult to provide suitable filtering and power factor correction for this type of equipment.

Hence modern practice is to use two voltage source converters (Heier, 2006) in a manner similar to the arrangement of the rotor circuit of the DFIG although all the equipment must be rated at the full power of the wind turbine. The generator converter rectifies all the power to DC, which is then inverted by the network converter.

Control strategies vary but one approach is to use the two degrees of freedom of the generator converter output (magnitude and angle of the output voltage, or direct and quadrature axis voltages) to control the torque and excitation of the generator. Vector control is used to control the torque to a set-point obtained from the optimal wind turbine speed characteristic (Figure 6.14, Chapter 6) while the reactive power flow is used to supply excitation. The network converter then maintains the DC link voltage and exchanges reactive power with the network.

An alternative is to control the generator converter to maintain the DC link voltage at a constant value and then use the network converter to control the power flowing out of the system and hence the torque on the generator (Jones and Smith, 1993). A power bandwidth of 200–500 radians/s is quoted in this paper indicating the very fast control possible with such equipment with an overall efficiency of 92.1% consisting of 95.9% for the generator and 96% for the power electronics. The network side converter may be arranged to operate at any power factor within the rating of the equipment with very low harmonic distortion.

7.6 Mechanical brake

7.6.1 Brake duty

As indicated in Section 6.8.3, Chapter 6, a mechanical brake can be called on to fulfil a variety of roles, according to the braking philosophy adopted for the machine in question. The minimum requirement is for the mechanical brake to act as a parking brake, so that the machine can be stopped for maintenance purposes. The brake will also be used to bring the rotor to a standstill during high wind shutdowns for the majority of machine designs, and during low speed shutdowns as well in some cases. Aerodynamic braking is used to decelerate the rotor initially, so the mechanical brake torque can be quite low. However, IEC 61400-1 requires that the mechanical brake be capable of bringing the rotor to a complete stop from a hazardous idling state in any wind speed less than the one year return period three second gust (see Table 5.1).

If the mechanical brake is required to arrest the rotor in the event of a complete failure of the aerodynamic braking system, then there are two deployment options to consider. Either the mechanical brake can be actuated when an overspeed resulting from the failure of the aerodynamic system is detected, or actuated simultaneously with the aerodynamic brake as part of the standard emergency shut-down procedure. The advantage of the former strategy is that the mechanical brake will rarely, if ever, have to be deployed in this way, so that some pad or even disc damage can be tolerated when deployment actually occurs. In addition, fatigue loading of the gearbox will be reduced if the brake is mounted on the high speed shaft. On the other hand, if the mechanical brake is actuated before significant overspeed has developed, then the aerodynamic torque to be overcome by the mechanical brake in the event of aerodynamic braking failure will be less.

The most severe emergency braking case will arise following a grid loss during generation in winds above rated. In the case of pitch-regulated machines, the maximum overspeed will occur after grid loss at rated wind speed because the rate of change of aerodynamic torque with rotational speed decreases and soon becomes negative at higher wind speeds. Conversely, if the pitch mechanism should jam, the braking duty becomes more severe at wind speeds at or above cut-out, because much higher aerodynamic torques are developed as the rotor slows



Figure 7.41 High-speed shaft brake disc and calliper. Reproduced by permission of NEG-Micon

down and the angle of attack increases. For stall-regulated machines the critical wind speed is generally at an intermediate value between rated and cut-out.

7.6.2 Factors governing brake design

The braking torque provided by callipers gripping a disc brake (Figure 7.41) is simply the product of twice the calliper force, the coefficient of friction (typically 0.4), the number of callipers and the effective pad radius. Callipers providing clamping forces of up to 500 kN are available. However the brake design is also limited by:

- Centrifugal stresses in the disc.
- Pad rubbing speed.
- Power dissipation per unit area of pad.
- Disc temperature rise.

The nature of these constraints is described below.

The critical stress generated by centrifugal stresses is in the tangential direction at the inner radius of the brake disc, but it is governed principally by the disc rim speed according to the following formula:

$$\sigma_{\theta}(a) = \frac{3 + \nu}{4} \rho \omega^2 b^2 \left(1 + \frac{1 - \nu}{3 + \nu} \frac{a^2}{b^2} \right) \quad (7.58)$$

where a and b are the inner and outer disc radii respectively and ω is the disc rotational speed. One brake manufacturer, Twiflex, quotes a maximum safe disc rim speeds of around 90 m/s for their discs manufactured in spheroidal graphite cast iron.

Brake pads are generally made from sintered metal or a cheaper, resin based material. The former can accept rubbing speeds of up to 100 m/s, but some manufacturers quote permitted rubbing speeds for the latter of only about 30 m/s. However, Wilson (1990) reports satisfactory performance of resin based pads at a rubbing speed of up to 105 m/s if the power dissipation rate per unit area, Q , is kept low enough. The criterion, ascribed to Ferodo, is that $Q = \mu PV \leq 11.6 \text{ MW/m}^2$, where μ is the coefficient of friction, P is the brake pad pressure in KN/m^2 and V is the rubbing speed in m/s. This requires the pad pressure to be reduced to 275 KN/m^2 , assuming a friction coefficient of 0.4.

During braking the kinetic energy of the rotor and drive train together with the additional energy fed in by the aerodynamic torque are dissipated in the brake disc and pads as heat, resulting in rapid initial temperature rise near the surface of the brake disc. The rate of energy dissipation is equal to the product of the braking torque and the disc rotational speed, so in the latter stages of braking the rate of energy dissipation cannot sustain the high surface temperatures and they begin to fall again.

The coefficient of friction for pads of resin based materials is sensibly constant at a level of about 0.4 at temperatures up to 250°C , but begins to drop thereafter, reaching 0.25 at 400°C . Although in theory the brake can be designed to reach the latter temperature, in practice the varying torque complicates the calculations and leaves little margin of error against a runaway loss of brake torque. Accordingly 300°C is often taken as the upper temperature limit for resin based pads.

Sintered metal pads have a constant coefficient of friction of about 0.4 up to a temperature of at least 400°C , but manufacturers indicate that the material can perform satisfactorily at temperatures up to 600°C on a routine basis, or up to 850°C intermittently. Wilson (1990) reports a reduced friction coefficient of 0.33 at 750°C . Such temperatures cannot be realised in practice because the temperature of the disc itself is limited to 600°C in the case of spheroidal graphite cast iron or to a much smaller value in the case of steel (op cit).

Clearly the use of the more expensive sintered brake pads allows the brake disc to absorb much more energy. However, the sintered metal is a much more effective conductor of heat than resin based material, so it is often necessary to incorporate heat insulation into the calliper design to prevent overheating of the oil in the hydraulic cylinder.

A method of calculating brake disc temperature rise is given in the next section.

7.6.3 Calculation of brake disc temperature rise

The build up in temperature across the width of a brake disc over the duration of the stop can be calculated quite easily if a number of assumptions are made. Firstly, the heat generated is assumed to be fed into the disc at a uniform intensity over the areas swept out by the brake pads as the disc rotates. This is a reasonable approximation for a high speed shaft mounted brake and for a low speed shaft mounted brake with several callipers until rotation has almost ceased, but the energy input by this stage is much lower. Within the disc heat flow is assumed to be perpendicular to the disc faces only – that is, radial flows are ignored.

Consider a brake disc slice at a distance x from the nearest braking surface, of thickness Δx and cross sectional area A . The rate of heat flow away from the nearest braking surface entering the slice is, $\dot{Q} = -kA(d\theta/dx)$ (where θ is the temperature and k the thermal conductivity)

and the rate of heat flow leaving it on the far side is $\dot{Q} + d\dot{Q}/dx$. The temperature rise of an element of thickness Δx over a time interval Δt is given by:

$$\Delta\theta \cdot A \cdot \Delta x \rho C_p = \Delta Q = -\frac{d\dot{Q}}{dx} \Delta x \Delta t = k A \frac{d^2\theta}{dx^2} \Delta x \Delta t$$

where ρ is the density and C_p is the specific heat, so that:

$$\frac{d\theta}{dt} = \frac{k}{\rho C_p} \frac{d^2\theta}{dx^2} \quad (7.59)$$

Adopting a finite element approach, Equation 7.59 can be written:

$$\theta(x, t + \Delta t) = \theta(x, t) + \frac{k}{\rho C_p} \frac{\Delta t}{(\Delta x)^2} [\theta(x + \Delta x, t) + \theta(x - \Delta x, t) - 2\theta(x, t)] \quad (7.60)$$

Substituting values of $k = 36$ W/m per °K, $C_p = 502$ J/kg per °K and $\rho = 7085$ kg/m³ for Grade 450 spheroidal graphite cast iron yields a value for the thermal diffusivity $\alpha = k/(\rho C_p)$ of 1.01×10^{-5} m²/s. If the time increment, Δt , is selected at 0.025 seconds and the element thickness is taken as 1.005 mm, then Equation 7.60 simplifies to

$$\theta(x, t + \Delta t) = \theta(x, t) + 0.25 [\theta(x + \Delta x, t) + \theta(x - \Delta x, t) - 2\theta(x, t)] \quad (7.61)$$

This equation can be used to calculate the temperature distribution across the brake disc, starting with a uniform distribution and imposing suitable increments at the braking surfaces at the boundaries. The behaviour at the boundaries is simpler to follow through if they are treated as planes of symmetry like the disc mid-plane, with imagined discs flanking the real one. The temperature increment at the boundary at each time step, which is added to that calculated from Equation 7.61, is given by:

$$\Delta\theta_0 = \frac{2T\omega(t)\Delta t}{\Delta x \rho C_p S} \quad (7.62)$$

where T is the braking torque (assumed constant), $\omega(t)$ is the disc rotational speed at time t , and S is the area swept out by the brake pad (or pads) on one side of the disc. For a disc diameter D and pad width w , $S = \pi(D - w)w$. The factor 2 is required because heat is assumed to flow into the imagined disc as well as into the real one. Hence the initial temperature build up can be calculated as illustrated in Table 7.5, taking an arbitrary value of $\Delta\theta_0$ of 40°C. (The gradual reduction in $\Delta\theta_0$ over time due to deceleration is ignored here for simplicity.)

The brake disc surface temperature rise is found to be a minimum when the ratio of the braking torque to the maximum aerodynamic torque is about 1.6. As the ratio is reduced below this value, the extended stopping time results in more energy being abstracted from the wind, so temperatures begin to rise rapidly. On the other hand the maximum brake temperature is relatively insensitive to increases in the ratio above 1.6. The variation in maximum brake disc surface temperature with braking torque is illustrated for the emergency braking of a stall regulated machine following an overspeed in Figure 7.42, where the continuous line gives the surface temperature rise calculated by the finite element method outlined above.

Table 7.5 Illustrative example of calculation of brake disc temperature rise using finite element model

		Element	0	1	2	3	4	5
Time step	Time (sec)	Distance from braking surface (mm)	0	1.0	2.0	3.0	4.0	5.0
1	0.025	Initial temperature	0	0	0	0	0	0
		Boundary temperature increment	40					
		Temperature at end of time step	20	10	0	0	0	0
2	0.05	Boundary temperature increment	40					
		Sum	60	10	0	0	0	0
		Temperature at end of time step	35	20	2.5	0	0	0
3	0.075	Boundary temperature increment	40					
		Sum	75	20	2.5	0	0	0
		Temperature at end of time step	47.5	29.4	6.3	0.6	0	0
4	0.1	Boundary temperature increment	40					
		Sum	87.5	29.4	6.3	0.6	0	0
		Temperature at end of time step	58.5	38.2	10.6	1.9	0.1	0

It transpires that the maximum temperature rise can be estimated quite accurately by the following empirical formula:

$$\theta_{\max} - \theta_0 = \frac{E}{\sqrt{t}} \frac{1}{64600w(D-w)} = \frac{E}{\sqrt{t}} \frac{\pi}{64600S} \quad (7.63)$$

where E is the total energy dissipated in Joules, t is the duration of the stop in seconds and S is the area of the disc surfaces swept by the brake pads. The temperature derived using this formula is plotted as a dotted line in Figure 7.42 for comparison.

7.6.4 High speed shaft brake design

A key parameter to be chosen in brake design is the design braking torque. The coefficient of friction can vary substantially above and below the design value due to such factors as bedding in of the brake pads and contamination, so the design braking torque calculated on the nominal friction value must be increased by a suitable materials factor. The 1993 edition of the Germanischer Lloyd 'Regulation for the certification of Wind Energy Conversion Systems' specified a materials factor of 1.2 for the coefficient of friction, and added in another factor

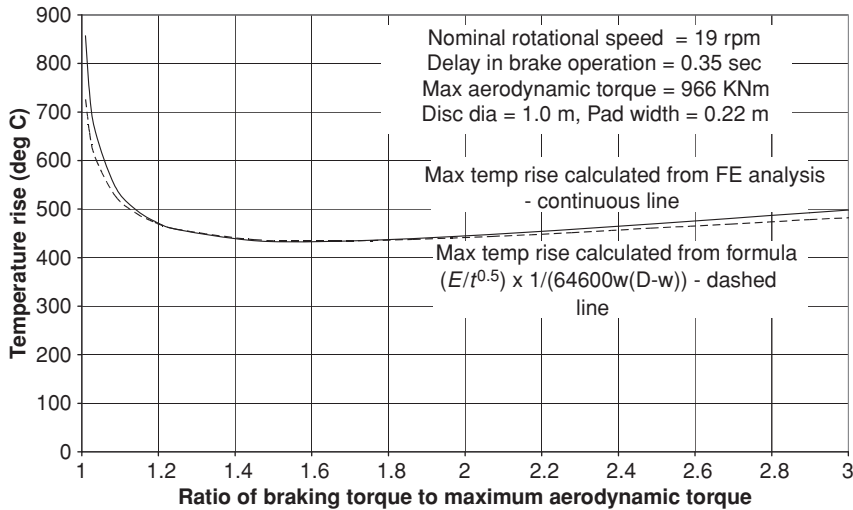


Figure 7.42 Brake disc surface maximum temperature rise for emergency braking of 60 m diameter, 1.3 MW stall regulated turbine from 10% overspeed in 20 m/s wind with HSS brake acting alone

of 1.1 for possible loss of calliper spring force. If these factors are adopted, the minimum design braking moment is 1.78 times the maximum aerodynamic torque, after including the aerodynamic load factor of 1.35. A small additional margin of, say, 5% should be added to ensure that the rotor is still brought to rest without a very large temperature rise should the 1.78 safety factor be completely eroded.

The procedure to be followed for the design of a brake on the high speed shaft (HSS) can conveniently be illustrated by an example.

Example 7.1

Design a HSS brake for a 60 m diameter, 1.3 MW stall-regulated machine capable of shutting the machine down in a 20 m/s wind from a 10% overspeed occurring after a grid loss, with or without assistance from the aerodynamic braking system. The nominal LSS and HSS rotational speeds are 19 rpm and 1500 rpm respectively, ignoring generator slip. Assume that the brake application delay time is 0.35 seconds, and that the inertia of the turbine rotor, drive train, brake disc and generator rotor – all referred to the low speed shaft – totals 2873 Tm^2 .

a) *Derivation of the brake design torque:* The peak aerodynamic torque occurs when the maximum rotational speed is reached just prior to brake application. The first step is to determine the relationship between rotational speed and aerodynamic torque for the stated wind speed of 20 m/s. From this the acceleration of the rotor and build-up of aerodynamic torque during the 0.35 seconds delay before the brake comes on can be determined. The speed increase in this case is 1 rpm, giving a maximum rotor speed of $19 \times 1.1 + 1 = 21.9$ rpm and peak aerodynamic torque of 966 KNm. Hence the brake design torque is $966 \times 1.78 \times 1.05 = 1800$ KNm referred to the low speed shaft, or $1800 \times 19/1500 = 22.8$ KNm at the brake.

b) *Brake disc diameter selection:* The maximum rotor speed corresponds to a high speed shaft speed of $21.9 \times (1500/19) = 1729$ rpm = 181 rad/sec, so the maximum permissible

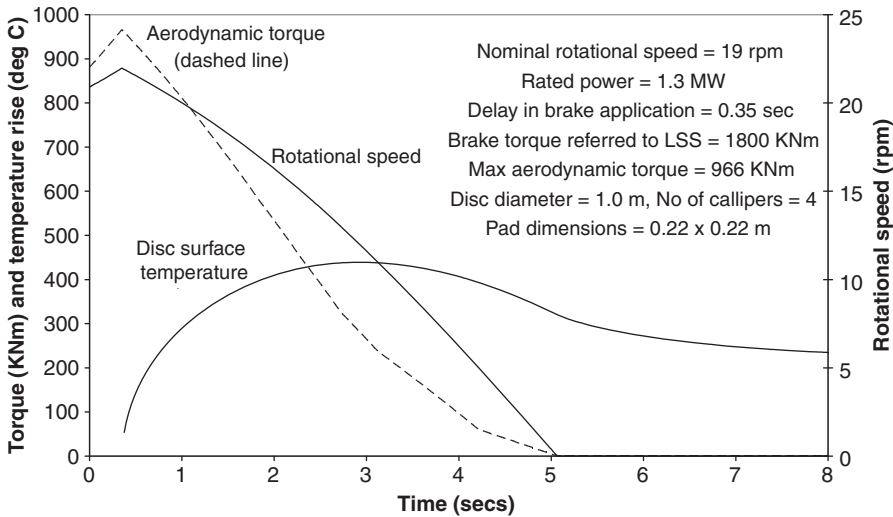


Figure 7.43 Emergency braking of stall regulated 60 m diameter turbine from 10 % over-speed in 20 m/s wind with HSS mechanical brake acting alone

brake disc radius as regards centrifugal stresses is about $90/181 = 0.497$ m. It is advisable to choose the largest permitted size in order to minimise temperature rise, so 1.0 m diameter is selected in this case. The pad rubbing speed will be quite acceptable if sintered pads are used.

c) *Selection of number and size of brake pads:* The total brake pad area is governed by the need to keep the maximum power dissipation per unit pad area below 11.6 MW/m^2 . The power dissipation is equal to the product of the braking torque and the rotational speed, so it is at a maximum at the onset of braking – that is, $22.8 \times 181 = 4128 \text{ kW}$, giving a required total area of the brake pads of $4128/11600 = 0.356 \text{ m}^2$. This area can be provided by four callipers fitted with 0.22×0.22 m pads, giving 0.387 m^2 in all.

d) *Maximum brake disc temperature check:* The variation in disc surface temperature over the duration of the stop can be calculated using the finite element method outlined in the preceding section. The resulting variation in this case is plotted in Figure 7.43. The surface temperature reaches a maximum of 440°C , just after halfway through the stop, which lasts 4.7 seconds from the time the brake comes on. This temperature is well below the limit for sintered pads.

e) *Calliper force:* The braking friction force required is 58.5 kN, calculated from the torque divided by the effective pad radius of 0.39 m. Hence the required calliper force is $58.5/(8 \times 0.4) = 18.3 \text{ kN}$ which is rather low for a calliper sized for a 0.22×0.22 m brake pad.

The design process outlined above results in an excessive number of lightly loaded callipers, because of the limitation on power dissipation per unit area. If the relative infrequency of emergency braking events allowed this limitation to be relaxed, then a more economic solution would result.

7.6.5 Two level braking

During normal as opposed to emergency shut-downs, the rotor is decelerated to a much lower speed by aerodynamic braking before the brake is applied, so the brake torque required

is much reduced. In view of the benefit of reduced loads on the braking system, and on the gearbox in particular, some manufacturers arrange for a reduced braking torque for normal shut-downs. This is achieved on the usual ‘spring applied, hydraulically released’ brake callipers by allowing oil to discharge from the hydraulic cylinder via a pressure relief valve when the brake is applied, so that the hydraulic pressure drops to a reduced level. After the rotor has come to rest, the remaining hydraulic pressure can be released, so that the brake torque rises to the full level.

7.6.6 Low speed shaft brake design

The procedure for designing a low speed shaft disc brake is much simpler than that for the high speed shaft brake, because the limits on disc rim speed, pad rubbing speed, power dissipation per unit area and temperature rise do not influence the design, which is solely torque driven. The large braking torque required means that a brake placed on the low speed shaft will be much bulkier than one with the same duty placed on the low speed shaft. For example, the design LSS braking torque of 1800 KNm from the example above would require a 1.8 m diameter disc fitted with seven callipers.

A study by Corbet *et al.* (1993), which investigated a range of machine diameters, concluded that the brake cost would double or treble if the brake were placed on the low speed shaft rather than on the high speed shaft. However, when the extra gearbox costs associated with a high speed brake were taken into account, the cost advantage of the high speed shaft brake disappeared.

7.7 Nacelle bedplate

The functions of the nacelle bedplate are to transfer the rotor loadings to the yaw bearing and to provide mountings for the gearbox and generator. Normally it is a separate entity, although in machines with an integrated gearbox, the gearbox casing and the nacelle bedplate could, in principle, be a single unit. The bedplate can either be a welded fabrication consisting of longitudinal and transverse beam members or a casting sculpted to fit the desired load paths more precisely. One fairly common arrangement is a casting in the form of an inverted frustum which supports the low speed shaft main bearing at the front and the port and starboard gearbox supports towards the rear, with the generator mounted on a fabricated platform projecting to the rear and attached to the main casting by bolts.

Although conventional methods of analysis can be used to design the bed plate for extreme loads, the complicated shape renders a finite element analysis essential for calculating the stress concentration effects needed for fatigue design. Fatigue analysis is complicated by the need to take into account up to six rotor load components. However, given stress distributions for each load component obtained by separate FE analyses, the stress time history at any point can be obtained by combining appropriately scaled load component time histories previously obtained from a load case simulation.

7.8 Yaw drive

The yaw drive is the name given to the mechanism used to rotate the nacelle with respect to the tower on its slewing bearing, in order to keep the turbine facing into the wind and to

unwind the power and other cables when they become excessively twisted. It usually consists of one or more electric or hydraulic motors mounted on the nacelle each of which drives a pinion mounted on a vertical shaft via a reducing gearbox. The pinion engages with gear teeth on the fixed slewing ring bolted to the tower, as shown in Figure 7.44. These gear teeth can either be on the inside or the outside of the tower, depending on the bearing arrangement, but they are generally located on the outside on smaller machines so that the gear does not present a safety hazard in the restricted space available for personnel access.

The yaw moments on rigid hub machines arise from differential loading on the blades, which may be broken down into deterministic and stochastic components. On a three bladed machine, the dominant cyclic yaw loading is at 3P, but it is generated by 2P blade loading, as is demonstrated below. Defining blade out-of-plane root bending moments containing harmonics of the rotational frequency, $\omega/2\pi$, as follows:

$$M_{Yj} = \sum_n a_n \sin \left(n \left\{ \omega t + \frac{2\pi(j-1)}{3} \right\} + \phi_n \right) \quad (7.64)$$

Hence, the yaw moment from all three blades is given by

$$\begin{aligned} M_{ZT} = & \sin \omega t \sum_n a_n \sin (n\omega t + \phi_n) + \sin \left(\omega t + \frac{2\pi}{3} \right) \sum_n a_n \sin \left(n \left\{ \omega t + \frac{2\pi}{3} \right\} + \phi_n \right) \\ & + \sin \left(\omega t - \frac{2\pi}{3} \right) \sum_n a_n \sin \left(n \left\{ \omega t - \frac{2\pi}{3} \right\} + \phi_n \right) \end{aligned} \quad (7.65)$$

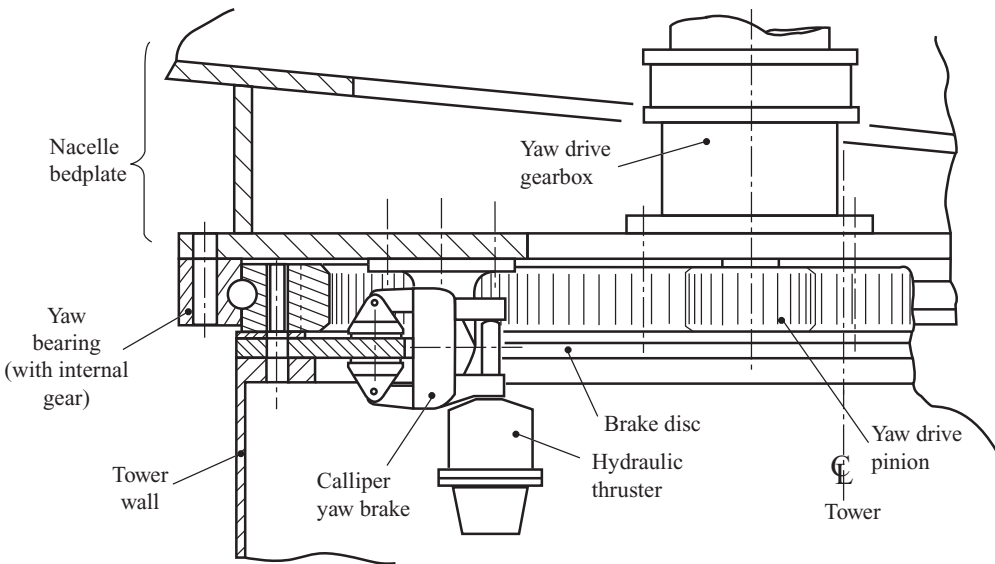


Figure 7.44 Typical arrangement of yaw bearing, yaw drive and yaw brake

that is,

$$M_{ZT} = \sum_n a_n \left[\sin \omega t \sin(n\omega t + \phi_n) \left\{ 1 - \cos \frac{2\pi n}{3} \right\} + \sqrt{3} \cos \omega t \cos(n\omega t + \phi_n) \sin \frac{2\pi n}{3} \right] \quad (7.66)$$

For the first four harmonics this gives:

$$M_{ZT} = 1.5 \{a_1 \cos \phi_1 - a_2 \cos(3\omega t + \phi_2) + a_4 \cos(3\omega t + \phi_4)\} \quad (7.67)$$

Thus, it is seen that the blade out-of-plane bending harmonics at 2P and 4P produce yaw moment at 3P, while those at 1P and 3P produce steady and zero yaw moments respectively.

As turbine size increases, the turbine diameter becomes larger in relation to gust dimensions, and the scope for differential loading on the blades due to turbulence increases. The expression for the standard deviation of the stochastic yawing moment on a three-bladed machine is the same as that for the shaft moment standard deviation – see Equation 5.119.

Anderson *et al.* (1993) investigated yaw moments on two sizes of Howden three-bladed turbines – (33 m diameter, 330 kW and 55 m diameter 1 MW) and concluded that the major source of cyclic yaw loading is stochastic at 3P. Yaw error, on the other hand, was not found to make a significant contribution. Given that yaw error results in a blade out-of-plane load fluctuation at rotational frequency, this result is in accordance with Equation 7.67.

Several different strategies have been evolved for dealing with the large cyclic yaw moments that arise on rigid hub machines due to turbulence, as follows.

1. *Fixed yaw*: A yaw brake is provided in the form of one or more callipers acting on an annular brake disc and is designed to prevent unwanted yaw motion under all circumstances. See Figure 7.44. This can require 6 callipers on a 60 m diameter machine. During yawing, the yaw motors drive against the brake callipers, which are partly released, so that the motion is smooth.
2. *Friction damped yaw*: Yaw motion is damped by friction in one of three different ways. In the first, the nacelle is supported on friction pads resting on a horizontal annular surface on the top of the tower. The yaw drive has to work against the friction pads, which also allow slippage under extreme yaw loads. This system was employed on the 500 kW Vestas V39 and the 3 MW WEG LS1.

In the second, the nacelle is mounted on a conventional rolling element slewing bearing, and the friction is provided by a permanently applied brake, using the same configuration as for fixed yaw. Optionally, the pressure on the brake pads can be increased when the machine is shutdown for high winds.

In the third, the nacelle is supported on a three row roller type slewing bearing (see Figure 7.21(d)), but with the rollers replaced by pads of elastomer composite to generate friction.

3. *Soft yaw*: This is hydraulically damped fixed yaw. The oil lines to each side of the hydraulic yaw motor are each connected to an accumulator via a choke valve, allowing limited damped motion to and fro to alleviate sudden yaw loads. This system is used

on the 300 kW WEG MS3, which has a two bladed, teetered rotor, but experiences significant yaw loads when teeter impacts occur.

4. *Damped free yaw*: A hydraulic yaw motor is used as before, but the oil lines to each side of the motor are connected together in a loop via a check valve, rather than being connected to a hydraulic power pack. This arrangement prevents sudden yaw movements in response to gusts, but depends on yaw stability over the full range of wind speeds. Unfortunately, yaw stability in high winds is rare.
5. *Controlled free yaw*: This is the same as damped free yaw, except that provision is made for yaw corrections when necessary. This strategy was adopted successfully on several Windmaster machines, including the two-bladed, fixed hub 750 kW machine.

Friction damped yaw is the strategy most commonly adopted.

7.9 Tower

7.9.1 Introduction

The vast majority of wind turbine towers are constructed from steel. Concrete towers are a perfectly practicable alternative, but, except at the smaller sizes, they require the transfer of a substantial element of work from the factory to the turbine site, which has not normally proved economic. Accordingly, this section concentrates on the two types of steel towers – tubular and lattice. The restrictions on first mode natural frequency are considered first.

7.9.2 Constraints on first mode natural frequency

As noted in Section 6.14, Chapter 6, it is important to avoid the excitation of resonant tower oscillations by rotor thrust fluctuations at blade passing frequency or, to a lesser extent, at rotational frequency. Dynamic magnification impacts directly on fatigue loads, so the further the first mode tower natural frequency is from the exciting frequencies, the better.

In the case of machines operating at one of two fixed speeds, the latitude available for the selection of the tower natural frequency is more restricted. Figure 7.45 shows the variation of dynamic magnification factor with tower natural frequency for excitation at upper and lower blade passing and rotational frequencies for a three bladed machine with a 3:2 ratio between the upper and lower speeds. The curves are plotted for a damping ratio of zero, but the difference if the curves were plotted for a realistic damping ratio of about 2% would be imperceptible. The figure also shows the tower natural frequency bands available if the dynamic magnification ratio were to be limited to 4 for all four sources of excitation. It is apparent that the minimum dynamic magnification ratio obtainable with a tower natural frequency between the upper rotational frequency and lower blade passing frequency is 1.65, for a tower natural frequency of 0.79 times the lower blade passing frequency.

Once a satisfactory tower design – in terms of strength and natural frequency – has been evolved for a given turbine, it is a straightforward matter to scale up the machine to larger rotor sizes, provided all the tower dimensions are scaled similarly, the hub height wind speed is unchanged, and the tip speed is maintained constant. It can be shown that in these circumstances the tower natural frequency varies inversely with rotor diameter, as does

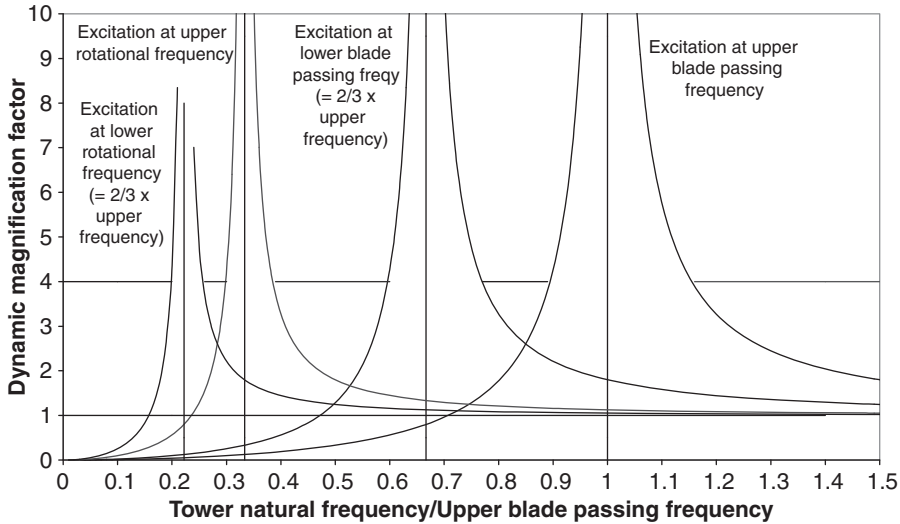


Figure 7.45 Variation of dynamic magnification factors with tower natural frequency for a two speed, three-bladed m/c

the rotational speed of the rotor, so that the dynamic magnification factors are unchanged. Similarly, tower stresses due to extreme wind loading are the same as before.

The situation is less straightforward if the tower height is to be varied for a particular turbine. Assuming, as before, that the extreme hub height wind speed remains the same, and that the wind loading on the tower is negligible compared with the wind loading on the rotor, then the tower base overturning moment is simply proportional to hub height, H . Constant stresses can be maintained at the tower base by scaling all cross section dimensions up in proportion to the cube root of the hub height. If the same scaling is maintained all the way up the tower, then the tower natural frequency will vary as $\sqrt{I_B/H^3} = \sqrt{H^{4/3}/H^3} = 1/H^{5/6}$, neglecting tower mass, where I_B is the second moment of area of the tower base cross section. Thus, doubling the tower height would result in a 44% reduction in natural frequency. Alternatively, if the tower base overturning moment were assumed to vary as $H^{1.5}$ to allow for the effect of wind shear on hub height wind speed and the contribution of wind loading on the tower, then constant tower base stresses could be maintained by scaling the cross section dimensions up by $\sqrt{H}$. On this basis, tower natural frequency would vary as $1/\sqrt{H}$.

The practical consequences of ‘tuning’ the tower natural frequency are discussed with respect of tubular towers in the next section.

7.9.3 Steel tubular towers

In the absence of buckling, a waisted conical shell, with a semi angle of 45° below the critical zone for tip clearance, would be the most efficient structure for transferring a horizontal rotor thrust acting in any direction to ground level. However, apart from the practicalities of transport and erection, instability of thin walled shells in compression precludes such a design solution, and the steel tubular towers in common use have a very modest taper. It can be noted in passing that the manufacture of gently tapering towers has only been made possible by the

development of increasingly sophisticated rolling techniques, and that early tubular towers were constructed from a series of cylindrical tubes of decreasing diameter with short ‘adaptor’ sections welded between them.

A tapered tower is generally fabricated from a series of pairs of plates rolled into half frusta and joined by two vertical welds. The height of each frustum so formed is limited to two or three metres by the capacity of the rolling equipment. Care has to be taken in the execution of the horizontal welds to minimise local distortion, which weakens the tower under compression loading.

Assuming that a tower design with a uniform taper is to be adopted, the key design parameters to establish are the diameter and wall thickness at the tower base. The tower top diameter, on the other hand, is governed by the size of the yaw bearing.

The main considerations determining the tower dimensions at the base are buckling of the shell wall in compression, strength under fatigue loading and stiffness requirements for ‘tuning’ the natural frequency. These are dealt with in separate sub-sections below.

As machines get larger, another important consideration is the maximum tower base diameter that can be accommodated on the highway when tower sections are transported overland. In the flat terrain of North Germany and Denmark, this limit is generally 4.0–4.2 m, but elsewhere it will often be less.

Design against buckling

Given perfect geometry, the strength of a cylindrical steel tube in axial compression is the lesser of the yield strength and the elastic critical buckling stress, given by

$$\sigma_{cr} = 0.605Et/r \quad (7.68)$$

where r is the cylinder radius and t is the wall thickness. Yield strength governs for r/t less than $0.605E/f_y$, which equates to 506 for mild steel, with $f_y = 245$ MPa. However, the presence of imperfections, particularly those introduced by welding, means that the tower wall compression resistance is significantly reduced, even at the relatively low tower wall radius to thickness ratios normally adopted. There is quite a wide disparity between the provisions of different national codes, with some making an explicit link between compression resistance and tolerances on imperfections and others not.

The provisions of EN 1993-1-6:2007 – ‘Eurocode 3 – Design of steel structures – Strength and stability of shell structures’ will be described here.

The first step is to decide the fabrication tolerance quality class, based on the imperfection tolerances that can be realistically achieved in the production facility.

The limits on the out-of-plane deviations, w , of the cylinder, or ‘dimples’, measured with

- (a) a rod of length $L = 4\sqrt{rt}$ placed vertically, away from welds,
- (b) a circular template of the same length placed horizontally, away from welds or
- (c) a rod of length $L = 25t$ placed vertically across horizontal welds

as a percentage of the requisite gauge lengths are given for different fabrication tolerance quality classes in Table 7.6, which also gives corresponding values of the fabrication quality parameter, Q .

Table 7.6 Recommended dimple tolerance and corresponding value of the fabrication quality parameter for different fabrication tolerance quality classes

Fabrication tolerance quality class	Description	Recommended limit on percentage deviation	Fabrication quality parameter, Q
Class A	Excellent	0.6 %	40
Class B	High	1.0 %	25
Class C	Normal	1.6%	16

Having determined the appropriate fabrication quality parameter, the meridional elastic imperfection reduction factor, α_x , and the plastic limit relative slenderness, λ_p , can be determined according to

$$\alpha_x = \frac{0.62}{1 + 1.91 \left(\frac{1}{Q} \sqrt{\frac{r}{t}} \right)^{1.44}} \quad (7.69)$$

and

$$\lambda_p = \sqrt{\frac{\alpha_x}{0.4}} \quad (7.70)$$

The buckling strength reduction factor, χ , is then given by

$$\chi = 1 - 0.6 \left(\frac{\lambda - \lambda_0}{\lambda_p - \lambda_0} \right) \quad (7.71)$$

where λ is the relative shell slenderness, $\sqrt{f_y/\sigma_{cr}}$, σ_{cr} is the elastic critical meridional buckling stress and λ_0 is the squash limit relative slenderness. Both the latter parameters depend on the proportion, ε , the axial stress forms of the total, as follows:

$$\sigma_{cr} = 0.605E \frac{t}{r} (1 - 0.4\varepsilon) \quad (7.72)$$

and

$$\lambda_0 = 0.3 - 0.1\varepsilon \quad (7.73)$$

As wind turbine tower stresses are dominated by bending stress, ε is small and can be ignored for preliminary design. Figure 7.46 shows how the buckling strength reduction factor varies with the shell radius to thickness ratio for the different fabrication tolerance quality classes under the assumption that axial stress is negligible. Note that the plot shows the buckling strength reduction factor divided by the partial safety factor for materials strength of 1.2 specified in IEC 61400-1 for global buckling of curved shells. Also plotted (as a dashed line) is the corresponding curve specified in the GL rules (2005) for a 1% limit on dimple depth. In this case the buckling strength reduction factor is divided by partial safety factor for materials

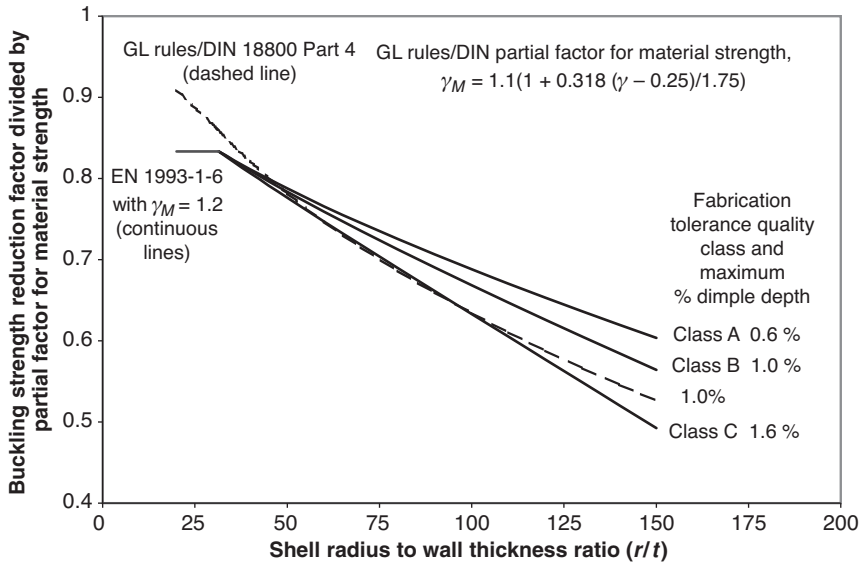


Figure 7.46 Variation of buckling strength reduction factor, divided by partial safety factor for material strength, with shell radius to thickness ratio, for zero axial stress

strength given by the GL rules/DIN (which varies with relative shell slenderness, λ), allowing the comparison of the design strengths obtained by the two methods.

The effect of the choice of tower base diameter on total tower weight is best illustrated by reference to a concrete example. Consider the design of a 50 m hub height tower in mild steel for a 60 m diameter, three-bladed, stall regulated turbine at a site with a 60 m/s extreme wind speed. The tower base wall thickness required to resist the overturning moment produced by this wind speed has been calculated for a range of tower base diameters with the aid of Equation 7.71 and plotted on Figure 7.47. Corresponding tower weights have also been plotted, based on a tower top diameter and wall thickness of 2.25 m and 11 mm respectively and assuming an idealised linear wall thickness variation between tower top and tower base. It can be seen that the tower weight reaches a minimum value at about 4.5 m diameter, indicating that beyond this point the reduction in cross-sectional area for constant section modulus is offset by the effects of the reducing buckling strength and the increasing wind loading on the tower itself. The weight penalty resulting from restricting the tower base diameter to 4.0 m for transport purposes would, in this case, be negligible.

Fatigue design

Clear rules for the fatigue design of steel welded structures are given in EN 1993-1-9:2005 ‘Eurocode 3: Design of steel structures – Part 1–9: Fatigue’, where a family of $S-N$ curves is defined for different weld details. On a log-log plot these curves in fact consist of two straight lines, with slopes of $\frac{1}{5}$ and $\frac{1}{3}$ for numbers of cycles above and below 5×10^6 respectively. In addition, there is a cut-off limit at $N = 10^8$ cycles, so that stress cycles with a stress range smaller than that defined at 10^8 cycles are deemed not to cause any fatigue damage at all.

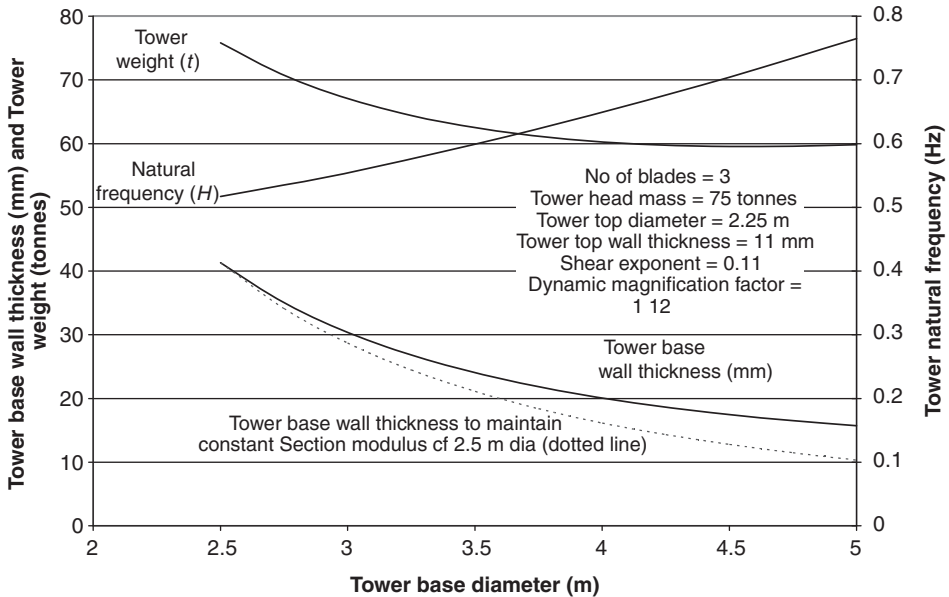


Figure 7.47 Variation in tower base wall thickness with diameter required for support of 60 m diameter stall regulated wind turbine at 50 m hub height in 70 m/s extreme wind speed

Excluding the tower doorway (which is considered later) the critical weld details on a steel tubular tower are likely to be at welded attachments for intermediate platform and cable support members and the horizontal welds to the tower base flange and intermediate bolted flanges. Assuming a full penetration butt weld is provided (Figure 7.49, top) and that the flange thickness is between 50 mm and 80 mm, the detail category number for the horizontal welds is 71 (where the number 71 indicates the stress range applicable at 2×10^6 cycles in MPa). The detail category number for longitudinal welded attachments reduces as the length of the attachment increases, but if the attachment length can be restricted to 80 mm, the detail category number of 71 applies here as well. The $S-N$ curve for this detail category is shown in Figure 7.48.

Where tower design is governed by fatigue, tower weight can be reduced by selecting weld details corresponding to higher detail categories. This has led to the introduction of ‘weld-neck’ flanges, where the ‘neck’ constitutes a short section of the tower wall (Figure 7.49, lower half), so that the weld is a standard transverse butt weld (detail category 90) rather than a tee-butt weld (detail category 71). Similarly the length of welded attachments can be reduced to 50 mm, to raise the detail category to 80.

Eurocode 3 recommends different partial safety factors for fatigue strength, γ_{Mf} , according to the consequences of failure and the assessment method. If load redistribution can occur in the event of fatigue damage, then the component concerned can be assessed by the ‘damage tolerant method’, with γ_{Mf} taken as 1.0 and 1.15, for low and high consequence of failure respectively. On the other hand, if local formation of cracks in a structural element could rapidly lead to its failure, assessment should be by the ‘safe-life method’, with increased values of γ_{Mf} of 1.15 and 1.35. In a welded tubular structure, there is no barrier to the

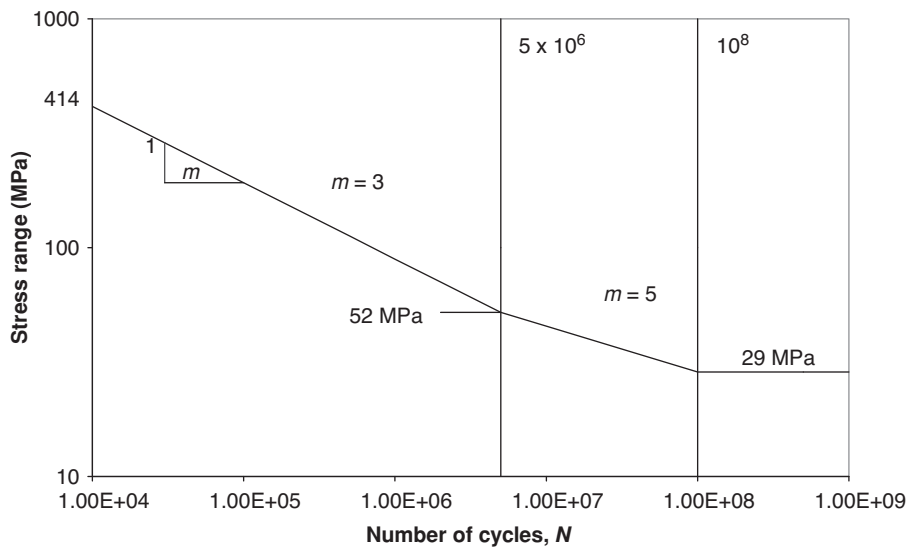


Figure 7.48 Eurocode 3 fatigue strength curve for detail category 71 (butt welded *T* joint)

propagation of a fatigue crack that has reached a critical length, so the designer must decide whether an inspection regime can be designed to detect incipient cracks before they become critical. Otherwise the tower needs to be assessed by the ‘safe-life’ method.

IEC 61400-1 adopts a similar approach, but some of the partial safety factors for fatigue strength are less conservative.

The derivation of fatigue load spectra and the combination of stress ranges due to M_X and M_Y load spectra are discussed in Section 5.12.6, Chapter 5.

Relative criticality of extreme and fatigue loads

The relative criticality of buckling failure (under extreme loads) and fatigue loads depends on a variety of factors. However, fatigue is more likely to be critical on pitch regulated machines than on stall regulated ones, because of the increased rotor thrust fluctuations above rated and the reduced extreme loading at standstill. Fatigue is also more likely to be critical at low wind speed sites, because the percentage reduction in extreme loads is less than the percentage reduction in fatigue equivalent load.

Tuning of tower natural frequency

Considerable scope exists, at least in theory, for adjusting the tower natural frequency to a suitable value by varying the base diameter, while maintaining the necessary strength against extreme and fatigue loading. The effect on natural frequency of varying tower base diameter by a factor of two, for a case where extreme loading governs, is illustrated for a 60 m diameter stall regulated machine at 50 m hub height in Figure 7.47. The frequency increases from 0.517 Hz for a 2.5 m base diameter to 0.765 Hz for a 5.0 m diameter. Now the rotational speed of a 60 m diameter turbine to yield a 60 m/s tip speed is about 19 rpm. If we assume

that the machine is two speed, with a lower rotational speed of $19 \times 2/3 = 12.67$ rpm, then the lower blade passing frequency will be 0.633 Hz – right in the middle of the available tower natural frequency range. Adopting a $\pm 15\%$ frequency exclusion zone, the tower natural frequency is required to be less than 0.538 Hz or more than 0.728 Hz. However, a frequency of 0.728 Hz would require a diameter of about 4.7 m (without making the tower wall thicker than necessary for the strength requirement), which is likely to be ruled out by transport considerations. Thus, the only strength limited design option is one with a base diameter of 2.75 m, with a weight penalty of about 10 tonnes compared with the 60 tonne optimum design, giving a natural frequency of about 0.535 Hz. Alternatively a 4 m base diameter could be chosen and the wall thickness increased by 37% to 27.5 mm to give a frequency of about 0.728 Hz. However, the weight penalty in this case is over 15 tonnes.

The above case study illustrates the fact that it is not always economic to satisfy the natural frequency requirements for a particular combination of turbine and hub height. In these circumstances it may well be preferable to change the hub height. For example, a hub height of 55 m would work much better for the case described, with a tower base diameter of 3.5 m yielding a natural frequency of 0.535 Hz and a tower weight of 74 Tonnes.

Joints between tower sections

Towers are normally fabricated in several sections for transport reasons, so joints are required. Welding on site is an expensive operation, so bolted joints are almost always used, although sleeved joints, in which each tapered tower section is threaded over the one beneath and forced into place by jacking, have been used successfully.

The structurally most effective joint is made with friction grip bolted splice plates oriented vertically and sandwiching the walls of the abutting tower sections between them. Provided the grip force is adequate, the joint will not slip even under the extreme load, with the result that the bolts are not subject to fatigue loads. Unfortunately, apart from the effect of splice plates on the external appearance, there are practical difficulties of joint assembly, because bolting requires the provision of some form of personnel access on the outside of the tower. Nevertheless splice plates have been used on some towers.

The most widely used bolted arrangement is the internal flanged joint as illustrated in Figure 7.49. The flanges are butt welded to the ends of the mating sections, with the flange outer edge flush with the tower wall. Alternatively the flange may be formed with a stub section of tower wall already attached. Such flanges, which are termed weld neck flanges, provide a smoother transition from wall to flange (as illustrated in the lower half of figure 7.49) and result in a higher butt weld detail category.

After assembly, each bolt is torqued or tensioned to induce a preload between the flanges in order to minimise in-service bolt fatigue stresses. The bolt should be initially sized to resist the prying force induced by the extreme tower wall tensile stresses – taking the fulcrum adjacent to the flange inner edge – and then checked for fatigue.

The fatigue calculation for the bolts in a flanged joint depends on the relationship between the bolt load and tower wall stress, which only remains linear while contact is maintained over the full flange width. The VDI Guideline for 'The Systematic Calculation of High Duty Bolted Joints', VDI 2230, gives a method for calculating the bolt load increment as a proportion of the load increment in the 'tributary' width of tower wall under these conditions. The axial loading on the flanged joint and the effect of the moment due to the eccentricity of loading are considered separately. The axial load is assumed to be shared between the bolt and the

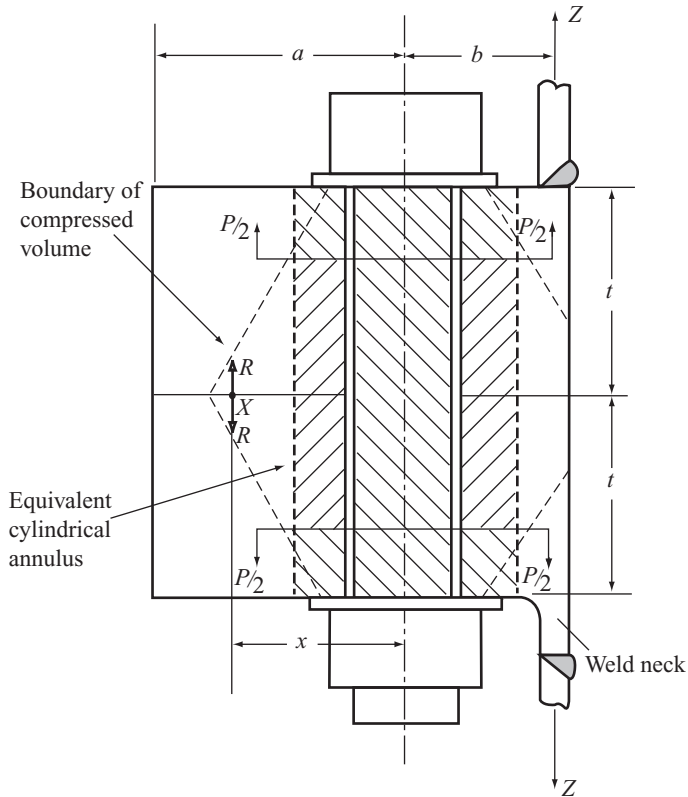


Figure 7.49 Bolted flange joint

preloaded flanges in proportion to the stiffnesses of the load paths, which, in the case of the flanges, is based on a reduced cross-sectional area related to the volume compressed by the preload according to:

$$A_{ers} = \frac{\pi}{4} (d_w^2 - d_h^2) + \frac{\pi}{8} d_w (D_A - d_w) [(k + 1)^2 - 1] \quad \text{where} \quad k = \sqrt[3]{\frac{l_k d_w}{D_A^2}} \quad (7.74)$$

and

d_w is the washer face diameter on the bolt head and nut

d_h is the bolt hole diameter

l_k is the clamping length between bolt head and nut

D_A is twice the distance from the bolt centreline to the nearest flange edge, or the bolt spacing whichever is the less.

The Guideline recognises that the effective plane of introduction of the external load will not necessarily be immediately under the bolt head or nut, but may lie nearer the flange mid-plane, giving the load paths distinguished by different cross hatching in Figure 7.49.

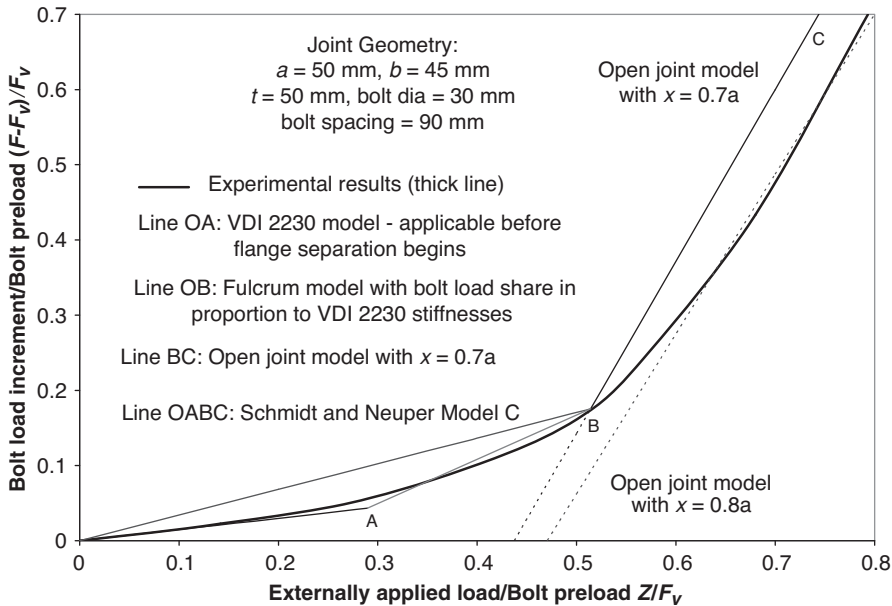


Figure 7.50 Flange joint bolt load variation with externally applied load – experimental results and engineering models compared

Stresses due to the eccentricity of the tower wall load to the flange contact area are dealt with by ordinary bending theory applied to the whole contact area.

The VDI 2230 method outlined above no longer applies once a gap has opened up between the flanges at the outer edge. For larger fluctuations in the externally applied load, Z , the fulcrum model can be used, although it is inevitably conservative at low loads. The axial load, P , applied to the bolt/flange combination is calculated on the basis that a fulcrum exists at X , a distance x from the bolt, so that $P = Z(1 + b/x)$, and the load share between the bolt and the compressed volume of flange is calculated according to the relative stiffnesses as before.

In Figure 7.50, the two linear relationships between bolt load increment and externally applied load are compared with experimental results for a particular test specimen with a single flange bolt. It is assumed that the planes of introduction of the load on the bolt/flange combination are immediately under the bolt head and nut in each case. Line OA shows the VDI 2230 model, with the point A representing the limit of its validity. Line OB shows the fulcrum model, with B representing the point at which the preload between the flanges at the position of the bolts disappears. Thereafter, the bolt load varies as $Z(1 + b/x)$ – That is, along line BC for $x/a = 0.7$. It may be noted from Figure 7.50 that a value of x/a of 0.8 results in better agreement with the test results at high loads, but these are not of interest for design purposes.

Schmidt and Neuper (1997) have proposed a more sophisticated model identified as ‘Model C’, which combines aspects of the two models already described and gives a bolt load characteristic consisting of the three straight lines OA, AB and BC (see Figure 7.50). Clearly this agrees much better with the experimental results, but it adds to the complexity of the fatigue load calculation.

Uniformity of bolt loading around the tower clearly depends on the accuracy of the mating flange surfaces. Schmidt *et al.* (1999) have investigated the effects of various imperfections using a finite element model and made tentative suggestions regarding permitted tolerance levels.

Tower tie-down

The tower is normally fitted with a base flange, which can either be attached to the foundation by screwed rods cast into the concrete or bolted to an embedded tower stub. This sub section is concerned with the former arrangement.

The screwed rods are normally anchored in some way at their base, and their capacity to resist overturning moment is determined by the pull-out resistance of the semi-circle of bolts on the upwind side. As this is governed by the concrete shear strength, the rods have to be anchored quite deep into the concrete, so that their length is typically similar to the tower base radius.

The fatigue loads in the tie-down rods can be considerably reduced by pre-tensioning. The share of tower wall uplift loads taken by the rods can be based on an estimate of the relative stiffnesses of the rod and the loaded volume of the concrete, assuming a dispersion angle of about 30° in the radial direction. The screwed rods should be sheathed, so that the pre-tension is applied over the full length.

Tower doorways

A doorway is required for access at or near the tower base, and additional doorways are sometimes required for a transformer in the tower base or for maintenance access to the blade tip mechanism. Often they have vertical sides with semi-circular ends at top and bottom. Vertical stiffeners have to be provided as standard down each side to compensate for the missing section of wall and to resist compression buckling, but attention has to be paid to the weld detail at the stiffener ends, where stress concentration due to the opening is likely to be an additional factor.

The weld detail at the stiffener end can be eliminated by reinforcing the inside edge of the doorway with a continuous flange all the way round. The detail category of the flange to tower wall butt weld under transverse loading is then 71, but there is no stress concentration factor to contend with at the top and bottom of the doorway.

7.9.4 Steel lattice towers

Steel lattice towers are usually assembled from angle sections, with bolting used for attaching the bracing members to the legs and splicing the leg sections together. Typically the towers are square in plan with four legs, facilitating the attachment of the bracing members.

One of the advantages of lattice towers is that material savings can be obtained by splaying the legs widely apart at the base, without jeopardising stability or posing transport problems. The latitude for doing this higher up is limited by tip clearance considerations, so waisted tower designs are common. A more elegant tower design results if the legs are rolled to a gentle concave curve, however.

The loads in the legs (or 'chords') result from the tower bending moments, while the loads in the bracing (or 'web') members result from a combination of tower shear and torsional loads. In each case member buckling under extreme loads has to be considered, and fatigue loading

at the joints. Two devices are employed to improve member stability – the web members are arranged as pairs of intersecting diagonals rather than adopting a single triangulated system, so that the tension diagonal can stabilise the compression diagonal at each intersection, and the web/chord intersection points on either side of each chord member are staggered vertically to reduce the spacing of chord supports restraining flexure about the minor axis. Note that care with detailing is needed at the waist, if present, in order to ensure adequate lateral restraint for the chords at the change of direction.

Fatigue loading of bolts is avoided by the use of friction grip bolts. Accordingly galvanising is normally used for corrosion protection rather than painting, in order to achieve an adequate coefficient of friction.

7.10 Foundations

The design of wind turbine foundations is largely driven by the tower base overturning moment under extreme wind conditions. A variety of slab, multi-pile and monopile solutions have been adopted for tubular towers, and these are discussed in turn below.

7.10.1 Slab foundations

Slab foundations are chosen when competent material exists within a few metres of the surface. The overturning moment is resisted by an eccentric reaction to the weight of the turbine, tower, foundation and overburden (allowing for buoyancy, if the water table can rise above the base of the slab). The eccentricity of the reaction, and hence the magnitude of the restoring moment, is limited by the load carrying capacity of the sub-strata, which determines the width of the area at the edge of the slab required to carry the gravity loads. Brinch Hansen (1970) provides straightforward rules for calculating the slab bearing capacity under these conditions, based on the simplifying assumption of uniform loading over the loaded area. However, if the substrata behave elastically, tilting of the slab foundation is likely to result in a linear distribution of bearing stress over the loaded area, so an alternative approach is to base the design on the maximum rather than the average value. The GL rules additionally require that positive bearing stress exists over the whole width of the foundation when the turbine is operating, which limits the maximum overturning moment under these conditions to $WB/6$, where W is the gravity load and B is the slab width. This requirement can add significantly to the required foundation size.

Four alternative slab foundation arrangements are shown in Figure 7.51. Figure 7.51(a) shows a slab of uniform thickness, with its upper surface just above ground level, which is chosen when bedrock is near the ground surface. The main reinforcement consists of top and bottom mats to resist slab bending and the slab is made thick enough for shear reinforcement not to be required. The second variant shown in Figure 7.51(b) is a slab surmounted by a pedestal. This is used when the bedrock is at a greater depth than the slab thickness required to resist the slab bending moments and shear loads. The gravity load on the substrata is increased by virtue of the overburden, so the overall slab plan dimensions can be reduced somewhat.

The third variant, shown in Figure 7.51(c) is similar to the second, but embodies two possible modifications which can be applied independently – replacement of the pedestal by a stub tower embedded in the slab and introduction of a tapering slab depth. The stub tower has to be perforated near the top of the slab to allow radial top face reinforcement to pass through it, and reinforcement to resist punching shear loads from the tower stub bottom flange must

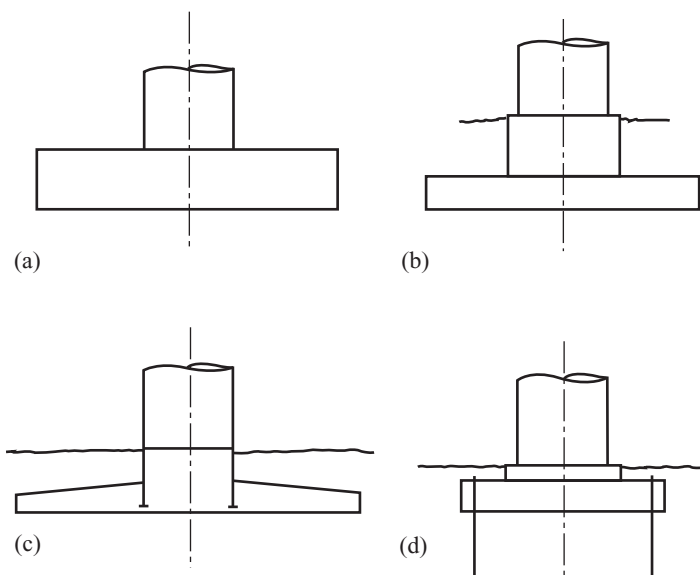


Figure 7.51 (a) Plain slab; (b) Slab and pedestal; (c) Stub tower embedded in tapered slab; (d) Slab held down by rock anchors

be incorporated. Tapering the slab depth has the merit of saving material, but is slightly more difficult to execute.

Rock anchors eliminate the need to add weight to a gravity foundation for counterbalance purposes, and thus enable the foundation size to be significantly reduced, provided bearing capacities are sufficiently high. See Figure 7.51(d). Specialist contractors are needed for rock anchor installation, so they only find occasional use.

The ideal shape of gravity foundation in plan is a circle, but in view of the complications of providing circular formwork, an octagonal shape is usually chosen instead. Sometimes slabs are square in plan to simplify the shuttering and reinforcement further.

7.10.2 Multi-pile foundations

In weaker ground, a piled foundation often makes more efficient use of materials than a slab. Figure 7.52(a) illustrates a foundation consisting of a pile cap resting on eight cylindrical piles arranged in a circle. Overturning is resisted by both pile vertical and lateral loads, the latter being generated by moments applied to the head of each pile. Consequently the reinforcement must be arranged to provide full moment continuity between the piles and the pile cap. Holes for the piles can be auger drilled and the piles cast in situ after the positioning of the reinforcement cage.

7.10.3 Concrete monopile foundations

A concrete monopile foundation consists of a single large diameter concrete cylinder, which resists overturning by mobilising soil lateral loads alone. See Figure 7.52(b). These lateral loads can be calculated conservatively for sand by using either simple Rankine theory for

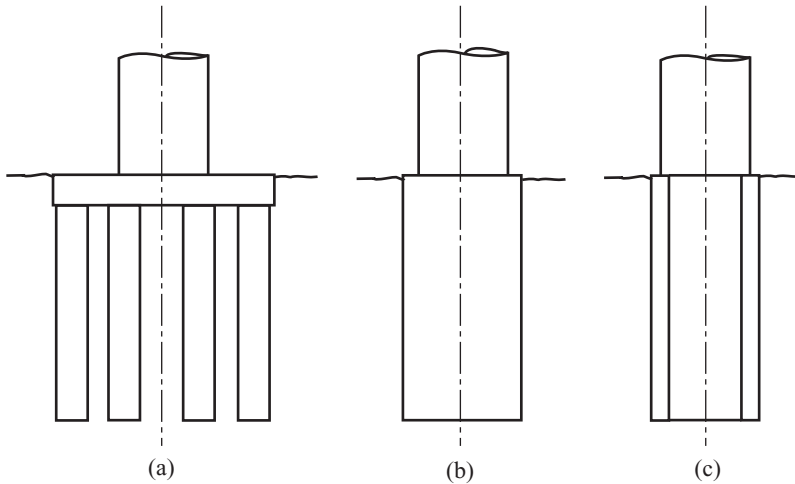


Figure 7.52 (a) Pile group and cap; (b) Solid monopile; (c) Hollow monopile

passive pressures on retaining walls, which ignores soil/wall friction, or Coulomb theory, which includes it. However, in the case of a monopile, friction on the sides of the soil wedge notionally displaced when the pile begins to tilt provides further resistance, and this is accounted for in the solution due to Brinch Hansen (1961).

This type of foundation is an attractive option when the water table is low and the soil properties enable a deep hole to be excavated from above without the sides caving in. However, while simple, the concept is relatively expensive in terms of materials.

The hollow cylinder variant illustrated in Figure 7.52(c) uses materials much less extravagantly by replacing the concrete in the body of the cylinder, which has no structural role to play, with fill.

7.10.4 Foundations for steel lattice towers

The legs of steel lattice towers are relatively widely spaced, and lend themselves to separate foundations. Bored cast in-situ piles are commonly used – see Figure 7.53. The mechanism for resisting overturning is simply uplift and downthrust on the piles, but the piles must also be designed for the bending moments induced by the horizontal shear load. Pile uplift is resisted by friction on the surface of the piles, which depends on both the soil/pile friction angle and the lateral soil pressure. Considerable uncertainty surrounds the magnitude of these quantities, so Eurocode 7 recommends the use of pile testing to establish pile capacity.

The angle sections forming the base of the tower legs are cast in place when the concrete for the piles is poured. A framework is assembled in advance, incorporating the leg base sections, so that the legs can be set at the correct spacing and inclination before concreting.

7.10.5 Foundation rotational stiffness

The assessment of foundation rotational stiffness is an important part of the design process because of the effect it has on tower natural frequency, and hence on fatigue loading. Figure 7.54 illustrates the effect of varying the foundation rotational stiffness for a tower supporting

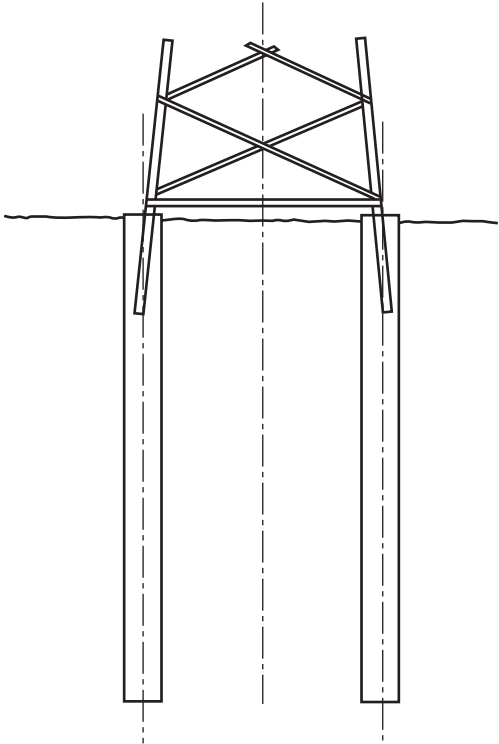


Figure 7.53 Piled foundation for steel lattice tower,

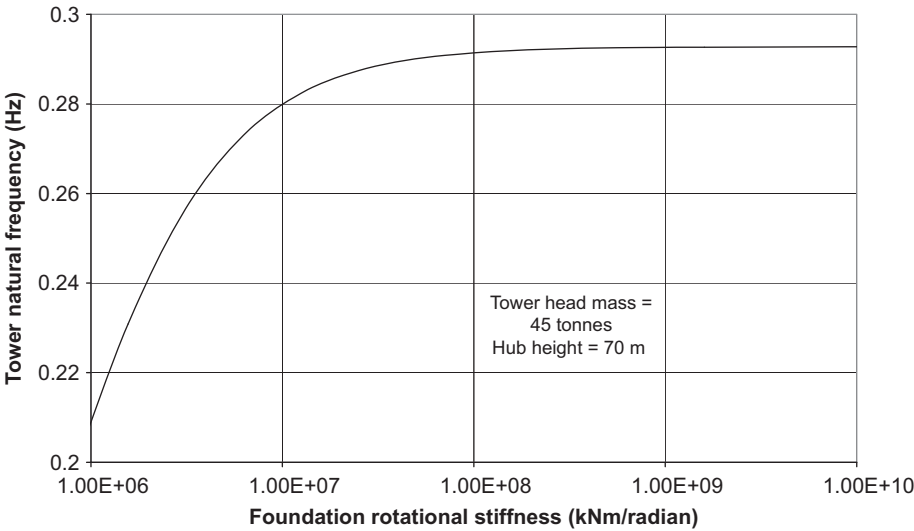


Figure 7.54 Example of variation of tower natural frequency with foundation rotational stiffness

a 45 tonne turbine at 70 m hub height. Manufacturers normally specify a minimum foundation rotational stiffness to ensure that the tower natural frequency is high enough for the fatigue loadings on which the tower design is based to be valid. It is then the task of the foundation designer to ensure that the foundation footprint (or depth, in the case of a monopile foundation) is sufficiently large to achieve this rotational stiffness.

A closed form solution exists for the rotational stiffness, K_θ of a rigid disc resting on an elastic half space, as follows:

$$K_\theta = \frac{8GR^3}{3(1-\nu)} \quad (7.75)$$

where G is the shear modulus of the soil, R is the disc radius and ν is Poisson's ratio. The DNV/Risø 'Guidelines for design of wind turbines' (2002) give modified versions of this formula that account for foundation embedment and soil layers with different shear moduli.

Tower base rotation will be increased by flexibility of the foundation itself and this may need to be accounted for as well.

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8

The controller

In the most general sense, the wind turbine control system consists of a number of sensors, a number of actuators and a system consisting of hardware and software which processes the input signals from the sensors and generates output signals for the actuators.

The sensors might include, for example:

- an anemometer;
- a wind vane;
- at least one rotor speed sensor;
- an electrical power sensor;
- accelerometers;
- load sensors;
- pitch position sensors;
- various limit switches;
- vibration sensors;
- temperature and oil level indicators;
- hydraulic pressure sensors;
- operator switches, push buttons, etc.

The actuators might include hydraulic or electric pitch actuators, an electrical generator which can be considered to be a torque actuator, generator contactors, switches for activating shaft brakes, yaw motors etc.

The system that processes the inputs to generate outputs usually consists of a computer or microprocessor based controller which carries out the normal control functions needed to operate the turbine, supplemented by a highly reliable hard-wired safety system. The safety system must be capable of overriding the normal controller in order to bring the turbine to a safe state if a serious problem occurs.

8.1 Functions of the wind turbine controller

8.1.1 Supervisory control

Supervisory control can be considered as the means whereby the turbine is brought from one operational state to another. The operational states might, for example, include the following:

- standby, when the turbine is available to run if external conditions permit;
- start-up;
- power production;
- shut-down;
- stopped with fault.

It is possible to envisage other states, or it may be useful to further subdivide some of these states. As well as deciding when to initiate a switch from one state to another, the supervisory controller will carry out the sequence control required. As an example, the sequence control for start-up of a pitch regulated wind turbine might consist of the following steps:

- Power up the pitch actuators and other subsystems.
- Release the shaft brake.
- Ramp the pitch position demand at a fixed rate to some starting pitch.
- Wait until the rotor speed exceeds a certain small value.
- Engage the closed loop pitch control of speed.
- Ramp the speed demand up to the generator minimum speed.
- Wait until the speed has been close to the target speed for a specified time.
- Close the generator contactors.
- Engage power or torque controller.
- Ramp the power/torque/speed set points up to the rated level.

The supervisory controller must check that each stage is successfully completed before moving on to the next. If any stage is not completed within a certain time, or if any faults are detected, the supervisory controller should change to shut-down mode.

8.1.2 Closed loop control

The closed loop controller is usually a software-based system that automatically adjusts the operational state of the turbine in order to keep it on some pre-defined operating curve or characteristic. Some examples of such control loops are:

- Control of blade pitch in order to regulate the power output or rotational speed of the turbine to a fixed or slowly-varying set point (for example, the rated level in above-rated wind speeds, or a predetermined speed ramp during start-up or shut-down of the turbine).
- Control of generator torque in order to regulate the rotational speed of a variable speed turbine.
- Control of yaw motors in order to minimise the yaw tracking error.

Some of these control loops may require very fast response in order to prevent the turbine wandering far from its correct operating curve. Such controllers may need to be designed very carefully if good performance is to be achieved without detrimental effects on other aspects of the turbine's operation. Others, such as yaw control, are typically rather slow acting, and careful design is then much less critical.

This chapter examines the main issues behind closed loop controller design, and presents some of the techniques that can be used to effect a successful design.

8.1.3 The safety system

It is helpful to consider the safety system as quite distinct from the main or 'normal' control system of the turbine. Its function is to bring the turbine to a safe condition in the event of a serious or potentially serious problem. This usually means bringing the turbine to rest or to a slow idling speed with the blades feathered and the generator switched off.

The normal wind turbine supervisory controller should be capable of starting and stopping the turbine safely in all foreseeable 'normal' conditions, including extreme winds, loss of the electrical network, and most fault conditions which are detected by the controller. The safety system acts as a backup to the main control system, and takes over if the main system appears to be failing to do this. It may also be activated by an operator-controlled emergency stop button.

Thus, the safety system must be independent from the main control system as far as possible, and must be designed to be fail-safe and highly reliable. Rather than utilising any form of computer or microprocessor based logic, the safety system would normally consist of a hard-wired fail-safe circuit linking a number of normally open relay contacts that are held closed when all is healthy. Then if any one of those contacts is lost, the safety system trips, causing the appropriate fail-safe actions to operate. This might include disconnecting all electrical systems from the supply and allowing fail-safe pitching to the feather position, for example.

The safety system might, for example, be tripped by any one of the following:

- Rotor overspeed, that is, reaching the hardware overspeed limit. This is set higher than the software overspeed limit, which would cause the normal supervisory controller to initiate a shut-down. Figure 8.1 shows a typical arrangement of rotor speed sensing equipment on the low speed shaft.

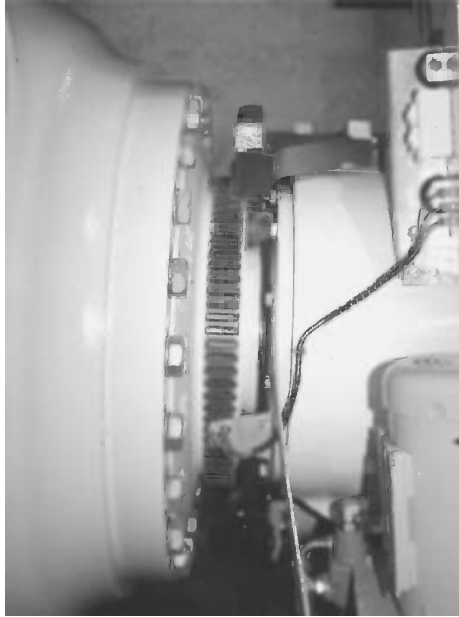


Figure 8.1 Low speed shaft sensing system. Three proximity sensors mounted on a bracket attached to the front of the (integrated) gearbox register the passage of the teeth on the shaft circumference, and provide an independent speed signal for the control and safety systems. The flange onto which the hub is bolted is immediately to the left of the teeth

- Vibration sensor trip, which might indicate that a major structural failure has occurred.
- Controller watchdog timer expired: the controller should have a watchdog timer which it resets every controller timestep. If it is not reset within this time, this indicates that the controller is faulty and the safety system should shut down the turbine.
- Emergency stop button pressed by an operator.
- Other faults indicating that the main controller might not be able to control the turbine.

In some cases the safety system may involve more than one circuit. For example, any safety system trip would normally cause the blades to pitch, but it may be feasible for the relay which disconnects the generator system to be on a different circuit which omits certain sensors, so that in the event of certain faults unrelated to the electrical system the braking action of the generator can be maintained to assist the shut-down.

8.2 Closed loop control: issues and objectives

8.2.1 Pitch control (See also Chapter 3, Section 3.13 and Chapter 6, Section 6.7.2)

Pitch control is the most common means of controlling the aerodynamic power generated by the turbine rotor. Pitch control also has a major effect on all the aerodynamic loads generated by the rotor.

Below rated wind speed, the turbine should simply be trying to produce as much power as possible, so there is generally no need to vary the pitch angle since the optimum pitch angle does not change much with wind speed. The aerodynamic loads below rated wind speed are generally lower than above rated, so again there is no need to modulate these using pitch control, although some pitch action to reduce fatigue loads is possible as explained below. However, for turbines operating below rated at constant speed, the optimum pitch angle for aerodynamic efficiency varies slightly with tip speed ratio, and therefore with wind speed. In this case the pitch angle can be varied slowly (by no more than a few degrees) to maintain optimum power production as the mean wind speed changes. This applies also to variable speed turbines when operating on a constant-speed portion of the operating curve.

Above rated wind speed, pitch control provides a very effective means of regulating the aerodynamic power and loads produced by the rotor so that design limits are not exceeded. In order to achieve good regulation, however, the pitch control needs to respond very rapidly to changing conditions. This highly active control action needs very careful design as it interacts strongly with the turbine dynamics.

One of the strongest interactions is with the tower dynamics. As the blades pitch to regulate the aerodynamic torque, the aerodynamic thrust on the rotor also changes substantially, and this feeds into the tower vibration. As the wind increases above rated, the pitch angle increases to maintain constant torque, but the rotor thrust decreases. This allows the downwind tower deflection to decrease, and as the tower top moves upwind the relative wind speed seen by the rotor increases. The aerodynamic torque increases further, causing more pitch action. Clearly if the pitch controller gain is too high this positive feedback can result in instability. It is therefore vital to take the tower dynamics into account when designing a pitch controller.

Below rated wind speed, the pitch setting should be at its optimum value to give maximum power. It follows that when the wind speed rises above rated, either an increase or a decrease in pitch angle will result in a reduction in torque. An increase in pitch angle, defined as turning the leading edge into wind, reduces the torque by decreasing the angle of attack and hence the lift. This is known as pitching towards feather. A decrease in pitch, that is, turning the leading edge downwind, reduces the torque by increasing the angle of attack towards stall, where the lift starts to decrease and the drag increases. This is known as pitching towards stall.

Although pitching towards feather is the more common strategy, some turbines pitch towards stall. This is commonly known as active stall or assisted stall (see Chapter 6, Section 6.7.4). Pitching to feather requires much more dynamic pitch activity than pitching to stall: once a large part of the blade is in stall, very small pitch movements suffice to control the torque. Pitching to stall results in significantly greater thrust loads because of the increased drag. On the other hand, the thrust is much more constant once the blade is stalled, so thrust-driven fatigue loads may well be smaller.

A further problem with pitching to stall is that the lift curve slope at the start of the stalled region is negative, that is, the lift coefficient decreases with increasing angle of attack. This results in negative aerodynamic damping, which can cause instability of the blade bending modes, both in-plane and out-of-plane. This can be a problem also with fixed pitch stall-regulated turbines.

Most pitch controlled turbines use full-span pitch control, in which the pitch bearing is close to the hub. It is also possible, though not common, to achieve aerodynamic control by pitching only the blade tips, or by using ailerons, flaps, air-jets or other devices to modify the aerodynamic properties. These strategies will result in most of the blade being stalled in high

winds. If only the blade tips are pitched, it may be difficult to fit a suitable actuator into the outboard portion of the blade, and accessibility for maintenance is problematic.

8.2.2 Stall control

Many smaller and older turbines are stall-regulated, which means that the blades are designed to stall in high winds without any pitch action being required. This means that pitch actuators are not required, although some means of aerodynamic braking is likely to be required, if only for emergencies (see Chapter 6, Section 6.8.2).

In order to achieve stall regulation at reasonable wind speeds, the turbine must operate closer to stall than its pitch regulated counterpart, resulting in lower aerodynamic efficiency below rated. This disadvantage may be mitigated in a variable speed turbine, when the rotor speed can be varied below rated in order to maintain peak power coefficient.

In order for the turbine to stall rather than accelerate in high winds, the rotor speed must be restrained. In a fixed speed turbine the rotor speed is restrained by the generator, and is linked to the network frequency, as long as the torque remains below the pull-out torque. In a variable speed turbine, the speed is maintained by ensuring that the generator torque is varied to match the aerodynamic torque. A variable speed turbine offers the possibility to slow the rotor down in high winds in order to bring it into stall. This means that the turbine can operate further from the stall point in low winds, resulting in higher aerodynamic efficiency. However, this strategy means that when a gust hits the turbine, the load torque not only has to rise to match the wind torque but also has to increase further in order to slow the rotor down into stall. This removes one of the main advantages of variable speed operation, namely that it allows very smooth control of torque and power above rated.

The benefits of pitch control as a means of braking mean that stall control is now rarely used for large commercial turbines.

8.2.3 Generator torque control (see also Chapter 6, Section 6.9 and Chapter 7, Section 7.5)

The torque developed by a fixed speed (i.e. directly connected) induction generator is determined purely by the slip speed. As the aerodynamic torque varies, the rotor speed varies by a very small amount such that the generator torque changes to match the aerodynamic torque. The generator torque cannot, therefore, be actively controlled.

However, if a frequency converter is interposed between the generator and the network, the generator speed will be able to vary. The frequency converter can be actively controlled to maintain constant generator torque or power output above rated wind speed. Below rated, the torque can be controlled to any desired value, for example with the aim of varying the rotor speed to maintain maximum aerodynamic efficiency.

There are several means of achieving variable speed operation. One is to connect the generator stator to the network through a frequency converter, which must then be rated for the full power output of the turbine. An alternative and very common arrangement is the doubly-fed induction generator, a wound-rotor machine in which the stator is connected directly to the network and the rotor is connected to the network through slip rings and a frequency converter. This means that the frequency converter need only be rated to handle a fraction of the total power, although the larger this fraction, the larger the achievable speed range will be.

A special case is the variable slip induction generator, where active control of a resistance in series with the rotor windings allows the torque/speed relationship to be modified. By means of closed loop control based on measured currents, it is possible to maintain constant torque above rated, effectively allowing variable speed operation in this region. Below rated it behaves just like a normal induction generator (Bossanyi *et al.*, 1991; Pedersen, 1995).

8.2.4 Yaw control

Turbines, whether upwind or downwind, can be designed to be stable in yaw (see also Chapter 4, Section 4.11), in the sense that if the nacelle is free to yaw, the turbine will naturally remain pointing into the wind. However, it may not point exactly into wind, in which case some active control of the nacelle angle may be needed to maximise the energy capture. Since a yaw drive is usually required anyway, for example for start-up and for unwinding the pendant cable, it may as well be used for active yaw tracking. Free yaw has the advantage that it does not generate any yaw moments at the yaw bearing. However, it is usually necessary to have at least some yaw damping, in which case there will be a yaw moment at the bearing.

In practice, most turbines use active yaw control. A yaw error signal from the nacelle-mounted wind vane is then used to calculate a demand signal for the yaw actuator. Frequently the demand signal will simply be a command to yaw at a slow fixed rate in one or the other direction. The yaw vane signal must be heavily averaged, especially for upwind turbines where the vane is behind the rotor. Because of the slow response of the yaw control system, a simple dead-band controller is often sufficient. The yaw motor is switched on when the averaged yaw error exceeds a certain value, and switched off again after a certain time or when the nacelle has moved through a certain angle. A yaw brake is usually applied when the turbine is not yawing, and often even while yawing, to prevent frequent load reversals at the yaw pinion due to the highly variable nature of the yawing moments.

More complex control algorithms are sometimes used, but the control is always slow-acting, and does not demand any special closed-loop design analysis; in fact rapid yawing is unnecessary, and can generate large gyroscopic loads. Because of this, yaw control is often classed as part of the supervisory controller; also because it remains active in standby mode to keep the turbine pointing into wind (except in very low winds when the wind direction becomes too variable).

One exception is the case of active yaw control to regulate aerodynamic power in high winds, as used on the variable speed Gamma 60 turbine referred to in Section 6.7.5, Chapter 6. This clearly requires very rapid yaw rates, and results in large yaw loads and gyroscopic and asymmetric aerodynamic loads on the rotor. This method of power regulation would be too slow for a fixed speed turbine, and even on the Gamma 60 the speed excursions during above-rated operation were quite large.

Instead of a yaw actuator, it is possible to use individual pitch control to generate a yawing moment – see Section 8.3.14.

8.2.5 Influence of the controller on loads

As well as regulating the turbine power in high winds and optimising it in low winds, it is clear that the action of the control system can have a major impact on the loads experienced by the turbine. The design of the controller must take into account the effect on loads, and

at least ensure that excessive loads will not result from the control action. It is possible to go further than this, and explicitly design the controller with the reduction of certain loads as an additional objective.

The reduction of certain loads is clearly compatible with the primary objective of limiting power in high winds. For example, the limitation of power output is clearly compatible with limitation of gearbox torque. In other cases however, there may be a conflict, in which case the controller design is bound to be a compromise involving a trade-off between competing goals. For example, there is a clear trade-off between good control of power output and pitch actuator loads. The more actuator activity can be tolerated, the better the power control can be. Of course it is always possible to reduce loads by reducing energy capture (after all the loads are minimised with the turbine switched off), but economic optimisation generally implies that reduced capital cost due to reduced loading is often only justified if it causes very little or no loss of energy production.

The interaction between pitch control and tower vibration referred to in Section 8.2.1 is another important example, since the amount of tower vibration has a major effect on tower base loads. The tighter the control of rotor speed by means of pitch control, the greater the tower vibration is likely to be. Blade, hub and other structural loads will also be influenced by pitch control activity. Generator torque control can have a major impact on gearbox loads, as described below.

8.2.6 Defining controller objectives

The primary objective of the closed loop controller can usually be stated fairly simply. For example, the primary objective of the pitch controller may be to limit power or rotor speed in high winds. There may be more than one ‘primary’ objective, as in the case where the pitch controller is also used to optimise energy capture in low winds.

However, since the controller can also have a major effect on structural loads and vibrations, it is vital to consider these when designing the control algorithm. Thus, a fuller description of the pitch controller objectives might be:

- to optimise power production in below-rated wind speeds;
- to regulate or limit aerodynamic torque in above-rated wind speeds;
- to minimise peaks in gearbox torque;
- to avoid excessive pitch activity;
- to minimise tower base loads as far as possible by controlling tower vibration; and
- to avoid exacerbating hub and blade root loads.

Especially with individual pitch control (Section 8.3.12) the last of these should be replaced by a much more positive objective:

- To actively reduce the loading on the rotor and the rest of the system.

Clearly some of these objectives may conflict with others, so the control design process will inevitably involve some degree of trade-off or optimisation. In order to do this, it is necessary to be able to quantify the different objectives. It is usually almost impossible

to do this with any precision, because the various loads may affect not only the costs of different components (sometimes in complex ways) but also their reliability. Even the trade-off between energy capture and component cost is not straightforward, as it will depend on the wind regime, the discount rate, and knowledge of future prices for the sale of electricity. Therefore, some degree of judgement will always be required in arriving at an acceptable controller design.

8.2.7 PI and PID controllers

A brief general description is given here of PI and PID controllers, since they will be referred to a number of times in the subsequent sections.

The proportional and integral (PI) controller is an algorithm which is very widely used for controlling all kinds of equipment and processes. The control action is calculated as the sum of two terms, one proportional to the control error, which is the difference between the desired and actual values of the quantity to be controlled, and one proportional to the integral of the control error. The integral term ensures that in the steady state the control error tends to zero, because if it did not, the control action would continue to increase indefinitely. The proportional term makes the algorithm more responsive to rapid changes in the quantity being controlled.

A differential term is often added, which gives a contribution to the control action proportional to the rate of change of the control error. This is then known as a PID controller. In terms of the Laplace operator s , which can usefully be thought of as a differentiation operator, the PID controller from measured signal x to control signal y can be written as follows:

$$y = \left(K_p + \frac{K_i}{s} + \frac{K_d s}{1 + sT_d} \right) x \quad (8.1)$$

where K_p , K_i and K_d are the proportional, integral and derivative gains respectively. The denominator of the differential term is essentially a low-pass filter, and is needed to ensure that the gain of the algorithm does not increase indefinitely with frequency, which would make the algorithm very sensitive to signal noise. Setting $K_d = 0$ results in a PI controller.

It is often the case that the control action is subject to limits. For example, if the control action represents the blade pitch used to control power above rated, then when the power drops below rated the pitch will be limited to the fine pitch setting, and will not be allowed to drop further. In this situation the integral term of the PI or PID controller will grow more and more negative as the power remains below rated. Then when the wind speed rises again and the power rises above rated, the integral term will start to grow again towards zero, but until it gets close to zero it will more than compensate for the proportional and derivative terms. Therefore, the pitch may remain ‘stuck’ at fine pitch for a considerable time, depending on how long the power has been below rated, until the integral term has come back close to zero. This is known as integrator wind-up, and clearly it must be prevented. This is done in effect by disabling the integrator when the pitch is on the limit. This is known as ‘integrator desaturation’, which is described more fully in Section 8.6.

The design of PI and PID controllers, including the choice of gains, is described in more detail in Section 8.4.

8.3 Closed loop control: general techniques

This section outlines the principles behind many of the types of closed loop controllers to be found in wind turbines. Mathematical methods for designing the closed loop algorithms are covered in Section 8.4.

8.3.1 Control of fixed speed, pitch regulated turbines

A fixed speed pitch regulated turbine usually means a turbine that has an induction generator connected directly to the AC network, and which, therefore, rotates at a nearly constant speed. As the wind speed varies, the power produced will vary roughly as the cube of the wind speed. At rated wind speed, the electrical power generated becomes equal to the rating of the turbine, and the blades are then pitched in order to reduce the aerodynamic efficiency of the rotor and limit the power to the rated value. The usual strategy is to pitch the blades in response to the power error, defined as the difference between the rated power and the actual power being generated, as measured by a power transducer. The primary objective is then to devise a dynamic pitch control algorithm that minimises the power error, although as explained above, this may not be the only objective.

The main elements of the control loop are shown in Figure 8.2. A PI or PID algorithm is often used for the controller.

When the power falls below rated, the pitch demand saturates at the fine pitch limit, maximising the aerodynamic efficiency of the rotor. Since the optimum pitch angle depends on the tip speed ratio, it is possible to increase energy capture below rated by a small percentage if the fine pitch limit is varied in response to the wind speed. The measured power itself is the best available measure of wind speed over the whole turbine (effectively using the whole turbine as an anemometer). However, the fine pitch limit should be varied relatively slowly compared to the control loop dynamics. Good performance can be obtained by changing the fine pitch limit in response to a moving average of the measured power, using the calculated steady-state relationship between power output and optimum pitch angle at each wind speed. The moving average time constant can be quite long since the underlying wind speed (averaged over the rotor swept area) varies relatively slowly. Of course it should be significantly slower than the blade passing frequency and the lowest structural frequency (generally the first tower mode) in order to avoid unnecessary pitch activity below rated.

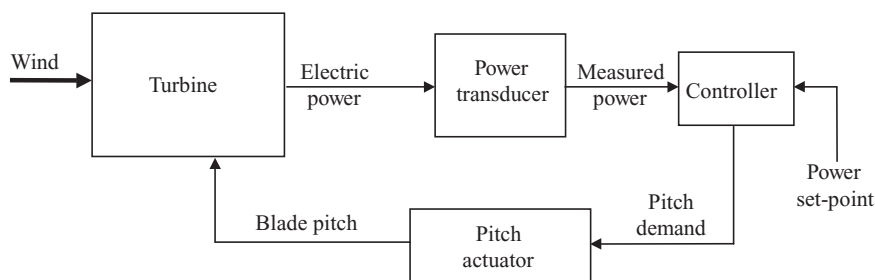


Figure 8.2 Main control loop for a fixed speed pitch regulated turbine

8.3.2 Control of variable speed pitch regulated turbines

A variable speed generator is decoupled from the grid frequency by a power converter, which can control the load torque at the generator directly, so that the speed of the turbine rotor can be allowed to vary between certain limits. An often-quoted advantage of variable speed operation is that below rated wind speed, the rotor speed can be adjusted in proportion to the wind speed so that the optimum tip speed ratio is maintained. At this tip speed ratio the power coefficient, C_p , is a maximum, which means that the aerodynamic power captured by the rotor is maximised. This is often used to suggest that a variable speed turbine can capture much more energy than a fixed speed turbine of the same diameter. In practice it may not be possible to realise all of this gain, partly because of losses in the power converter and partly because it is not possible to track optimum C_p perfectly.

Maximum aerodynamic efficiency is achieved at the optimum tip speed ratio $\lambda = \lambda_{\text{opt}}$, at which the power coefficient C_p has its maximum value $C_{p(\text{max})}$. Since the rotor speed Ω is then proportional to wind speed U , the power increases with U^3 and Ω^3 , and the torque with U^2 and Ω^2 . The aerodynamic torque is given by

$$Q_a = \frac{1}{2} \rho A C_q U^2 R = \frac{1}{2} \rho \pi R^3 \frac{C_p}{\lambda} U^2 \quad (8.2)$$

Since $U = \Omega R / \lambda$ we have

$$Q_a = \frac{1}{2} \rho \pi R^5 \frac{C_p}{\lambda^3} \Omega^2 \quad (8.3)$$

In the steady state, therefore, the optimum tip speed ratio can be maintained by setting the load torque at the generator, Q_g , to balance the aerodynamic torque, that is,

$$Q_g = \frac{1}{2} \frac{\pi \rho R^5 C_p}{\lambda^3 G^3} \omega_g^2 - Q_L \quad (8.4)$$

Here Q_L represents the mechanical torque loss in the drive train (which may itself be a function of rotational speed and torque), referred to the high speed shaft. The generator speed is $\omega_g = G\Omega$, where G is the gearbox ratio.

This torque-speed relationship is shown schematically in Figure 8.3 as the curve B1–C1. Although it represents the steady-state solution for optimum C_p , it can also be used dynamically to control generator torque demand as a function of measured generator speed. In many cases, this is a very benign and satisfactory way of controlling generator torque below rated wind speed.

For tracking peak C_p below rated in a variable speed turbine, the quadratic algorithm of Equation 8.4 works well and gives smooth, stable control. However, in turbulent winds, the large rotor inertia prevents it from changing speed fast enough to follow the wind, so rather than staying on the peak of the C_p curve it will constantly fall off either side, resulting in a lower mean C_p . This problem is clearly worse for heavy rotors, and also if the $C_p - \lambda$ curve has a sharp peak. Thus, in optimising a blade design for variable speed operation, it is not only important to try to maximise the peak C_p , but also to ensure that the $C_p - \lambda$ curve is reasonably flat-topped.

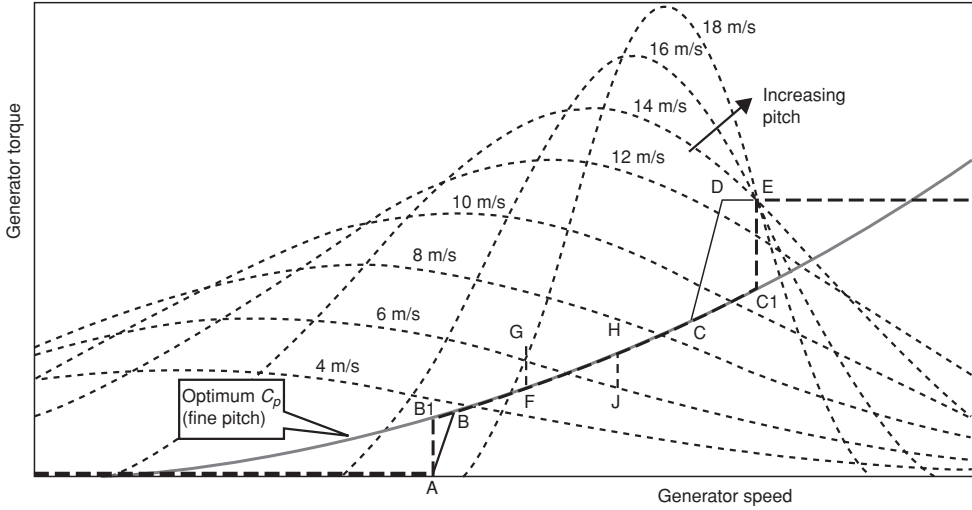


Figure 8.3 Schematic torque-speed curve for a variable speed pitch regulated turbine

It is possible to manipulate the generator torque to cause the rotor speed to change faster when required, so staying closer to the peak of the C_p curve. One way to do this is to modify the torque demand by a term proportional to rotor acceleration (Bossanyi, 1994):

$$Q_g = \frac{1}{2} \frac{\pi \rho R^5 C_p}{\lambda^3 G^3} \omega_g^2 - Q_L - B \dot{\omega}_g \quad (8.5)$$

For a stiff drive train, and ignoring frequency converter dynamics, the torque balance gives:

$$I \dot{\Omega} = Q_a - G Q_g \quad (8.6)$$

where I is the total inertia (of rotor, drive train and generator) and Ω is the rotational speed of the rotor. Hence,

$$(I - G^2 B) \dot{\Omega} = Q_a - \frac{1}{2} \frac{\pi \rho R^5 C_p}{\lambda^3 G^2} \omega_g^2 + G Q_L \quad (8.7)$$

Thus, the effective inertia is reduced from I to $I - G^2 B$, allowing the rotor speed to respond more rapidly to changes in wind speed.

Another possible method is to use available measurements to make an estimate of the wind speed, calculate the rotor speed required for optimum C_p , and then use the generator torque to achieve that speed as rapidly as possible. The aerodynamic torque can be expressed as

$$Q_a = \frac{1}{2} \rho A C_q R U^2 = \frac{1}{2} \rho \pi R^5 \Omega^2 C_q / \lambda^2 \quad (8.8)$$

where R is the turbine radius, Ω the rotational speed, and C_q the torque coefficient. If drive train torsional flexibility is ignored, a simple estimator for the aerodynamic torque is

$$Q_a^* = G Q_g + I \dot{\Omega} = G Q_g + I G \dot{\omega}_g \quad (8.9)$$

where I is the total inertia. A more sophisticated estimator could take into account drive train torsion, etc. From this it is possible to estimate the value of the function $F(\lambda) = C_q(\lambda)/\lambda^2$ as

$$F^*(\lambda) = \frac{Q_a^*}{\frac{1}{2} \rho \pi R^5 (G \omega_g)^2} \quad (8.10)$$

Knowing the function $F(\lambda)$ from steady state aerodynamic analysis, one can then deduce the current estimated tip speed ratio λ^* . The desired generator speed for optimum tip speed ratio can then be calculated as

$$\omega_d = \omega_g \hat{\lambda} / \lambda^* \quad (8.11)$$

where $\hat{\lambda}$ is the optimum tip speed ratio to be tracked. A simple PI controller can then be used, acting on the speed error $\omega_g - \omega_d$, to calculate a generator torque demand which will track ω_d . The higher the gain of PI controller, the better will be the C_p tracking, but at the expense of larger power variations. Simulations for a particular turbine showed that a below-rated energy gain of almost 1% could be achieved, with large but not unacceptable power variations.

Holley *et al.* (1999) demonstrated similar results with a more sophisticated scheme, and also showed that a perfect C_p tracker could capture 3% more energy below rated, but only by demanding huge power swings of plus and minus three to four times rated power, which is totally unacceptable.

Since such large torque variations are required to achieve only a modest increase in power output, it is usual simply to use the simple quadratic law, possibly augmented by some inertia compensation as in Equation 8.5 if the rotor inertia is large enough to justify it.

As turbine diameters increase in relation to the lateral and vertical length scales of turbulence, it becomes more difficult to achieve peak C_p anyway because of the non-uniformity of the wind speed over the rotor swept area. Thus, if one part of a blade is at its optimum angle of attack at some instant, other parts will not be.

In most cases, it is actually not practical to maintain peak C_p from cut-in all the way to rated wind speed. Although some variable speed systems can operate all the way down to zero rotational speed, this is not the case with limited range variable speed systems based on the widely-used doubly-fed induction generators. These systems only need a power converter rated to handle a fraction of the turbine power, which is a major cost saving. This means that in low wind speeds, just above cut-in, it may be necessary to operate at an essentially constant rotational speed, with the tip speed ratio above the optimum value.

At the other end of the range, it is usual to limit the rotational speed to some level, usually determined by aerodynamic noise constraints, which is reached at a wind speed which is still some way below rated. It is then cost-effective to increase to torque demand further, at essentially constant rotational speed, until rated power is reached. Figure 8.3 illustrates some typical torque-speed trajectories, which are explained in more detail below. Turbines designed for noise-insensitive sites may be designed to operate along the optimum- C_p trajectory all the way until rated power is reached. The higher rotational speed implies lower torque and

in-plane loads, but higher out of plane loads, for the same rated power. This strategy might be of interest for offshore wind turbines.

8.3.3 Pitch control for variable speed turbines

Once the rated torque has been reached, no further increase in load torque can occur, so the turbine will start to speed up. Pitch control is then used to regulate the rotor speed, with the load torque held constant. A PI or PID controller is often satisfactory for this application. In some situations it may be useful to include notch filters on the speed error to prevent excessive pitch action at, for example, the blade passing frequency or significant structural resonant frequencies, for example the drive train torsional frequency.

Rather than maintain a constant torque demand while the pitch control is regulating the rotational speed, it is possible to vary the torque demand in inverse proportion to the measured speed in order to keep the power output, rather than the torque, at a constant level. Provided the pitch controller is able to maintain the speed close to the set point, there will be little difference between these two approaches. The reduction of load torque with increasing speed has a slight destabilising effect on the pitch controller, but this is often not serious, and provided the gearbox torque and rotor speed variations are not greatly affected, the constant power approach is attractive from the perspective of power quality.

8.3.4 Switching between torque and pitch control

In practice, acoustic noise, loads or other design constraints usually mean that the maximum allowable rotor speed is reached at a relatively low wind speed. As the wind speed increases further, it is desirable to increase the torque and power without any further speed increase, in order to capture more energy from the wind. The simplest strategy is to implement a torque-speed ramp: line CD in Figure 8.3. Once rated power or torque is reached, pitch control is used to maintain the rotor speed at its rated value. In order to prevent the torque and pitch controllers from interfering with one another, the speed set-point for the pitch controller is set a little higher, at point E in Figure 8.3. If the speed set-point were at D then there would constantly be power dips in above-rated winds, whenever the speed fell transiently below D. Furthermore the pitch controller would act below rated, as the pitch and torque controllers would both be trying to control the speed.

It would be an improvement if the torque-speed trajectory A-B-C-D-E in Figure 8.3 could be changed to A-B1-C1-E. The turbine would then stay close to optimum C_p over a wider range of wind speeds, giving slightly higher energy capture for the same maximum operating speed (Bossanyi, 1994). The vertical sections A-B1 and C1-E can be achieved by using a PI controller for the torque demand, in response to the generator speed error with the set point at A or C1. Transitions between constant speed and optimum C_p operation are conveniently handled by using the optimum- C_p curve as the upper torque limit of the PI controller when operating at A, or the lower limit when at C1. The set point flips between A and C1 when the measured speed crosses the mid-point between A and C1. Despite this step change in set point the transition is completely smooth because the controller will be saturated on the optimum- C_p limit curve both before and after the transition.

This logic can easily be extended to implement ‘speed exclusion zones’, to avoid speeds at which blade passing frequency would excite, for example, the tower resonance, by introducing additional speed set points and some logic for switching between them – see lines FG, HJ

in Figure 8.3. When the torque demand exceeds G for a certain time, the set point ramps smoothly from F to H . Then if it falls below J , the set point ramps back again.

Another advantage of PI control of the torque is that the ‘compliance’ of the system can be controlled. Controlling to a steep ramp (CD in Figure 8.3) can be quite harsh in that the torque demand will be varying rapidly up and down the slope. A PI controller, on the other hand, can be tuned to achieve a desired level of ‘softness’. With high gain, the speed will be tightly controlled to the set point, requiring large torque variations. Lower gains will result in more benign torque variations, while the speed is allowed to vary more around the set point.

In order to use point C1 as the speed set point for both the torque and the pitch controllers, it is necessary to decouple the two. One technique is to arrange some switching logic which ensures that only one of the control loops is active at any one time. Thus, below rated the torque controller is active and the pitch demand is fixed at fine pitch, while above rated the pitch controller is active and the torque demand is fixed at the rated value. This can be done with fairly simple logic, although there will always be occasions when the controller is caught briefly in the ‘wrong’ mode. For example, if the wind is just below rated but rising rapidly, it might be useful to start pitching the blades a little before the torque demand reaches rated. If the pitch does not start moving until the torque reaches rated, it then has to move some way before it starts to control the acceleration, and a small overspeed may result.

A more satisfactory approach is to run both control loops together, but to couple them together with terms which drive one or the other loop into saturation when far above or below the rated wind speed. Thus, most of the time only one of the controllers is active, but they can be made to interact constructively when close to the rated point.

A useful method is to include a torque error term in the pitch PID in addition to the speed error. Above rated, since the torque demand saturates at rated, the torque error will be zero, but below rated it will be negative. An integral term will bias the pitch demand towards fine pitch, preventing the pitch controller from acting in low winds, while a proportional term may help to start the pitch moving a little before the torque reaches rated if the wind speed is rising rapidly.

It is also necessary to prevent the torque demand from dropping when operating well above rated wind speed. Here a useful strategy is a ‘ratchet’ which prevents the torque demand from falling while the pitch is not at fine. This can also smooth over brief lulls in the wind around rated, using the rotor kinetic energy to avoid transient power drops.

An alternative approach is to introduce separate bias terms to the speed errors for the two control loops, effectively modifying the set-points of both loops, which remain active throughout. When the torque is below rated, the pitch controller sees a higher speed set-point, forcing the pitch towards fine. As the torque approaches rated, the set-point is reduced to the nominal value so that the pitch control gradually takes over. As the pitch rises above fine, the torque controller set point is pushed down, forcing the torque up to the rated power limit. As the pitch comes down again the torque controller set-point rises back to the nominal value, allowing the torque controller to resume its duties by the time fine pitch is reached, and as the torque falls further the pitch controller set point rises again to keep the pitch at the fine limit. The movement of the set-points is decoupled from the control loop dynamics by introducing first-order lags with appropriate time constants. Shorter time constants are appropriate for rising set-points than for falling set-points. This helps prevent overspeeds, and also prevents the pitch angle from falling too sharply during a temporary wind lull which could cause unnecessary tower vibration.

8.3.5 Control of tower vibration

For both fixed and variable speed machines the influence of the pitch controller on tower vibration and loading, described in Section 8.2.1, is one of the major constraints on the design of the control algorithm. The first tower fore-aft vibrational mode is essentially very lightly damped, exhibiting a strong resonant response which can be maintained at quite a high level even by a small amount of excitation which is naturally present in the wind. The strength of the response depends critically on the small amount of damping which is present, mostly aerodynamic damping from the rotor. The pitch control action modifies the effective damping of that mode. In designing the pitch controller, it is therefore important to avoid further reducing the already small level of damping, and if possible to increase it.

The design of control algorithms is covered in Section 8.4. This includes the choice of PID gains, as well as the addition of further terms to the controller which modify the overall dynamics in such a way as to help increase the tower damping. The use of modern control methods such as optimal state feedback is also discussed. This technique can help to achieve a suitable compromise between the competing objectives of speed or power control (achieved by regulating the in-plane loading) and tower vibration control (which depends on modifying the out-of-plane loading).

There is, however, only a certain amount of information in the measured speed or power signal. State estimators such as Kalman filters (Section 8.4.5) can be used to try to distinguish between the effects of wind speed changes and tower motion on the measured signal. However, it is also possible to enhance the information available to the controller by using an accelerometer mounted in the nacelle, which provides a very direct measure of tower fore-aft motion. By using this extra signal, it is in fact possible to reduce tower loads significantly without adversely affecting the quality of speed or power regulation.

The tower dynamics can be modelled approximately as a second order system exhibiting damped simple harmonic motion, that is,

$$M\ddot{x} + D\dot{x} + Kx = F + \Delta F \quad (8.12)$$

where x is tower displacement and F is the applied force, which in this case is predominantly the rotor thrust. ΔF is the additional thrust caused by pitch action. We can equate M with the tower modal mass and K with the modal stiffness, such that the tower frequency is $\sqrt{K/M}$ rad/s. The damping term D is small. The effective damping can clearly be increased if ΔF is proportional to $-\dot{x}$. Clearly it is easier to measure acceleration than velocity, so the tower acceleration would have to be integrated to provide a measure of $\dot{x}$. A suitable gain for ΔF can be estimated from the partial derivative from pitch to thrust, $\partial F/\partial \beta$ where β is the pitch angle, in order to achieve any particular additional damping D_p :

$$\begin{aligned} \delta F &= \frac{\partial F}{\partial \beta} \delta \beta = -D_p \dot{x} \\ \delta \beta &= \frac{-D_p}{\partial F/\partial \beta} \dot{x} \end{aligned} \quad (8.13)$$

It may sometimes be necessary to place a notch filter in series with this feedback term to prevent unwanted feedback from other components of tower acceleration, for example at blade passing frequency. Lead-lag or other loop-shaping filters may also help to adjust the

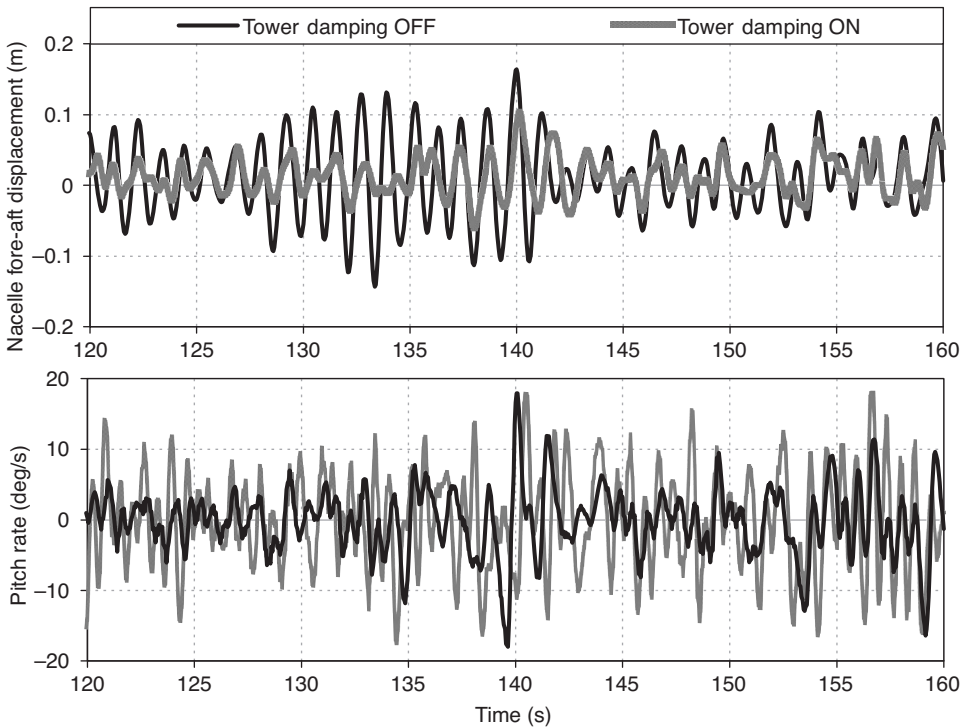


Figure 8.4 Use of a tower accelerometer to help control tower vibration

phase of the feedback to ensure maximum damping, taking into account the full dynamics of the system which are actually more complex than Equation 8.12. For example, the dynamic response of the pitch actuator should be taken into account, as well as other modes of vibration which couple to the tower dynamics. Figure 8.4 shows the results of a simulation with and without such an acceleration feedback term, in combination with a PID controller to control rotational speed. The simulations were driven with a realistic three-dimensional turbulent wind input. The speed control was hardly affected, and although there is a significant increase in pitch actuator activity, the additional pitch rates required are modest. Clearly this technique is capable of increasing the tower damping substantially, almost eliminating the resonant response and significantly reducing tower base loads. Although it requires an accelerometer, this is usually present anyway to trigger a shut-down in the event of excessive vibration. Accelerometers are also relatively cheap, robust and reliable devices.

Field test results demonstrating the effectiveness of this tower damping action have been published by Rossetti *et al.* (2004) and Bossanyi *et al.* (2010).

Because rotor thrust varies rapidly with pitch angle close to fine pitch, a common cause of fore-aft tower excitation is the case of short-lived lull in wind speed starting a little above rated wind speed. The pitch responds by falling rapidly towards fine pitch, causing a rapid thrust reduction and large consequent tower vibration. This can be avoided by preventing the rapid decrease in pitch: if the wind picks up again quickly, as is often the case, little energy is lost. Various algorithms can be used for this. Simply limiting the negative pitch rate when

close to fine pitch is a possibility, although this will generally reduce the effectiveness of speed regulation in this region, and the asymmetrical rate limits will result in loss of energy (restoring symmetry by similarly decreasing the positive rate limit is not advisable, as transient overspeeds will result). These disadvantages can be mitigated to some extent by limiting the downward pitch rate only after a large negative rate has already been sustained for a short time. However, a particularly effective technique is to use a dynamically varying fine pitch angle: whenever the pitch angle is above fine pitch, the fine pitch limit is increased, but always staying below the actual pitch by at least a certain margin, so that it is always well below the actual pitch in high winds. The dynamic fine pitch is allowed to decrease only slowly, so during a wind lull the pitch may decrease to the level of the dynamic fine pitch, which then continues to fall slowly to ensure that the true fine pitch is reached if the lull is prolonged. The generator torque control can act while the pitch is on the dynamic fine pitch limit, preventing any significant reduction in rotor speed during a short wind lull.

Tower loading is generally dominated by the fore-aft vibration, but side-side vibration can be significant in some situations, for example in offshore turbines operating during periods of wind-wave misalignment. The side-side vibration is even more lightly damped than fore-aft, because there is even less aerodynamic damping from the rotor in this direction. When the principal wave direction is coming from the side, therefore, the excitation of the side-side vibrations can be considerable, and may even become more significant than the fore-aft vibration at such times. In principle, the damping of the side-side vibration can be increased by appropriate control of the generator torque, adding a component of torque demand derived from the measured side-side acceleration, in addition to the torsional damping described in the next section (Markou *et al.*, 2009; Fischer, 2010).

8.3.6 Control of drive train torsional vibration

A typical drive train can be considered to consist of a large rotor inertia and a (smaller) high speed shaft inertia (mainly the generator and brake disc), separated by a torsional spring which represents twisting of shafts and couplings, bending of gear teeth and deflection of any soft mountings. It is important to consider also the coupling of the torsional mode of vibration with the first rotor in-plane collective mode, in which case the drive train can be approximated by three inertias and two torsional springs (Ramtharan *et al.*, 2007). In some cases the coupling to the second tower side-to-side mode, which has a lot of rotation at the tower top, is also important.

In a fixed speed turbine, the induction generator slip curve (Section 7.5) essentially acts like a strong damper, with the torque increasing rapidly with speed – see Figure 7.29, Chapter 7. Therefore, the torsional mode of the drive train is well damped and generally does not cause a problem. In a variable speed turbine operating at constant generator torque however, there is very little damping for this mode, since the torque no longer varies with generator speed. The aerodynamic damping due to the rotor is small because the blades are vibrating in the in-plane direction. There is a small amount of structural damping in shafts and couplings, and some damping from the gearbox, but these effects contribute typically only a small fraction of 1% of critical damping. The very low damping can lead to large torque oscillations at the gearbox, effectively negating one of the principal advantages of variable speed operation, the ability to control the torque.

Although it may be possible to provide some further damping mechanically, for example by means of appropriately designed rubber mounts or couplings, it is difficult to provide

enough damping and there is a cost associated with this. A widely-used solution, which has been successfully adopted on many variable speed turbines, is to modify the generator torque control to provide some damping. Instead of demanding a constant generator torque above rated, (or a torque varying slightly in inverse proportion to speed in the case of the constant power algorithm described in Section 8.3.3), a small ripple at the drive train frequency is added on to this basic torque demand, with the phase adjusted to counteract the effect of the resonance and effectively increase the damping. A band-pass filter of the form

$$G \frac{2\zeta\omega s(1 + s\tau)}{s^2 + 2\zeta\omega s + \omega^2} \quad (8.14)$$

(where G is a gain) acting on the measured generator speed can be used to generate this additional ripple. The frequency ω must be close to the resonant frequency which is to be damped. The time constant τ modifies the phase and contributes more of a high-pass filter characteristic, and can sometimes be used to compensate for time lags or other dynamics in the system. A root locus plot (Section 8.4) is very useful for tuning the filter parameters.

Although a very effective filter can be made by tuning it to give a frequency response with a very broad peak (large ζ), this may be detrimental to the overall performance in that more low frequency variations in torque and power are then introduced. Even with a narrow peak, there can be sufficient response at multiples of blade passing frequency such as 3P or 6P to disturb the system, in which case a notch filter (Section 8.4) can be cascaded with the filter of Equation 8.14. Of course if the resonant frequency nearly coincides with an excitation frequency such as 6P then the resonance will be very difficult to control because it will be strongly excited.

Figure 8.5 shows some simulation results for a variable speed turbine operating in simulated three-dimensional turbulence. A large drive train resonance can be seen to be building up. Although the power and generator torque are smooth, the gearbox would be very badly affected. The effect of introducing a damping filter as described above is also shown. It almost completely damps out the resonance without increasing the electrical power variations. This is because the torque ripple needed to damp the resonance is actually very small, because the amount of excitation is small.

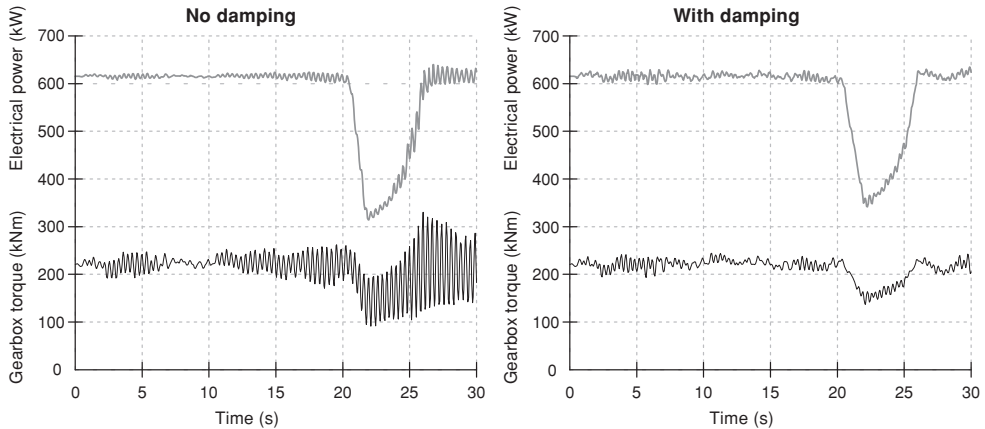


Figure 8.5 Effect of a drive train damping filter

In many cases the drive train damping can be further improved by using an input signal representative of the ‘twisting speed’ rather than just the generator speed: the twisting speed is just the difference between the generator speed and the rotor speed (scaled by gearbox ratio). However, this requires two speed measurements, and the low speed shaft sensor in particular sometimes has insufficient resolution and may need to be upgraded. Care is needed to ensure that the required difference between the two signals is not too susceptible to noise on the signals.

These torsional vibrations are typically much less of a problem on direct drive systems, where in some cases a damping filter of this sort may not even be required at all.

8.3.7 Variable speed stall regulation

Figure 8.6 shows two power curves for the same rotor, one running as a 600 kW fixed speed pitch regulated turbine and one adjusted to run as a fixed speed stall regulated turbine with the same rating. The rotational speed of the stall regulated turbine has been reduced in order to limit the power to the same rated level. Therefore, although the stall regulated turbine generates slightly more energy at very low wind speeds, as the blades approach stall above 8 m/s there is a large loss of output compared to the pitch regulated machine. (In practice of course, if the turbine was designed to operate in stall, the blade design, solidity and rotor speed could be reoptimised, reducing this difference.)

By making use of variable speed, it is quite possible to correct this loss of energy by operating either turbine at the optimum tip speed ratio up to rated, or until the maximum rpm is reached. At rated power, it is then possible to reduce the speed of the rotor to bring it into stall, although this has rarely been done to date on commercial machines. This can be done by closed loop control of the generator torque in response to power error, allowing the turbine to follow exactly the same power curve as the pitch regulated turbine. Thus, the variable speed stall regulated turbine can achieve the same energy output as the variable speed pitch regulated turbine, but without the need for an active pitch mechanism. As explained in Section 8.2.2, however, significant torque and power transients will result from this strategy. The smooth torque and power, which are one of the main advantages of variable speed systems, will therefore not be realised.

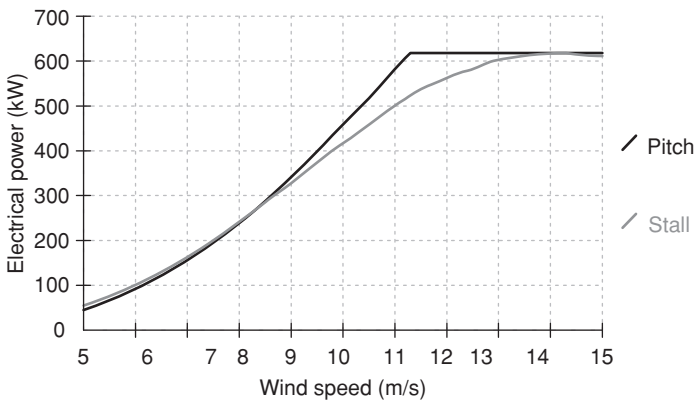


Figure 8.6 Comparison of pitch and stall control

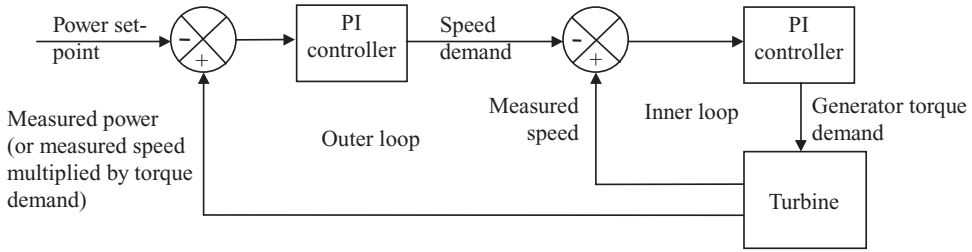


Figure 8.7 A simple control algorithm for variable speed stall regulation

One simple and effective control algorithm for this case is illustrated in Figure 8.7. It consists of two nested loops, an outer power loop which demands a generator speed, and an inner speed loop which demands a generator torque. As in Section 8.3.4, a PI controller can be used for the inner loop. This is the same controller as for sections A-B1 and C1-E of Figure 8.3, making it particularly easy to arrange the transition between control modes at the rated point since the inner loop is always active.

8.3.8 Control of variable slip turbines

The operating envelope for a variable slip generator is shown in Figure 8.8. Note that the slip speed represents the increase in speed above synchronous (conventionally for motors this would be a negative slip). Below rated, the generator acts just like a conventional induction machine, with the torque related to the slip speed according to the slip curve AB. Once point B is reached, a resistor in series with the rotor circuit, previously short-circuited by a semiconductor switch, is progressively brought into play by switching the semiconductor switch on and off at several kHz, and varying the mark-space ratio to change the average resistance. As the average resistance increases, the generator slip curve changes so that its slope varies inversely with the total resistance of the rotor circuit. Figure 8.8 shows a typical example in which the rotor resistance can increase by a factor 10, changing the slip curve from AB to AD. By controlling the resistance, therefore, the generator can operate anywhere within the shaded region. The resistance is usually varied by a closed loop algorithm which

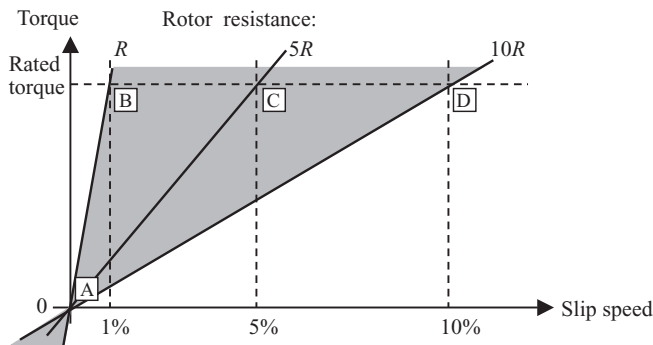


Figure 8.8 Operating envelope for a variable slip generator

seeks to regulate the torque to any desired value. For example, this might be PI algorithm with torque error input and the mark-space ratio of the switch as output.

In practice, it is usual to keep the torque demand at the rated value. Then the generator will simply act as a conventional induction generator following the slip curve AB until rated torque is reached, at which point it will accelerate along the constant torque line BCD just like a variable speed system. If the speed increases beyond D the torque is forced to increase again. Pitch control is used to regulate the speed to a chosen set-point such as point C. The higher the speed C, the higher the mechanical power input for the same output power. Thus, the power dissipated in the rotor circuit corresponds exactly to the slip. Therefore, C should be chosen as low as possible to minimise the cooling requirements (as well as turbine loads which increase with speed). However, if C is too close to B then the torque will occasionally dip down the slope AB as the speed varies around the set point, causing power dips even when operating well above rated wind speed. How small the interval between B and C can be made depends on the rotor inertia and the responsiveness of the pitch control algorithm. As for a variable speed system, the latter can be a PI or PID algorithm. It is possible to change the rate limits of the PID to force the pitch towards fine at maximum rate if the speed gets too close to B, or to feather at maximum rate if it gets too close to D.

As with a variable speed system, it may be desirable to modify the torque demand as in Section 8.3.6 to control drive train torsional vibrations. However, in order to do this, it is necessary to be able to update the torque demand at relatively high frequency, at least five and preferably ten times the drive train frequency which is typically of the order of 3–5 Hz.

8.3.9 Individual pitch control

Large pitch-regulated turbines invariably have a separate pitch actuator for each blade, since these can be used to provide effectively independent aerodynamic braking systems on the rotor. This means that no shaft brake is required, other than a small parking brake, because if one pitch actuator fails, the remaining actuator(s) should still be capable of stopping the rotor. Given that each blade has its own independent actuator, it is possible to send different pitch demands to each blade, and this can be used to reduce the asymmetrical aerodynamic loadings across the rotor, which are responsible for a significant contribution to fatigue loads (Donham *et al.*, 1979; Caselitz *et al.*, 1997; Bossanyi, 2004).

The simplest concept is cyclic pitch control based on the rotor azimuth. There are a number of effects which cause a systematic azimuthal variation of loading on each blade, in particular the wind speed variations caused by wind shear and tower shadow, and changes in angle of attack due to yaw misalignment, shaft tilt and upflow. In principle it should be possible to impose an azimuth-dependent change to the demanded pitch for each blade to compensate for these effects.

The tower shadow is very systematic and predictable, but to have any effect would require a very rapid and short-lived ‘blip’ in pitch demand as each blade passes the tower, which could have other adverse consequences. The effect of shaft tilt is also very predictable, and possibly upflow too in some situations, but these will affect only the angle of attack and not the local wind speed. Yaw misalignment also affects just the angle of attack, and to compensate for it would require an additional sensor to measure it, since the magnitude and direction of the misalignment will vary continuously. Wind shear does cause a significant difference in wind speed across the rotor, and therefore causes large blade load variations at the rotational frequency (1P), but again it is not a constant effect and would require one or more additional

sensors to detect it. Furthermore, the wind shear can only be regarded as a mean effect, and because of turbulence the instantaneous variation in wind speed across the rotor may be very different: in fact the highest wind speed could occur instantaneously anywhere on the rotor disc, not always at the top.

Indeed it is the turbulent variations in wind speed across a large wind turbine rotor which usually dominate the asymmetrical loading, since the size of a large rotor is comparable to the scale of turbulent eddies. Therefore, azimuth-dependent cyclic pitch control tends not to be very successful: while on average some reduction in loading ought to be achievable by compensating for the mean wind shear, this is insignificant compared to the stochastic effects of turbulence. An exception might be for a turbine operating in highly stratified flow with low turbulence.

Normally though, to achieve any significant reduction in these asymmetrical loads requires some additional measurement of the instantaneous turbulence, so that the individual pitch angles can be adjusted to compensate for these effects.

One possibility is actually to measure the incident wind flow just in front of each blade, for example using a set of pitot tubes along the blades, or to use pressure taps at appropriate locations, and then to adjust the pitch of each blade in response to the measurements on that blade. Such sensors have also been proposed to control ‘smart’ blades, which have actuators distributed along the span of the blade to alter its aerodynamic properties locally, such as flaps or ailerons, deformable trailing edges, or possibly air jets to modify the boundary layer flow. Such ideas are the subject of ongoing research and are currently a long way from any commercial deployment.

A more realistic possibility is to use load sensors to measure the blade bending moments at the root, or possibly at various points along the blade. It makes some sense to measure the very loads which we wish to reduce. With full-span pitch control, it would seem to make sense to measure the blade root loads on each blade and use the measurements to adjust the pitch of each blade in a feedback loop. Such a scheme could be called ‘independent pitch control’, although there is no definite consensus on nomenclature.

The possibility then arises to use the load measurement on each blade root as a predictor of the load which will be seen by the next blade when it sweeps past that position. This provides a degree of anticipation which should allow a further improvement in the control of each blade, and it works because turbulent eddies tend to be large enough so that as they pass through the rotor, each blade will slice through the same turbulence structure, perhaps even several times, before it has passed. Since the pitch of each blade is now calculated from the load measurements on all the blades, this can no longer be called ‘independent’ pitch control, and the term ‘individual pitch control’ may be more appropriate.

8.3.10 Multivariable control – decoupling the wind turbine control loops

The wind turbine controller is now a multivariable controller, with a number of inputs and outputs:

Inputs (measured signals):

- Generator speed (for speed regulation and drive train damping).
- Two tower accelerations (for tower damping).

- Three blade root loads (for a three-bladed turbine).

Outputs (demanded signals):

- Generator torque.
- Three pitch angles or rates (for a three-bladed turbine).

There are modern control methods which are appropriate to the design of controllers for such MIMO (Multiple Input, Multiple Output) systems – see Section 8.4.5. However, a MIMO system can sometimes be ‘diagonalised’ or transformed into a set of independent SISO (Single Input, Single Output) systems, in which case the controller for each SISO system can be optimised in isolation from the others. In fact this is possible to some extent for a wind turbine controller, and so all the control loops mentioned above can be designed using classical SISO design methods (Section 8.4.1). Actually the SISO loops are not quite independent, but the dynamic coupling between them can be small enough to make this a very successful approach in many cases.

It is relatively straightforward to decouple the pitch control from the torque control, as implied in the discussions above. In fact it is also possible to decouple the individual pitch control from the collective pitch control – the latter provides a collective pitch demand for speed regulation and tower damping, while the individual pitch control generates a separate pitch demand increment for each blade for minimising asymmetrical rotor loads. The pitch demand increments are all zero-mean, in such a way that the collective pitch control is not affected.

The main independent turbine control loops can now be summarised as follows:

- 1) Speed regulation loop using torque (using generator speed error to calculate the torque demand).
- 2) Drive train damping loop (using generator speed to calculate a modification to the torque demand).
- 3) Side-side tower damping loop (using side-side nacelle acceleration to calculate a further modification to the torque demand).
- 4) Speed regulation loop (using generator speed error to calculate the collective pitch demand).
- 5) Fore-aft tower damping loop (using fore-aft nacelle acceleration to calculate a modification to the collective pitch demand).
- 6) Individual pitch loop (using blade root loads to calculate individual pitch demand increments).

Loop 3 is not generally used but may become more interesting for offshore turbines which can be excited by wave action when this is misaligned with the wind direction.

There is actually some interaction between some of these loops; for example, loops 1 and 4 sometimes require notch filters tuned to the drive train resonant frequency to suppress coupling which would otherwise arise through the control action itself. Also loops 4 and 5 must be coupled in principle, since any change in pitch angle affects both the torque and the

thrust, but because loop 5 acts only in a restricted frequency range close to the first tower frequency it is usually possible to tune the loops independently; although a better result can be obtained with one or two iterations: one of the loops is tuned first using the open-loop plant model, then the plant is redefined by closing this loop while the other loop is tuned, and so on.

Loop 6 is still a MIMO loop, with as many inputs and outputs as there are blades. However, this can also be decoupled, as explained in the next section, by exploiting the symmetry which exists between the blades.

8.3.11 Two-axis decoupling for individual pitch control

To first order, the asymmetrical wind field across the rotor swept area can be linearised and described by two orthogonal components, for example as wind speed shear gradients in the horizontal vertical directions. Blade loading is closely related to wind speed, so this representation can also be used for a 'blade load field' (the 'blade load field' can be considered to include the effects of all three components of the local wind speed on the blade load). This description is independent of the number of blades or their speed of rotation, and the actual load seen by a blade at any instant can be thought of as the value of that field as sampled by the blade at its instantaneous position.

Furthermore, the pitch action needed to compensate for this variation in loading can also be described by a 'field' covering the swept area, and at any instant the pitch required by each blade is obtained by sampling that field at the instantaneous position of the blade.

Since each field is described by just two orthogonal components, a two-input, two-output controller is required to generate the pitch action 'field' from the load 'field'. Again this is independent of the number of blades.

Thus for a three-bladed rotor, the three measured blade root loads can be used to calculate the two components of the 'load field' at that instant. These are used to calculate the two components of the 'pitch field', from which the three individual pitch increments are calculated. The transformation between the three rotating blades and the two (non-rotating) field components is identical to Park's transformation for three-phase electrical machines (Park, 1929), which relates the currents or voltages in each phase winding to two notional orthogonal currents or voltages in the 'direct' and 'quadrature' axes. For this reason it is known as the *d-q* axis transformation. The same concept has also been used for helicopter rotors, where it is known as the Coleman transformation. The transformation from three rotating blade root loads L_1, L_2, L_3 to the non-rotating *d* and *q* axes can be written as follows:

$$\begin{bmatrix} L_d \\ L_q \end{bmatrix} = \frac{2}{3} \begin{bmatrix} \cos(\varphi) & \cos(\varphi + 2\pi/3) & \cos(\varphi + 4\pi/3) \\ \sin(\varphi) & \sin(\varphi + 2\pi/3) & \sin(\varphi + 4\pi/3) \end{bmatrix} \begin{bmatrix} L_1 \\ L_2 \\ L_3 \end{bmatrix}$$

where φ is the azimuth angle. The reverse transformation is:

$$\begin{bmatrix} \theta_1 \\ \theta_2 \\ \theta_3 \end{bmatrix} = \begin{bmatrix} \cos(\varphi) & \sin(\varphi) \\ \cos(\varphi + 2\pi/3) & \sin(\varphi + 2\pi/3) \\ \cos(\varphi + 4\pi/3) & \sin(\varphi + 4\pi/3) \end{bmatrix} \begin{bmatrix} \theta_d \\ \theta_q \end{bmatrix}$$

where θ would represent pitch angle in this case. This can be extended to any number of blades B , as follows:

$$\begin{bmatrix} L_d \\ L_q \end{bmatrix} = \frac{2}{B} \begin{bmatrix} \cos(\varphi) & \cos(\varphi + 2\pi/B) & \cos(\varphi + 4\pi/B) & \dots \\ \sin(\varphi) & \sin(\varphi + 2\pi/B) & \sin(\varphi + 4\pi/B) & \dots \end{bmatrix} \begin{bmatrix} L_1 \\ L_2 \\ L_3 \\ \dots \end{bmatrix}$$

$$\begin{bmatrix} \theta_1 \\ \theta_2 \\ \theta_3 \\ \dots \end{bmatrix} = \begin{bmatrix} \cos(\varphi) & \sin(\varphi) \\ \cos(\varphi + 2\pi/B) & \sin(\varphi + 2\pi/B) \\ \cos(\varphi + 4\pi/B) & \sin(\varphi + 4\pi/B) \\ \dots & \dots \end{bmatrix} \begin{bmatrix} \theta_d \\ \theta_q \end{bmatrix}$$

For a two-bladed machine this reduces simply to:

$$L_d = (L_1 - L_2) \cos(\varphi)$$

$$L_q = (L_1 - L_2) \sin(\varphi)$$

and

$$\theta_1 = -\theta_2 = \theta_d \cos(\varphi) + \theta_q \sin(\varphi)$$

In practice it is important to introduce an azimuthal phase shift into the reverse d - q axis transformation, by adding an offset to the rotor azimuth angle to compensate for the controller timestep and any other time delays in the control loop. In other words the pitch angle is calculated for the azimuth angle which will be reached by the time the pitch demand has been fully realised.

All that remains is to design a two-input, two-output controller $[C]$ to calculate the d - q axis pitch demands from the d - q axis loads:

$$\begin{bmatrix} \theta_d \\ \theta_q \end{bmatrix} = [C] \begin{bmatrix} L_d \\ L_q \end{bmatrix}$$

In the steady state there is clearly a one-to-one correspondence between the load and the pitch angle needed to compensate for it. It seems logical, therefore, to suppose that $[C]$ can be diagonal matrix, and furthermore because the rotor is rotationally symmetrical, the two diagonal terms should be identical. The design of the controller, therefore, boils down to designing a single SISO controller, and using two independent instances of it for the d and q axes. Since the wind field in the non-rotating frame varies relatively slowly, a straightforward, fairly low-bandwidth PI controller can be used for this.

Taking into account the dynamics, rotational sampling at the blade passing frequency means that there will be a certain speed variation at that frequency, resulting in corresponding variations in L_d and L_q . A notch filter at the blade passing frequency is, therefore, added in series with each PI controller. As for other PI loops, further notch or loop-shaping filters can be added if required.

Once the dynamics are taken into account, the rotor is no longer symmetrical because of its interaction with the tower dynamics. In principle this could lead to some asymmetry

between the d and q axes, and possibly also a small amount of dynamic coupling. Therefore, in principle there might be some advantage in designing in a coupled two-input, two-output controller (Bossanyi, 2003). In practice however, any advantage is likely to be small, and two independent and identical SISO controllers have been found to work extremely well. Furthermore, these simple controllers have been found to be very robust: they tend to be rather insensitive to the turbine dynamics, and also to load sensor calibration errors or drift.

The inverse d - q transformation converts the relatively slowly-varying d - q pitch demands into near-sinusoidal individual pitch demand increments for each blade. The near-sinusoids are of frequency 1P and phase-shifted between the blades, for example, by 120° for a three-bladed turbine. This form of control is, therefore, sometimes referred to as cyclic pitch control, but this is not correct: the controller is responding dynamically to the changing loads, so the pitch action is not actually sinusoidal, although it could be interpreted as sinusoidal with constantly changing amplitudes and phases. With PI controllers it is easy to limit the controller output to a maximum level, which corresponds to an upper limit on the amplitude of the sinusoids, and given the frequency (1P) this also determines the maximum additional pitch rate which would be demanded. This upper limit can be ramped down to zero in low winds, preventing any individual pitch action when the loads are small enough to contribute little to lifetime fatigue damage, so that the additional pitch action would not be worthwhile. It can also be used to ensure that the pitch demand does not fall below any physical pitch limit if this is close to the collective pitch demand in low winds. Another use is to prevent individual pitch action when it is more important to use the available pitch rates for collective pitch control, for example if the rotor is accelerating rapidly towards an overspeed trip (Savini *et al.*, 2010).

8.3.12 Load reduction with individual pitch control

The main effect of the once-per-revolution (1P) individual pitch control is to reduce the 1P out of plane loading on the blades, and hence also the rotating hub or shaft moments. Figure 8.9 shows spectra of the blade root out of plane and shaft bending moments in simulations with and without individual pitch control. In fact the spectral peak in these loads at the

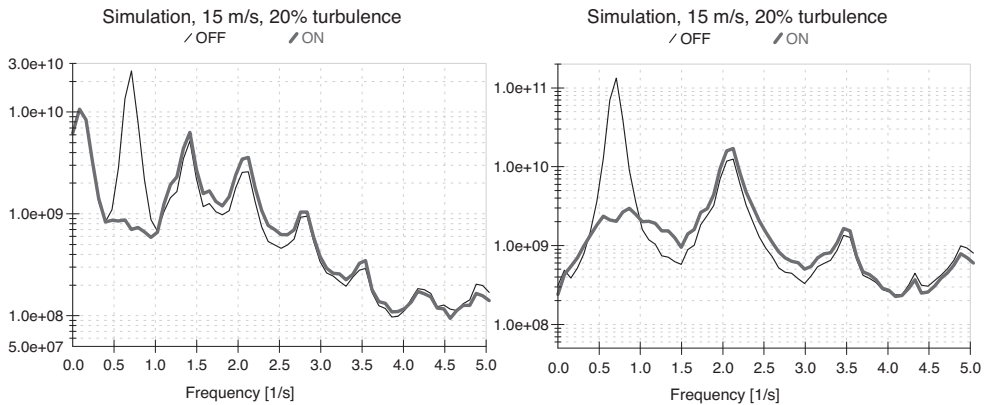


Figure 8.9 Effect of individual pitch control on rotating out of plane loads: blade root out of plane moment (left) and shaft bending moment (right)

1P frequency is virtually eliminated, an effect which has been confirmed also in field tests on an actual turbine (Bossanyi, 2010). Since the 1P load component dominates the fatigue, significant fatigue load reductions are obtained: typically of the order of 20% for blade root out of plane bending moment, and more for shaft bending moments (30–40%) because the low frequency variations cancel out between the blades so the 1P peak is even more significant.

On a two-bladed turbine, therefore, the use of individual pitch control represents a good alternative to the use of a teetered hub (Bossanyi *et al.*, 2009). Although it does not eliminate the teetering moment completely, a teetered hub often needs some kind of teeter restraint, which re-introduces a moment, and it is almost certainly necessary to consider the possibility of a teeter end-stop impact which can generate huge loads.

The 1P loading component on the rotor, when transformed to the non-rotating reference frame, results in loading contributions at 0P and 2P, so it is these load components which are reduced by individual pitch control. Hence the low frequency variation of nacelle nodding and yawing moments is removed, resulting in a reduction in peak loading – Figure 8.10 shows the effect on yaw moment, which may be of significant benefit in reducing the required yaw motor rating and duty. The nodding moment is reduced in a very similar way.

On a three-bladed turbine, there is no significant 2P component in the non-rotating loads, so only the low frequency load reduction is important here – the dominant source of fatigue loading on the non-rotating components is at 3P, and so is largely unaffected by the individual pitch control. However, for a two-bladed turbine, where this fatigue loading is dominated by 2P, the individual pitch control does significantly reduce the non-rotating fatigue loading. Again this has been confirmed in field tests (Bossanyi *et al.*, 2010).

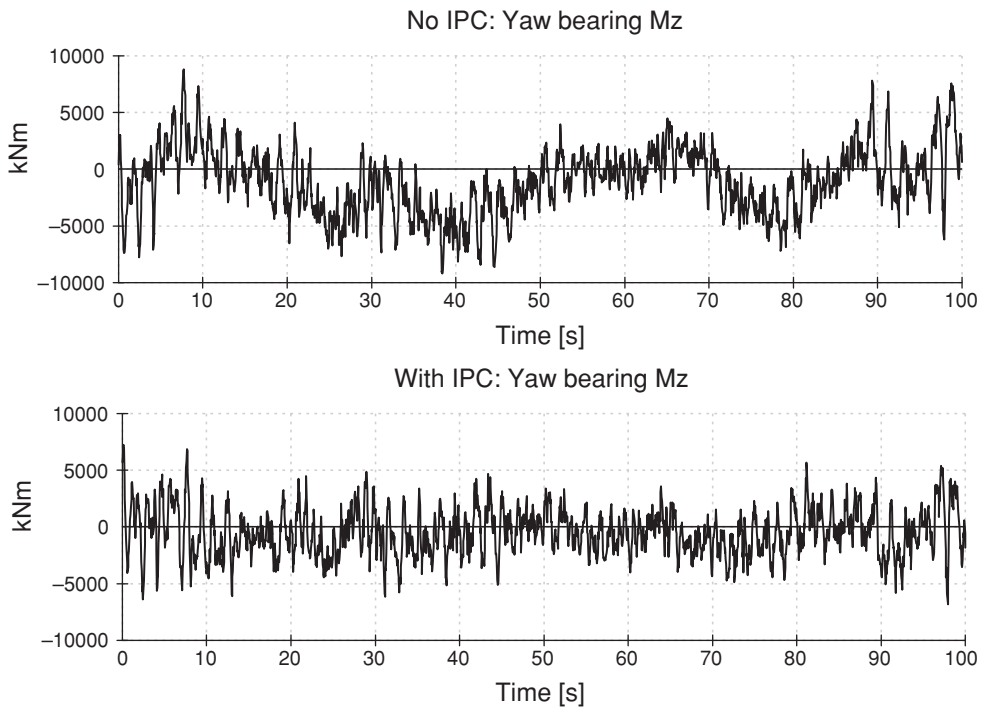


Figure 8.10 Effect of individual pitch control on yaw moment of a three-bladed turbine

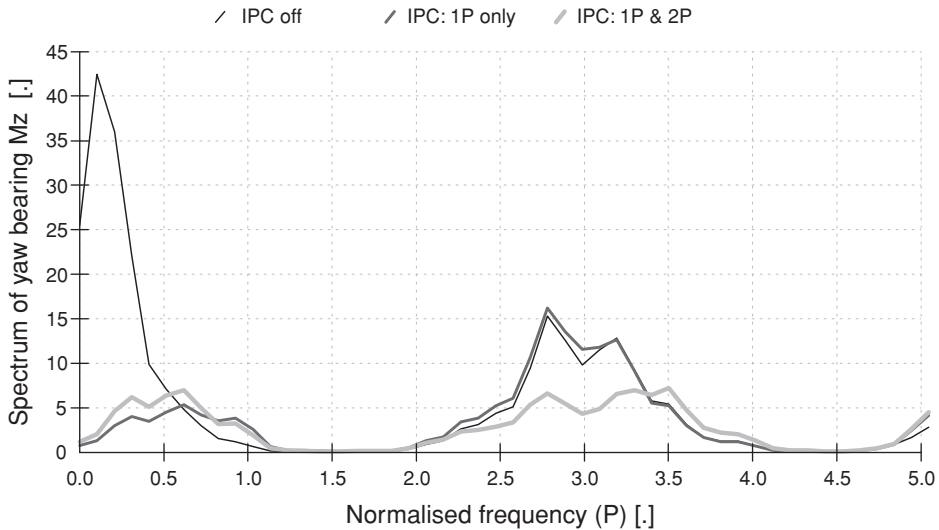


Figure 8.11 Effect of 1P and 2P individual pitch control on non-rotating loads

However, even for a three-bladed turbine it is possible to reduce these non-rotating fatigue loads by making use of second-harmonic individual pitch control. Taking 1P as the first harmonic, second-harmonic individual pitch control is achieved in exactly the same way but with the arguments to the sine and cosine functions in the rotational transformations multiplied by two, or by n for the n^{th} harmonic (although it may not be worthwhile to use more than the second harmonic in practice). Second-harmonic control results in 2P pitch action – hence, any 2P loading in the rotating components is reduced – but also the 1P and 3P loading in the non-rotating components (van Engelen *et al.*, 2005; Bossanyi *et al.*, 2009). On a three-bladed turbine, therefore, the dominant 3P non-rotating fatigue loads are reduced by this means, as shown in Figure 8.11.

Any number of harmonics may be used together as shown in Figure 8.12, simply by using parallel control loops.

8.3.13 Individual pitch control implementation

Individual pitch control requires additional sensors, so it is important to ensure that these are very reliable, otherwise the overall reliability of the turbine would be compromised. Conventional strain gauges are notoriously unreliable, although they certainly can last well

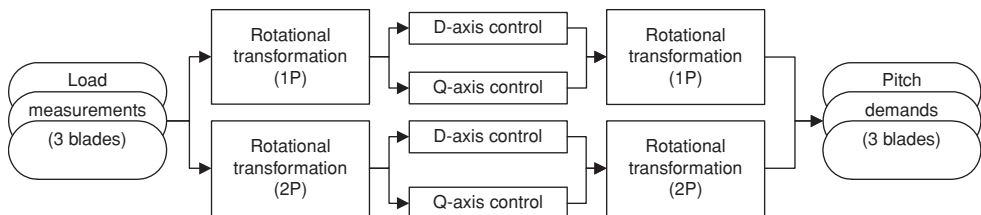


Figure 8.12 Adding higher harmonic individual pitch control loops

if very carefully installed. However, strain sensors based on fibre Bragg gratings are now available which have the potential to be sufficiently reliable for this application. Pulses of laser light are directed along an optical fibre, and a fine grating 'burnt' into the fibre at certain location reflects light of the same wavelength as the grating. The frequency of the reflected light is detected, and gives a direct measure of the strain at the position of the grating. Many gratings can be burnt into the same fibre, so strains at multiple locations can be measured at little extra cost: the time delay between sending the pulse and detecting the reflected signal determines the position of each grating. The components required are based on communications technology and are therefore becoming readily available. Furthermore, the special optical fibres can easily be included as part of the GRP layup during construction of a wind turbine blade.

Individual pitch control could equally be implemented using shaft bending sensors or even other sensors in the nacelle or tower top (Bossanyi, 2003), but it may be difficult to find suitable sensor locations. On the other hand if blade root sensors are used, they can also provide a measure of hub torque and rotor thrust. Although not currently used in this way, they might be useful as additional or alternative inputs to the other control loops described above, for example for damping tower fore-aft or drive train torsional vibrations.

If there is any failure of the strain sensors, there is the potential for individual pitch control to increase rather than reduce loading, which could be serious. Some failures are readily detected in a well-designed sensor system, but it is always possible that some types of failure may be hard to detect without some analysis of the signals by the controller, for example by comparing the sensor signals between different blades. If any failure is detected or suspected, the individual pitch control (being completely decoupled from the other control loops) can simply be switched off without the necessity to shut down the turbine, at least for a limited period or at a reduced power level until the fault can be rectified.

Below rated, the individual pitch control would normally be phased out since the loads are already smaller and so the additional pitch activity may not be justified. Also in principle there should be a small loss of energy output since the pitch angles are constantly moving either side of the optimum, although in practice this loss of output is usually very small. Above rated there is no loss of output, as the pitch angles are already well away from the optimum, and the collective pitch control loop ensures that rated output is maintained.

Clearly individual pitch control results in additional pitch actuator duty. The additional pitch action is concentrated around the 1P frequency. As turbines grow larger, the pitch rates required will diminish, since the rotational frequency will decrease as rotor diameter increases. If higher harmonic individual pitch control is used, for example at 2P, then there will also be additional pitch action at that frequency. Because the pitch action is near-sinusoidal, the maximum pitch rate required can be estimated as the product of the maximum amplitude limit and the frequency; this should be multiplied by $\sqrt{2}$ in case the d and q axis demands are simultaneously at the limit. The required actuator torque is no greater than normal (and may even be slightly reduced because the lower blade root bending moment implies lower bearing friction), but the actuators will be working harder because of the increased pitch rates. This will have implications for the thermal rating of the actuators.

The total lifetime pitch travel will increase, typically by a factor of around 3 (Bossanyi, 2003), which must be taken into account in the design of the pitch bearings.

Although fatigue loads can be significantly reduced by individual pitch control, there remains the possibility of increased extreme loads in the event of a forced shutdown by the safety system: if the pitch angles are all different by several degrees and that difference is

maintained as the pitch angles are ramped to feather, large asymmetrical loads, sometimes design-driving, can be generated. Where possible the individual pitch control is ramped out during the shutdown, but the safety system is unlikely to be allowed the sophistication required to do this. However, by reducing the individual pitch control amplitude as a function of rotor acceleration, this situation can effectively be avoided since the pitch angles are then likely to be much closer together when a safety system trip occurs (Savini *et al.*, 2010). The fatigue load reduction is hardly affected because these situations occur only rarely.

8.3.14 Further extensions to individual pitch control

Another theoretical possibility for individual pitch control is for actually generating yawing loads in response to measured yaw misalignment, in order to keep the turbine pointing into the wind without the use of a yaw motor. A yaw moment can easily be generated, simply by setting a non-zero set-point for one of the PI controllers. However, it is unlikely that the yaw motor can be dispensed with completely, as it will probably be needed to yaw the nacelle while the rotor is not turning, and also for cable unwinds, etc., so it may be better simply to use the individual pitch control with zero set-point to minimise the yaw moment which the yaw actuator has to overcome.

It is also possible that the d - q axis loads could help to infer rotor-averaged yaw misalignment, conceivably leading to better yaw control than just using the wind vane.

Nodding moments can be generated in the same way as yaw moments by setting a non-zero set-point: this could possibly be used for damping of higher fore-aft tower modes, or to help stabilise floating turbines (Namik and Stol, 2010). Some damping of side-side tower vibration is also possible by means of azimuth-dependent individual pitch control responding to side-side acceleration (Fischer *et al.*, 2010). Any of these applications would of course compromise the reduction of blade fatigue loads.

8.3.15 Commercial use of individual pitch control

Individual pitch control is now starting to be used on commercial turbines. While it makes little sense to retrofit this to an existing design, new commercial designs are starting to benefit from the reduced loads in two main ways:

- Some existing designs have been uprated or the rotor diameter increased, with individual pitch control being added at the same time to keep the loads within the existing design envelope, resulting in higher energy capture with minimal redesign of components and hence minimal change in capital cost.
- Some completely new turbines are now designed with individual pitch control from the start, leading to lower component costs. The significant change in the loading regime gives scope to reoptimise the whole design.

8.3.16 Feedforward control using lidars

In recent years, the development of lidar (laser Doppler anemometry) systems has reached the point where these devices can be used effectively for wind speed measurements at a distance, and their potential for site wind speed assessment and possibly power curve measurements is clear. The possibility to use a nacelle-mounted lidar to scan the approaching wind field

in front of the turbine for the purposes of improving the control has been suggested many times over the years, and is perhaps now becoming a possibility. The cost of these devices is still substantial, but may yet decrease enough to justify their use on large turbines, provided sufficient benefits can be found in terms of reduced loading or increased energy capture. Measuring the wind far upstream gives the control system time to take action in anticipation; but the further ahead the measurement is, the more the wind will have changed by the time it reaches the turbine. Nevertheless, this could be useful for anticipating rare extreme gusts: extreme loads can be reduced, for example by pitching the blades. Some promising results using simulations have been reported by Schlipf *et al.* (2010). However, it would be difficult to detect approaching wind direction changes, which are responsible for some extreme loading, because only the wind speed component in the direction of the beam is measured. For the same reason it would be difficult to scan the longitudinal wind speed at a very close distance in front of the whole rotor.

8.4 Closed loop control: analytical design methods

Clearly the choice of controller gains is crucial to the performance of the controller. With too little overall gain, the turbine will wander around the set point, while too much gain can make the system completely unstable. Inappropriate combinations of gains can cause structural responses to become excited. This section outlines some of the techniques which have been found to be useful in designing closed loop control algorithms for wind turbines, such as the gains of a PI or PID controller, for example. Clearly it is only appropriate here to give some useful hints and pointers. There are many standard texts on control theory and controller design methods, to which the reader should refer for more detailed information, for example D'Azzo and Houpis (1981), Anderson and Moore (1979), Astrom and Wittenmark (1990).

8.4.1 Classical design methods

A linearised model of the turbine dynamics is an essential starting point for controller design. This allows various techniques to be used for rapidly evaluating the performance and stability of the control algorithm. Detailed non-linear simulations using a three-dimensional turbulent wind input should then be used to verify the design before it is implemented on the real turbine.

For a variable speed turbine below rated wind speed, a PI speed controller using demanded torque can be quite slow and gentle, and the linearised model can be very simple. It must include the rotational dynamics of the drive train, but other dynamics are not usually important. For pitch control, however, the aerodynamics of the rotor and some of the structural dynamics can be critical. The linearised model for pitch controller design should contain at least the following dynamics:

- Rotor and generator rotation.
- Tower fore-aft vibration.
- Power or speed transducer response.
- Pitch actuator response.

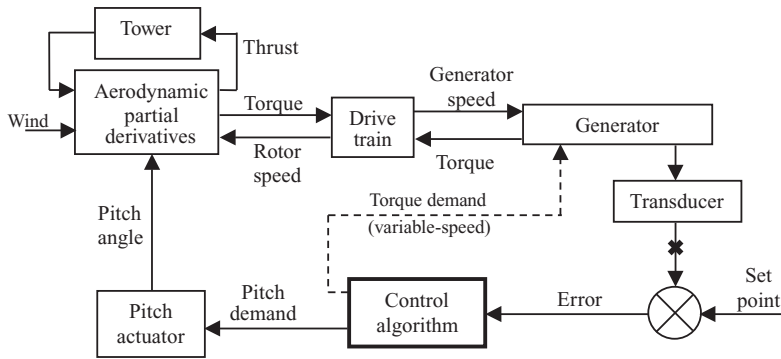


Figure 8.13 Typical linearised turbine model

The generator characteristics are also necessary for fixed speed systems, and drive train torsion is particularly important for variable speed turbines. In all cases a linearised description of the aerodynamics of the rotor is required, for example as a set of partial derivatives of torque and thrust with respect to pitch angle, wind speed and rotor speed. The thrust is important as it affects the tower dynamics, which couple strongly with pitch control.

A typical linear model is shown in Figure 8.13. With such a linear model, it is then possible to vary the gains and other parameters, and then rapidly carry out a number of tests which help to evaluate the performance of the controller with those gain settings. Some of these tests are open loop tests, which means they are applied to the open loop system obtained by breaking the feedback loop, for example at the symbol X in Figure 8.13. Other tests are carried out on the closed loop system. Before describing some of these tests, some basic theory on open and closed loop dynamics is outlined.

Figure 8.14 shows a simplified general model in which the turbine (i.e. from pitch actuator to transducer in Figure 8.13) is represented by the 'plant model' with transfer function $G(s)$, and the control algorithm is represented by the controller transfer function $k \cdot C(s)$, where s is the Laplace variable and k an overall controller gain.

Now the open loop system can be represented by the transfer function $k \cdot C(s) \cdot G(s) = H(s)$. If the input to the transfer function is denoted x and the output is y , then $Y(s) = H(s) \cdot X(s)$, where $X(s)$ and $Y(s)$ are the Laplace transforms of x and y . When the loop is closed at X the closed loop dynamics can be derived as

$$Y'(s) = H(s)(X(s) - Y'(s)) \quad (8.15)$$

where $Y'(s)$ is the Laplace transform of the closed loop output y' . This can be rewritten as

$$Y'(s) = \frac{H(s)}{1 + H(s)} X(s) \quad (8.16)$$

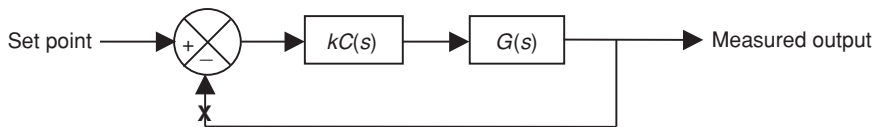


Figure 8.14 Simplified general model of plant and controller

In other words if the open loop system is $H(s)$, the closed loop system will have dynamics represented by $H'(s) = H(s)/(1 + H(s))$.

Now a linear transfer function can be expressed as the ratio of two polynomials in s . Thus, for the open loop system, $A(s)Y(s) = B(s)X(s)$, and so $H(s) = B(s)/A(s)$, where $A(s)$ and $B(s)$ are polynomials in s . The roots of the polynomial $A(s)$ give important information about the system response. Consider for example a first order system

$$\tau \dot{y}_1 = x - y_1 \quad (8.17)$$

representing a first order lagged response of y_1 with respect to x . This system can be represented by the transfer function

$$H(s) = \frac{B(s)}{A(s)}, \quad \text{where} \quad B(s) = 1 \quad \text{and} \quad A(s) = 1 + \tau s \quad (8.18)$$

The single root of $A(s)$ is given by $\sigma = -1/\tau$, while Equation 8.17 has solutions of the form $y = a + be^{\sigma t}$, with $\sigma = -1/\tau$ again. These solutions are stable if τ is positive, in other words if the root of $A(s)$ is negative. A second order system will have solutions of the form $y = a + be^{\sigma_1 t} + ce^{\sigma_2 t}$, where once again σ_1 and σ_2 are the roots of the second order polynomial which forms the denominator of the transfer function. Now σ_1 and σ_2 may be real numbers or they may form a complex conjugate pair $\sigma \pm j\omega$. The solutions are stable if σ_1 and σ_2 are both negative, or if σ is negative. In general, it can be stated that a linear system is stable if all the roots of the denominator polynomial have negative real parts. These roots are known as the *poles* of the system, and they represent values of the Laplace variable which make the transfer function infinite. The roots of the numerator polynomial are known as the *zeros* of the system, since the transfer function is zero at these points.

Now let us rewrite Equation 8.16 in terms of the polynomials A and B :

$$Y(s) = \frac{B(s)}{A(s) + B(s)} X(s) = \frac{k \cdot B'(s)}{A(s) + k \cdot B'(s)} X(s) \quad (8.19)$$

where we have reintroduced the overall controller gain k such that $B(s) = k \cdot B'(s)$. Clearly when the gain k is small, the closed loop transfer function tends towards the open loop transfer function $k \cdot B'/A$. However, when the gain is large, A can be neglected and so the poles will tend towards the roots of B' . In other words as the gain increases from zero to infinity, the poles of the closed loop system move from the open loop poles and end up at the open loop zeros. They move along complicated trajectories in the complex plane. A plot of these trajectories is known as a root locus plot, and is very useful for helping to select the feedback gain k . The gain is selected such that all the closed loop poles are in the left half-plane, making the system stable, and preferably as well damped as possible. The damping factor for a pole pair at $\sigma \pm j\omega = re^{j\theta}$ is given by $-\cos(\theta) = -\sigma/r$, as shown in Figure 8.15.

Figure 8.16 shows an example of a root locus plot for a variable speed pitch controller. As the gain increases, the closed loop poles (+) move from the open loop poles (x), corresponding to zero feedback gain, to the open loop zeros (O). (Actually there are usually more poles than zeros; the 'missing' zeros can be considered to be equally spaced around a circle of infinite radius.) In this example, the gain has been chosen to maximise the damping of the lightly-damped tower poles (B). Any further increase in gain would exacerbate tower vibration,

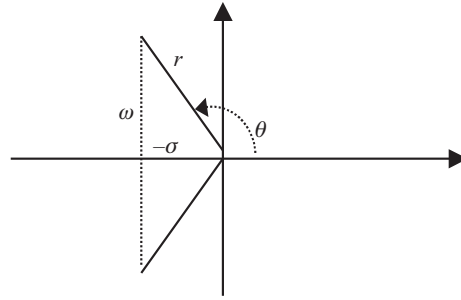


Figure 8.15 Damping ratio for a complex pole pair

eventually leading to instability as the poles cross the imaginary axis. At the chosen gain, the controller poles (A) are well damped. The poles at (C) result from the pitch actuator dynamics. They remain sufficiently well damped, although again, excessive gain would drive them to instability.

Although a root locus plot is useful for helping to select the overall controller gain, this can only be done once the other parameters defining the controller have been fixed. A PI controller (Equation 8.1 with $k_d = 0$) is characterised by only two parameters, K_p and K_i . It can be re-written as

$$y = K_p \left(1 + \frac{1}{sT_i} \right) x \quad (8.20)$$

where $T_i = K_p/K_i$ is known as the integral time constant. The root locus plot can be used to select K_p once T_i has been defined, but the shape of the loci will change with different T_i .

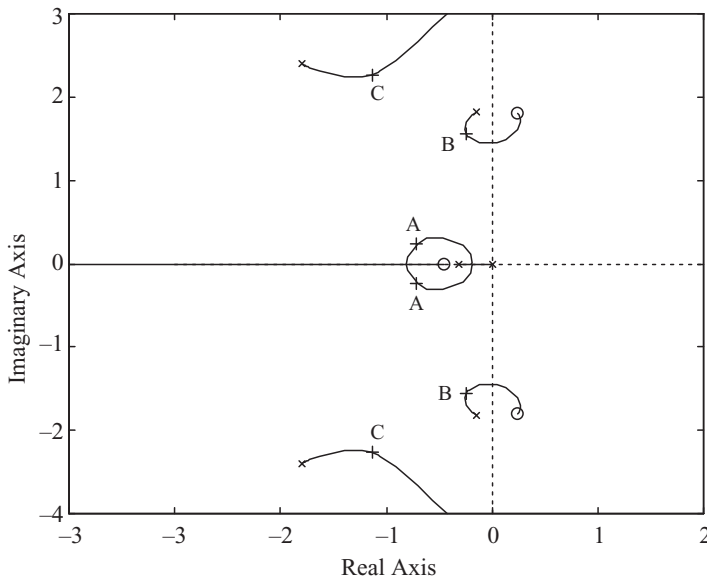


Figure 8.16 Example root locus plot for a variable speed pitch controller

However, it is relatively straightforward to iterate on the value of T_i , using the root locus plot each time to select K_p , until a suitable overall performance is achieved, using criteria such as those listed below. In the case of PID and more complex controllers, where more than two parameters must be selected, other ways must be found to select the parameters, although it is always possible to use a root locus plot for the final choice of the overall gain.

The choice of parameters will usually be an iterative process, often using a certain amount of trial and error, and on each iteration the performance of the resulting controller must be assessed. Useful measures of performance include:

- Gain and phase margins: these are calculated from the open-loop frequency response, and give an indication of how close the system is to instability. If the margins are too narrow, the system may tend to become unstable. The system will be unstable if the open loop system displays a 180° phase lag with unity gain. The phase margin represents the difference between the actual phase lag and 180° at the point where the open loop gain crosses unity. A phase margin of at least 45° is usually recommended, although there is no firm rule. Similarly, the gain margin represents the amount by which the open loop gain is less than unity where the open loop phase lag crosses 180° . A gain margin of at least a few decibels is recommended.
- The cross-over frequency, which is the frequency at which the open loop gain crosses unity, gives a useful measure of the responsiveness of the controller.
- The positions of the closed loop poles of the system indicate how well various resonances will be damped.
- Closed loop step responses, for example the response of the system to a step change in wind speed, give a useful indication of the effectiveness of the controller. For example, in tuning a pitch controller, the rotor speed and power excursions should return rapidly and smoothly to zero, the tower vibration should damp out reasonably fast, and the pitch angle should change smoothly to its new value, without too much overshoot and without too much oscillation.
- Frequency responses of the closed loop system also give some very useful indications. For example, in the case of pitch controller:
 - The frequency response from wind speed to rotor speed or electrical power should die away at low frequencies, as the low frequency wind variations are controlled away.
 - The frequency response from wind speed to pitch angle must die away at high frequencies, and must not be too great at critical disturbance frequencies such as the blade passing frequency, or the drive train resonant frequency in variable speed systems.
 - The frequency response from wind speed to tower velocity should not have too large a peak at the tower resonant frequency.

and so forth.

With experience it is possible, by examining measures such as these, to converge rapidly on a controller tuning which will work well in practice.

8.4.2 Gain scheduling for pitch controllers

Close to rated, since the fine pitch angle is selected to maximise power, it follows that the sensitivity of aerodynamic torque to pitch angle is very small. Thus, a much larger controller gain is required here than at higher wind speeds, where small change in pitch can have a large effect on torque. Frequently the torque sensitivity changes almost linearly with pitch angle, and so can be compensated for by varying the overall gain of the controller linearly in inverse proportion to the pitch angle. Such a modification of gain with operating point is termed a ‘gain schedule’. However, the sensitivity of thrust to pitch angle varies in a different way, and because of its effect on tower dynamics, which couples strongly with the pitch controller, it may be necessary to modify the gain schedule further to ensure good performance in all winds. In some cases just varying the overall gain may not be sufficient to achieve satisfactory response at all operating points, in which case it may be necessary to change the proportional and integral gains separately, each as a different function of pitch angle.

It is therefore important to generate linearised models of the system corresponding to several different operating points between rated and cut-out wind speed, and to choose a gain schedule which ensures that the above performance measures are satisfactory over the whole range.

For an active stall controller, the pitch angle may not change much with operating point; a gain schedule may not be required, but if it is, it may have to be a direct function of wind speed rather than pitch angle. This is one of the few occasions when the nacelle anemometer signal may have to be used as an input to the controller.

8.4.3 Adding more terms to the controller

It is often possible to improve the performance of a basic PI or PID controller by adding extra terms to modify the behaviour in a particular frequency range.

For example, a pitch control algorithm may be found to cause a large amount of pitch actuator activity at a relatively high frequency, which is of little benefit in controlling the turbine and may be quite counter-productive. This may occur if some dynamic mode was not taken into account in the linearised model which was used to design the turbine. An example of this is the drive train torsional resonance in a variable speed turbine, which can feed through to the measured generator speed and hence to the pitch control, causing high frequency pitch activity which is of no benefit. Another likely cause is the pitch response to a major external forcing frequency, such as the blade passing frequency. While a low pass filter in series with the controller will certainly reduce high frequency response, the resulting phase shift at lower frequencies may significantly impair the overall performance of the controller. A better ‘cure’ for excessive activity at some well-defined frequency is to include a notch filter in series with the controller. A simple second-order notch filter tuned to filter out a particular frequency of ω rad/s has a transfer function

$$\frac{1 + s^2/\omega^2}{1 + 2\zeta s/\omega + s^2/\omega^2} \quad (8.21)$$

where the ‘damping’ parameter ζ represents the width or ‘strength’ of the notch filter. This should be increased until the filtering effect is sufficient at the target frequency, without too much detriment to the control performance at lower frequencies.

Another useful filter is the phase advance or phase lag filter,

$$\frac{(1 + s/\omega_1)}{(1 + s/\omega_2)} \quad (8.22)$$

which increases the open loop phase lag between frequencies ω_1 and ω_2 ($\omega_1 < \omega_2$), or decreases it if $\omega_1 > \omega_2$. Phase advance can sometimes be useful for improving the stability margins. Open loop gain and phase plots can, therefore, be useful for helping to select ω_1 and ω_2 . A PID controller can be rewritten as a PI controller in series with a phase advance (or phase lag) filter.

A general second-order filter of the form

$$\frac{1 + 2\zeta_1 s/\omega_1 + s^2/\omega_1^2}{1 + 2\zeta_2 s/\omega_2 + s^2/\omega_2^2} \quad (8.23)$$

can sometimes be useful for modifying the frequency response in a particular area. With $\omega_1 = \omega_2$ and $\zeta_1 = 0$ this is just a notch filter, as described above. With $\zeta_1 > \zeta_2$ the filter has a band-pass effect, which can be used to increase control action at a particular frequency. With different ω_1 and ω_2 there is also a high-pass or low-pass effect, since the high-frequency gain tends towards $(\omega_2/\omega_1)^2$.

A root locus plot is often useful for investigating the effect of such filters. With experience, the effect on the loci of placing the filter poles and zeros in particular ways can be anticipated. Such techniques can help to see how, for example, a pair of lightly damped poles due to a structural resonance can be dragged further away from the imaginary axis, so as to increase the damping.

8.4.4 Other extensions to classical controllers

Other extensions to classical controllers have sometimes been used in order to further improve the performance in particular ways, for example the use of non-linear gains, and variable or asymmetrical limits.

Non-linear gains are sometimes used to penalise large peaks or excursions in controlled variables. For example, the gain of a PI pitch controller can be increased as the power or speed error increases; or, rather than changing the gain, an additional term can be added to the demanded pitch rate which may be a function of the error, its rate of change, or both. Often the additional term would be normally zero, increasing only in case of large deviations from the desired operating condition. The extra term can conveniently be added before the PI integrator (see Section 8.6.2). Such techniques should be used with caution, however, as too much non-linearity will drive the system towards instability, in much the same way as if the linear gain is too high. This technique requires a trial-and-error approach since it is very difficult to analyse the closed loop behaviour of non-linear systems using standard methods. Any asymmetry in the additional term, for example, if used only when the power or speed is above the set-point to help reduce peaks, will cause a reduction in the mean power or speed relative to the set-point.

Asymmetrical pitch rate limits can also be used to reduce peaks. By allowing the blades to pitch faster towards feather than towards fine, power or speed peaks will be reduced. Once again the mean level will also be reduced by introducing this asymmetry. However, this

technique is somewhat more ‘comfortable’ than the use of non-linear gains because it is less likely to lead to instability.

There is often a desire to reduce the set-point in high winds, to reduce the infrequent but highly-damaging loads experienced in those conditions at the expense of a small loss of output. It is straightforward to reduce the set-point as a function of wind speed (the pitch angle is usually used as a measure of the rotor-averaged wind speed, as for gain scheduling). However, the most damaging loads occur during high turbulence, and so it would be better to reduce the set-point in high winds only when the turbulence is also high. Rather than actually reducing the set-point, asymmetrical rate limits provide a simple but effective means of achieving this effect, since the rate limits will only be reached when the turbulence is high.

A further extension of this technique is to modify the rate limits dynamically, even to the extent of changing the sign of a rate limit in order to force the pitch in one direction during certain conditions such as large power or speed excursions. A useful application of this is in the control of variable slip systems, where it is important to keep the speed above the minimum slip point (point B in Figure 8.8). If the speed falls below this point, it then ceases to vary much as it is constrained by the minimum slip curve, and so the proportional term in the PI controller ceases to respond. Modifying the rate limits as a function of speed error as in Figure 8.17 is a useful technique to prevent this happening. Another useful application is to force some temporary pitch action such as a pitch ramp in response to a severe gust: by ramping the pitch rate limits, perhaps in response to an unusually large rate of generator acceleration, the PI controller can continue to act, albeit constrained by the rate limits, so that it resumes normal duty seamlessly when the rate limits are relaxed again.

Another case for set-point modification in high winds is to prevent a sudden loss of power arising from high wind cut-out. If an increase in wind speed can cause all the turbines in a large wind farm to shut down suddenly within a few minutes, the network will have to cope with this by providing spinning reserve. Instead of shutting down suddenly at 25 m/s, ramping the power output down smoothly from full power to zero between, say, 24 and 35 m/s will result in a much lower probability of a sudden shortfall, and the wind farm output can be considered more predictable (Bossanyi, 1982), both effects leading to lower spinning reserve requirements and, hence, a higher value for the generated power. This will have little effect on fatigue loading and energy yield because of the small number of hours involved, but the effect on extreme loads will need to be considered. For offshore turbines, where wave-induced tower vibration is better damped if the turbine is operating, extending the operating range to higher wind speeds may help to reduce tower loading (Markou *et al.*, 2009).

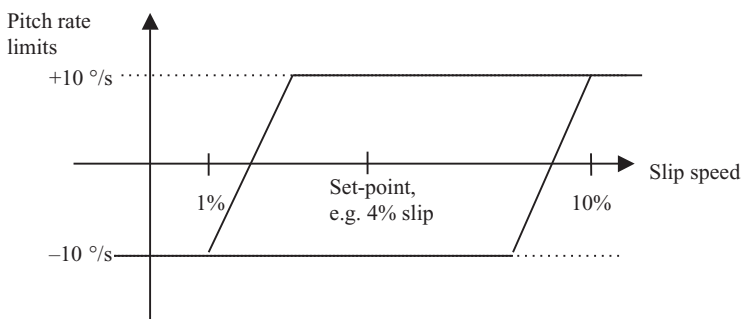


Figure 8.17 Pitch rate limit modification for a variable slip wind turbine

8.4.5 Optimal feedback methods

The controller design methods described above are based on classical design techniques, and often result in relatively simple PI or PID algorithms together with various filters in series or in parallel, such as phase shift, notch or band-pass filters, and sometimes using additional sensor inputs. These methods can be used to design fairly complex high-order controllers, but only with a considerable amount of experience on the part of the designer.

There is, however, a huge body of theory (and practice, although to a lesser extent) relating to more advanced controller design methods, some of which have been investigated to some extent in the context of wind turbine control, for example:

- Self-tuning controllers.
- Model-based controllers such as LQG/optimal feedback and H_∞ .
- Fuzzy logic controllers.
- Neural network methods.

Self-tuning controllers (Clarke and Gawthrop, 1975) are generally fixed-order controllers defined by a set of coefficients, which are based on an empirical linear model of the system. The model is used to make predictions of the sensor measurements, and the prediction errors are used to update the coefficients of the model and the feedback law.

If the system dynamics are known, then some very similar mathematical theory can be used, but applied in a different way. Rather than fitting an empirical model, a linearised physical model is used to predict sensor outputs, and the prediction errors are used to update estimates of the system state variables. These variables may include rotational speeds, torques, deflections, etc., as well as the actual wind speed, and so their values can be used to calculate appropriate control actions even though those particular variables are not actually measured.

Observers. A subset of the known dynamics may be used to make estimates of a particular variable: for example, some controllers use a wind speed observer to estimate the wind speed seen by the rotor from the measured power and/or rotational speed and the pitch angle. The estimated wind speed can then be used to define the appropriate desired pitch angle.

State estimators. Alternatively, using a full model of the dynamics, a Kalman filter can be used to estimate all the system states from the prediction errors (Bossanyi, 1987). This technique can explicitly use knowledge of the variance of any stochastic contributions to the dynamics, as well as noise on the measured signals, in a mathematically optimum way to generate the best estimates of the states. This relies on an assumption of Gaussian characteristics for the stochastic inputs. Thus, it is possible explicitly to take account of the stochastic nature of the wind input by formulating a wind model driven by a Gaussian input. This can even be extended to include blade passing effects.

The Kalman filter can readily take account of more than one sensor input in generating its ‘optimal’ state estimates. Thus, it is ideal for making use of, for example, an accelerometer measuring tower fore-aft motion as well as the normal power or speed transducer. It would be straightforward to add other sensors, if available, to improve the state estimates further.

Optimal feedback. Knowing the state estimates, it is then possible to define a cost function, which is a function of the system states and control actions. The controller objective can then be defined mathematically: the objective is to minimise the selected cost function. If the cost function is defined as a quadratic function of the states and control actions (which is actually a

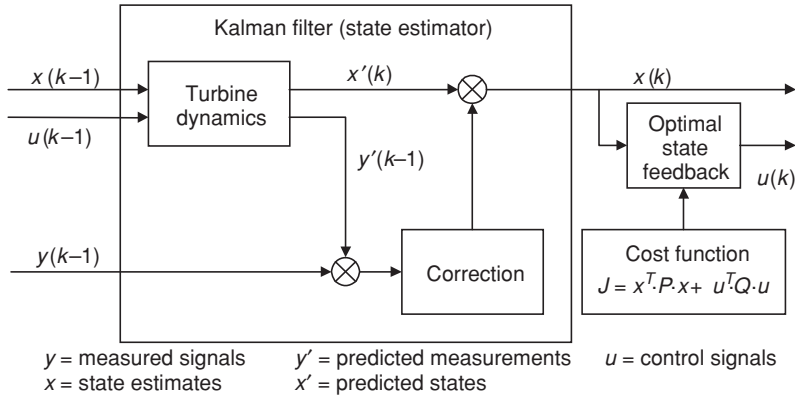


Figure 8.18 Structure of the LQG controller

rather convenient formulation), then it is relatively straightforward to calculate the ‘optimal’ feedback law. This is defined as the feedback law which generates control signals as a linear combination of the states such that the cost function will be minimised. Since a *Linear* model is required, with a *Quadratic* cost function and *Gaussian* disturbances, this is known as an LQG controller.

This cost function approach means that the trade-off between a number of partially competing objectives is explicitly defined, by selecting suitable weights for the terms of the cost function. This makes such a method ideal for a controller which attempts to reduce loads as well as achieving its primary function of regulating power or speed. Although it is not practical to calculate the weightings in the cost function rigorously, they can be adjusted in a very intuitive way. This approach is also readily configured for multiple inputs and outputs, so, for example, as well as using generator speed and tower acceleration inputs, it can in principle simultaneously produce the pitch demand and torque demand outputs which will minimise the cost function.

Figure 8.18 illustrates the structure of the LQG controller, showing the state estimator and the optimal state feedback. For implementation, the entire controller can be reduced to a set of difference equations connecting the measured outputs (y) to the new control signals (u). This means that once the design is completed, the algorithm is easy to implement and does not require massive processing power.

The linearised dynamics of the system are expressed in discrete state-space form:

$$x'(k) = Ax(k-1) + Bu(k-1)$$

The Kalman gain L is calculated taking into account the stochastic disturbances affecting the system, and allows the state estimates to be improved by comparing the predicted sensor outputs y' to the actual outputs y :

$$x(k) = x'(k) + L(y(k-1) - y'(k-1))$$

where

$$y'(k-1) = Cx(k-1) + Du(k-1)$$

The optimal state feedback gain K generates the control actions

$$u(k) = -Kx(k)$$

where K is calculated such that the quadratic cost function J is minimised. The cost function is

$$J = x^T P x + u^T Q u$$

(actually the integral, or the mean value over time, or the expected value of this quantity). P and Q are the state and control weighting matrices. It is usually more useful to define the cost function in terms of other quantities, v , which can be considered as extra (often un-measured) outputs of the system:

$$v = C_v x + D_v u$$

Hence the cost function is

$$J = v^T R v + u^T S u = x^T C_v^T R C_v x + u^T D_v^T R D_v u + u^T S u$$

so that $P = C_v^T R C_v$ and $Q = D_v^T R D_v + S$.

Another possibility is to generate optimal control signals directly as a function of the sensor outputs. This is known as optimal output feedback (Steinbuch, 1989). However, the mathematical solution of this problem is based on *necessary* conditions for optimality which are not always *sufficient* for optimality. Therefore, the solutions generated can be, and in practice often are, non-optimal and potentially very far from optimal. This variation is therefore rather problematic.

As turbines become larger and the requirements placed on the controller become more demanding, advanced control methods such as LQG are likely to become increasingly used, although as yet there are few published examples of the practical application of these techniques in commercial wind turbines. However, this approach was used to design a controller for a 300 kW fixed speed two-bladed teetered turbine in the UK in 1992. After testing on a prototype turbine in the field, this controller was shown to give significant reductions in pitch activity and power excursions compared to the original PI controller, and it was subsequently adopted for the production machine and successfully used on over 70 turbines (Bossanyi, 2000). Stol *et al.* (2004) reported tests with a similar control scheme on a 600 kW research turbine.

LQG controllers are not necessarily robust, which means that they can be sensitive to errors in the turbine model. A similar approach is the H_∞ controller, in which uncertainties in the turbine and wind models can explicitly be taken into account. Such a controller was evaluated in the field on a 400 kW fixed speed pitch regulated turbine by (Knudsen *et al.*, 1997), who reported a reduction in pitch activity and some potential for reduced fatigue loads compared to a PI controller.

8.4.6 Pros and cons of model-based control methods

The methods of Section 8.4.5 are appealing as they use mathematical rigour to calculate an ‘optimal’ controller in the sense that it minimises a pre-defined and reasonably intuitive cost

function, suggesting that the tuning could be an automatic process, whereas the classical approach relies on the skill and experience of the designer for each new tuning. They are also ideal for designing MIMO controllers, which could require a cumbersome iterative approach using classical design methods.

There are also some disadvantages, however, which may explain the continuing prevalence of classically-designed controllers in commercial wind turbines.

In practice, however, ‘tuning’ the cost function may end up being just as difficult as tuning a classical controller, and the tuning may need to be repeated for each new turbine even though in principle this ought to be unnecessary. The cost function needs to include terms for any states or outputs which should be minimised, but the choice of such variables is not as straightforward as it might appear. For example, for a variable speed controller it would be logical to include a term to minimise the speed error; but in practice a term is also required to minimise the integral of the speed error, and adjusting the relative weights for these two terms is very similar to adjusting the proportional and integral gains in a classical design.

Also the cost function is defined as a quadratic function of the states and other variables, and this may not be appropriate for minimisation of fatigue loads, for example, as fatigue is a highly non-linear process. Even for speed regulation, one could argue that minimising the speed error is not important (this may even contradict the need to minimise loads) but minimising extreme speed excursions to avoid any overspeed trips is all that matters. A quadratic cost function is not ideal for this, as the true ‘cost’ increases dramatically at the overspeed trip limit.

Classical controllers are simpler to implement; they can easily deal with non-linearities through techniques such as gain scheduling, and simple adjustments such as the addition of notch filters are straightforward, as is the imposition of fixed or variable rate limits. Model-based controllers require further sophistication such as extended Kalman filters or fuzzy transitions in order to deal with non-linearities, and any adjustment requires a complete recalculation of the controller. Integration with the supervisory control is also much less straightforward; for example it might be desirable to modify the tower acceleration feedback and/or the individual pitch control during a shutdown in order to reduce extreme loads. With a classical controller it is easy to impose variable schedules or limits to achieve this, but it is much more difficult to do this with model-based controllers.

As explained in Section 8.3.10, most of the wind turbine control problem can be decomposed into separate, almost uncoupled SISO loops. This makes it perfectly feasible to use straightforward classical tuning techniques. The only significant coupling between these loops is between speed regulation and tower damping, but this is easily dealt with by means of just one or two iterations, tuning each loop on its own with the other loop implemented as part of the plant.

Nevertheless, as turbines become larger, lighter and more flexible, it is possible that model-based multivariable methods, perhaps in conjunction with additional sensors, will increasingly find a role.

8.4.7 Other methods

Rule-based or ‘fuzzy logic’ controllers are useful when the system dynamics are not well known or when they contain important non-linearities. Control actions are calculated by weighting the outcomes of a set of rules applied to the measured signals. Although there has been some work on fuzzy controllers for wind turbines, there is no clear evidence of benefits.

In practice, quite a good knowledge of the system dynamics is usually available, and the dynamics can reasonably be linearised at each operating point, so there is no clear motivation for such an approach.

The same could be said of controllers based on neural networks. These are learning algorithms, which are ‘trained’ to generate suitable control actions using a particular set of conditions, and then allowed to use their learnt behaviour as a general control algorithm. While this is potentially a powerful technique, it is difficult to be sure that such a controller will generate acceptable control actions in all circumstances.

Nevertheless, there may be some potential for such methods where significant non-linearities or non-stationary dynamics are involved. These might be in the turbine itself (stall hysteresis might be one example), in the driving disturbances (the wind characteristics are not stationary), or in the controller objectives. For example, the controller objectives might change around rated wind speed, or non-linear effects such as fatigue damage might be included in the cost function.

8.5 Pitch actuators (see also, Chapter 6 Section 6.7.2)

An important part of the control system of a pitch-controlled turbine is the pitch actuation system. Both hydraulic and electric actuators are commonly used, each type having its own particular advantages and disadvantages which should be considered at the design stage.

Smaller machines might have a single pitch actuator to control all the blades simultaneously, but larger turbines usually have individual pitch actuators for each blade. This has the advantage that it is then possible to dispense with the large and expensive shaft brake which would otherwise be needed. This is because of the requirement for a turbine to have at least two independent braking systems capable of bringing the turbine from full load to a safe state in the event of a failure. Provided the individual pitch actuators can be made independently failsafe, and as long as the aerodynamic braking torque is always sufficient to slow the rotor down to a safe speed even if one pitch actuator has failed at the working pitch angle, then multiple actuators may be considered to be independent braking systems for this purpose. There may still be a need for a parking brake, at least for the use of maintenance crews, but this may then be fairly small. It must at least be capable of bringing the rotor to a complete stop from a low speed, not necessarily in high or extreme wind speeds, for long enough to allow a rotor lock to be inserted.

A collective pitch actuation system commonly consists of an electric or hydraulic actuator in the nacelle, driving a push-rod which passes through the centre of the gearbox and hollow main shaft. The push-rod is attached to the pitchable blade roots through mechanical linkages in the hub. The actuator in the nacelle is often a simple hydraulic cylinder and piston. A charged hydraulic accumulator ensures that the blades can always be feathered even if the hydraulic pump loses power. An alternative arrangement is to use an electric servo motor to drive a ballnut which engages with a ballscrew on the push-rod. Since the push-rod turns with the rotor, loss of power to the motor causes the ballscrew to wind the pitch to feather, giving failsafe pitch action. This requires a failsafe brake on servo motor to ensure that the ballnut stops turning if power is lost.

Individual pitch control requires separate actuators in the hub for each blade. Therefore, there must be some means of transmitting power to the rotating hub to drive the actuators. This can be achieved by means of slip rings in the case of electric actuators; or a rotary hydraulic

joint for hydraulic actuators if the hydraulic power pack is located in the nacelle. A rotary transformer could be used to transmit electrical power to the hub without the inconvenience of slip rings, which require maintenance.

The need to ensure a backup power supply on the hub to enable the blades to pitch even in the event of power loss can be a source of problems. A hydraulic system needs an accumulator for each blade, while electric actuators usually have battery packs in the hub for this purpose. Such battery packs are large, heavy and expensive, and alternative methods such as the use of hub-mounted generators, which can always generate power as long as the hub is turning, have been proposed. If a battery is used, the actuator motors must either be DC motors or (more commonly) AC motors with a frequency converter, with the batteries on the DC link. Since this will form part of the safety system, the reliability of the inverter between the DC link and the pitch motor is important. A hub-mounted generator would produce either DC or variable frequency AC, and once again the reliability of the connection to the pitch motor is important. Since the pitch actuators have to be independently failsafe, separate battery packs or generators and frequency converters etc. must be provided for each blade.

The friction in the pitch bearing is often a significant factor in the design of the pitch actuation system. The bearing friction depends on the loading applied to the bearing, and the large overturning moment acting on the bearing can lead to very high levels of friction: often most of the actuator torque is required to overcome the bearing friction.

A hydraulic actuator would usually be controlled by means of a proportional valve controlling the flow of oil to the actuator cylinder. The valve opening, and hence the oil flow rate, would be set in proportion to the required pitch rate. The demanded pitch rate may come directly from the turbine controller, or it might come from a pitch position feedback loop. In this case the turbine controller generates a pitch position demand. This is compared to the measured pitch position, and the pitch position error is turned into a pitch rate demand through a fast PI or PID control loop, implemented either digitally or by means of a simple analogue circuit.

In the case of an electric actuator, the motor controller usually requires a torque demand signal. This may be derived from a speed controller, which uses a fast PI or PID controller acting on speed error to generate a torque demand. Once again the speed demand may come directly from the turbine controller or from a position feedback loop.

Simpler actuators could be used if a fast pitch response is not important, for example in a turbine which is controlled by pitching to stall rather than to feather. In this case an actuator which merely pitches at a fixed rate in either direction may be adequate.

8.6 Control system implementation

Previous sections have explained some of the techniques whereby control algorithms can be designed. The system and controller dynamics have been described in continuous time in terms of the Laplace operator, s . While it is possible to implement a continuous-time controller, for example using analogue circuitry, the use of digital controllers is now almost universal. The greater flexibility of digital systems is a factor here: simply by making software changes, the control logic can be changed completely.

A consequence of using digital control is that the control actions are calculated and updated on a discrete time step, rather than in continuous time. Control algorithms designed in continuous time must, therefore, be converted to discrete time for implementation in a

digital controller. It is also possible to design controllers in discrete time, if the linearised model of the turbine is first discretised.

The following sections briefly describe some of the practical issues involved in implementing a control algorithm in a real digital controller. Once again, the reader is referred to standard control theory texts for more detailed treatments.

8.6.1 Discretisation

Supposing a control algorithm has been designed in continuous time as a transfer function (such as Equation 8.1 for a PID controller, for example), it must be discretised before it can be implemented in a digital controller. Discretised transfer functions are usually represented in terms of the delay operator, z , where $z^{-k}x$ represents the value of x sampled k timesteps ago. As a simple example, a moving average or lag filter from x to y is often implemented as

$$y_k = Fy_{k-1} + (1 - F)x_k$$

This is a difference equation which can readily be implemented in code in a discrete controller. In terms of the delay operator, it can be written as

$$(1 - Fz^{-1})y = (1 - F)x$$

or alternatively as a transfer function consisting of a ratio of polynomials in z^{-1} :

$$y = \frac{(1 - F)}{(1 - Fz^{-1})}x$$

Now the Laplace operator can be considered as a differentiation operator, and so as a simple approximation, it might be possible to convert a continuous transfer function into discrete form by replacing s by $(1 - z^{-1})/T$, where T is the timestep.

In fact by simple algebraic manipulation, it is straightforward to show that with this substitution, the above discrete transfer function is in fact equivalent to the continuous transfer function representation of a first order lag with time constant τ , namely

$$y = \frac{1}{1 + s\tau}x$$

with the factor F being given by $\tau/(T + \tau)$.

Clearly any discretised equation can only be an approximation to the continuous-time behaviour. There are other discretisation methods, and the so-called bilinear or ‘Tustin’ approximation often works better in practice. In this case the Laplace operator is replaced by

$$\frac{2}{T} \frac{(1 - z^{-1})}{(1 + z^{-1})}$$

Discretisation results in a phase shift compared to the continuous time process. This phase shift increases with frequency. If the algorithm performance is particularly sensitive to the phase shift at a certain frequency, then the discretisation can be ‘pre-warped’ to this

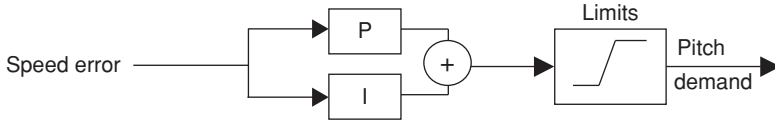
frequency. Pre-warping modifies the phase shift so that the phase of the discrete transfer function is correct at the chosen frequency, but deviates at lower and higher frequencies. An example of a situation where pre-warping may be important is in the case of a drive train resonance damper in a variable speed turbine (Section 8.3.5). The resonant frequency which is being targeted is usually fairly high, typically around 3 or 4 Hz, and unless the controller timestep is very short the phase lag caused by discretisation may significantly affect the performance of the damping algorithm.

The approximation for s used for discretisation with pre-warping about a frequency ω is

$$\frac{\omega}{\tan(\omega T/2)} \frac{(1 - z^{-1})}{(1 + z^{-1})}$$

8.6.2 Integrator desaturation

Controllers containing integral terms, such as PI or PID controllers, experience a particular problem known as integrator wind-up when the control action saturates at a limiting value. A common example is in pitch control, where the pitch angle is limited to the fine pitch position when the wind is below rated. For example, a PI pitch controller for a variable speed turbine can be represented as:



Above rated, the speed error will be zero on average because of the integral term. Below rated, the pitch saturates at the fine pitch position, and the speed error will remain negative. The integral of the error will, therefore, grow more and more negative, and only the application of the limits prevents the actual pitch demand from doing the same. However, when the wind speed reaches rated again and the speed error becomes positive, it will take a long time before the integrated power error climbs back up to zero and starts to demand a positive pitch angle. To prevent this problem of integrator wind-up, the integral term must be prevented from integrating when the pitch is at the limit. This is easily implemented by separating out the integrator, $I(z)$, from the rest of the controller, $R(z)$. $R(z)$ generates a change in demanded pitch angle and $I(z)$ then integrates this by adding it to the previous pitch demand *after* the limits have been applied.

As an example, a PI controller (Equation 8.20) discretised using the bilinear approximation would be:

$$y = K_p[(T/2T_i + 1) + (T/2T_i - 1)z^{-1}] \cdot \frac{1}{[1 - z^{-1}]} \cdot x = R(z) \cdot I(z) \cdot x$$

To avoid integrator wind-up, this can be implemented as follows:

$$\Delta y_k = K_p[(T/2T_i + 1)x_k + (T/2T_i - 1)x_{k-1}]. \text{ (implementation of } R(z)\text{)}$$

$$y_k^* = y_{k-1} + \Delta y_k \text{ (integrator } I(z) \text{ using previous } \textit{limited} \text{ output } y_{k-1}\text{)}$$

$$y_k = \lim(y_k^*) \text{ (application of limits)}$$

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Wind turbine installations and wind farms

For any wind turbine installation, there are certain additional activities, (e.g. construction of foundations and access roads, electrical connections, site erection and project development and management) that must be undertaken. For flat onshore sites, which might be found typically in Denmark or North Germany, the total investment cost is approximately 1.3 times the ex-works turbine cost. In the UK, where sites are often located in more remote, upland areas the balance-of-plant costs (i.e. all costs other than the wind turbines) tend to be higher and a more typical breakdown is shown in Table 9.1.

Civil engineering costs are increased by difficult ground conditions, rock or very boggy soil and difficult access. High wind speed sites will give higher energy output but require more expensive foundations to withstand the extreme loads. Electrical grid connection costs are determined by the distance to the point of connection and the voltage level at which connection is made (EWEA, 2009). The ex-works price of the turbines (65–75% of the project cost) will depend on the state of the market for wind turbines at the time the order is placed.

Commercial developers of wind farms will often prefer larger projects as, in that way, the fixed-costs, particularly grid connection and project set-up and management costs, may be spread over a bigger investment. A further encouragement for large projects is that the fixed costs of arranging project finance can be high. However, there are individuals, community groups and commercial organisations that develop smaller wind farms and projects of this type have been common in Denmark and Germany. The advantage of community involvement in the project is, of course, that planning permission is more readily obtained if it is seen that there is tangible benefit to local people.

Table 9.1 Typical breakdown of costs for a 10 MW wind farm on an upland UK site.

Element of wind farm	% of total cost
Wind turbines	65
Civil works (including foundations)	13
Wind farm electrical infrastructure	8
Electrical network connection	6
Project development and management costs	8

9.1 Project development

The development of a wind farm follows a broadly similar process to that of any other power generation project, but with the particular requirements that the wind turbines must be located in high wind speed sites to maximise energy production and their size makes visual effects a particularly important aspect of the environmental assessment. Guidance on the development of wind energy schemes was issued originally by the British Wind Energy Association, now renamed RenewableUK, in their ‘Best Practice Guidelines for Wind Energy Development’ (BWEA, 1994). Comprehensive guidance on the development of wind farms and many aspects of environmental assessment is given in EWEA (2009) and in AWEA (2008).

Three main elements of the development of a wind farm project may be identified as: (1) technical and commercial issues, (2) environmental considerations and (3) dialogue and consultation. Perhaps surprisingly to many engineers and technologists the technical and commercial considerations are often the more straightforward and the success of a wind farm project hinges critically on environmental considerations and the dialogue and consultation process with local residents and planning authorities.

Wind farm development may be divided into a number of phases:

- Initial site selection.
- Project feasibility assessment.
- Preparation and submission of the planning (permitting) application.
- Construction.
- Operation.
- Decommissioning and land reinstatement.

Consideration needs to be given to how the wind farm electrical output is to be sold. In countries with a Feed-in-Tariff this is fairly straightforward but with alternative support mechanisms, such as Renewable Portfolio Standards, it is necessary to identify a purchaser of the electrical output as well as any ‘Green Certificates’ if these are traded separately.

9.1.1 Initial site selection

Initially a desk based study is carried out to locate a suitable site and to confirm it as a potential candidate for the location of a wind farm. It may be recalled that the mean power production

for a wind turbine, assuming 100% availability, is given by (see Section 3.16, Chapter 3):

$$E = T \int P(U)f(U)dU \quad (9.1)$$

where $P(U)$ is the Power Curve of the wind turbine, $f(U)$ Probability Density Function (PDF) of the wind speed and T is the time period.

The Power Curve is available from the potential turbine supplier(s) while an initial estimate of the PDF of the wind speed may be obtained from a wind atlas (for example, Troen and Petersen, 1989). The PDF is generally based on a Weibull distribution and takes account of regional climatology, roughness of the surrounding terrain, local obstacles and topology. PDFs are calculated for twelve 30° sectors and integrated with the Power Curve usually using numerical techniques. At this stage of the project development only an approximate indication of the wind farm output is required in order to confirm the potential of the site(s). However, it may be useful to supplement the wind atlas with some form of initial computer modelling.

A wind speed database covers the UK based on the NOABL airflow model (ETSU, 1997a). The model represents the effect of topography on wind speed and provides estimates of annual mean wind speeds with a resolution of 1 km square at 10 m, 25 m or 45 m above ground level. It does not take account of any local or thermal effects. These can have a considerable effect on wind speeds and so on-site measurements are required for accurate assessment of the energy output of a wind farm.

If measured site wind speed data is available then the energy yield of a wind turbine can be estimated, as shown in Figure 9.1, by combining the binned wind speed distribution with the power curve:

$$\text{Energy} = \sum_{i=1}^{i=n} H(U_i)W(U_i) \quad (9.2)$$

where $H(U_i)$ is the number of hours in wind speed bin U_i , $W(U_i)$ is the power output at that wind speed and there are n wind speed bins.

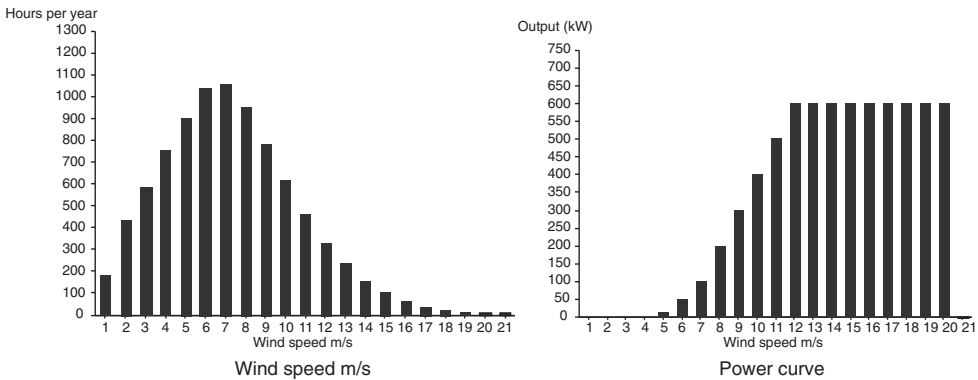


Figure 9.1 Annual energy calculation of a wind turbine

Table 9.2 Typical separation distances for medium sized wind turbines

Feature	Typical separation distances
Residential property	350–500 m
Motorway, trunk road, railway	Tip of turbine +10%
Public right-of-way (e.g. footpath)	50 m – tip of turbine, depending on use of right-of-way
Electric overhead line	Tip of turbine +10%
Turbines	5–6 rotor diameters in prevailing wind direction 3–4 rotor diameters across prevailing wind direction

In addition to the wind resource it is also necessary to confirm that road access is available, or can be developed at reasonable cost, for transporting the turbines and other equipment. Blades of large wind turbines can be up to 50 m in length and so clearly can pose difficulties for transport on minor roads. For a large wind farm, the heaviest piece of equipment is likely to be the main transformer if a substation is located at the site.

The local electricity utility should be able to provide information on the amount of generation which the distribution network can accept, although for a first approximation it may be useful to consider rules-of-thumb as indicated in Table 10.4, Chapter 10. Such rules-of-thumb give an approximate indication only but do serve to highlight those sites where the electrical connection will require long extensions of the power network with associated high cost, environmental impact and delay.

The initial technical assessment will be accompanied by a review of the main environmental considerations. The most important constraints include avoidance of National Parks or other areas designated as being of particular environmental or amenity value and ensuring that no turbine is located so close to domestic dwellings that a nuisance will be caused (e.g. by noise, visual domination or light shadow flicker). Table 9.2 shows some typical separation distances for medium sized wind turbines.

A preliminary assessment of visual effects is also required considering the visibility of the wind farm particularly from important public viewpoints. If, within the wind farm perimeter, there are areas of particular ecological value, due to flora or fauna, then these need to be avoided as well as any locations of particular archaeological or historical interest. Communication systems (e.g. microwave links, TV, radar or radio) may be adversely affected by wind turbines and this needs to be considered at an early stage.

In parallel with the technical and environmental assessments it is normal to open discussion with the local civic or planning/permitting authorities to identify and agree the major potential issues which will need to be addressed in more detail if the project development is to continue. These are recorded in the scoping report of the Environmental Impact Assessment.

9.1.2 Project feasibility assessment

Once a potential site has been identified then more detailed, and expensive, investigations are required in order to confirm the feasibility of the project. The wind farm energy output, and hence the financial viability of the scheme, will be very sensitive to the wind speed seen by

the turbines over the life of the project. Hence, it is not generally considered acceptable in complex terrain to rely on the estimates of wind speed made during the initial site selection but to use the Measure-Correlate-Predict (MCP) technique to establish a prediction of the long-term wind resource (Derrick, 1993; EWEA, 2009).

9.1.3 The Measure-Correlate-Predict (MCP) technique

The MCP approach is based on taking a series of measurements of wind speed at the wind farm site and correlating them with simultaneous wind speed measurements made at a nearby meteorological station. The averaging periods of the measured site data is chosen to be the same as that of the meteorological station data. In its simplest implementation, linear regression is used to establish a relationship between the measured site wind speed and the long-term meteorological wind speed data of the form:

$$U_{\text{site}} = a + bU_{\text{long-term}} \quad (9.3)$$

Coefficients are calculated for the twelve 30° directional sectors and the correction for the site applied to the long-term data record of the meteorological station. This allows the long-term wind speed record held by the meteorological station to be used to estimate what the wind speed at the wind farm site would have been over the last, say, 20 years. It is then assumed that the site long-term wind speed is represented by this estimate which is used as a prediction of the wind speed during the life of the project. Thus, MCP requires the installation of a mast at the wind farm site on which are mounted anemometers and a wind vane. If possible one anemometer is mounted at the hub height of the proposed wind turbines with others lower to allow wind shear to be measured. Measurements are made over at least a six-month period, although clearly the more data obtained the more confidence there will be in the result, and correlated with the measurements made concurrently at the meteorological station.

There are a number of difficulties of using MCP (Landberg and Mortensen, 1993) including:

- With modern wind turbines, high site meteorological masts are necessary and these may themselves require planning permission.
- There may not be a suitable meteorological station nearby (within say 50–100 km) or with a similar exposure and wind climate.
- The data obtained from the meteorological station may not always be of good quality and may include gaps. Therefore, it may be time consuming to ensure that it is properly correlated with the site data.
- It is based on the assumption that the previous long-term record provides a good estimate of the wind resource over the lifetime of the wind farm.

A number of variations in how the MCP method can be applied are described in (EWEA, 2009). If the site and long-term wind speeds are well correlated these different approaches to their correlation are of limited significance. Errors due to the location of the anemometers on the site meteorological mast and difficulties with the long-term record from the meteorological stations(s) are likely to be more important.

9.1.4 Micrositing

The MCP technique is now well established and specially designed site data loggers, temporary meteorological masts and software programs for data processing are commercially available. The estimate of the long-term wind speeds obtained from MCP may then be used together with a wind farm design software package to investigate the performance of a number of potential turbine layouts. These sophisticated programs take the wind distribution data and combine it with topographic wind speed variations and the effect of the wakes of the other wind turbines to generate the energy yield of any particular turbine layout. Two models commonly used to calculate the effect of topology and surface roughness of the site and surrounding area are the Wind Atlas Analysis and Application Program (WAsP) and MS-Micro/3 (a development of the MS3DJH/3R model). (Walmesley *et al.*, 1982, 1986). With increasing computer power, the use of computational fluid dynamic techniques to calculate wind flows over a site is becoming more common. Other constraints such as turbine separation, terrain slope, wind turbine noise, radar and land ownership boundaries may also be applied. Optimisation techniques are then used to optimise the layout of the turbines for maximum energy yield of the site taking into account local wind speeds and wake effects. The packages also have visualisation facilities to generate zones of visual impact, views of the wind farm either as wire frames and photomontages and even use 3D virtual reality techniques. Figure 9.2 shows an energy map of a prospective wind farm site created with wind farm design software. The areas on the top of the hills show the highest energy density.

9.1.5 Site investigations

At the same time as wind speed data is being collected more detailed investigations of the proposed site may also be undertaken. These include a careful assessment of existing land use and how best the wind farm may be integrated with, for example, agricultural operations. The ground conditions at the site also need to be investigated to ensure that the turbine foundations, access roads and construction areas can be provided at reasonable cost. Local ground conditions may influence the position of turbines in order to reduce foundation costs. It may also be important to undertake a hydrological study to determine whether spring water supplies are taken from the wind farm site and if the proposed foundations or trenches will cause disruption of the ground water flow. More detailed investigations of the site access requirements will include assessment of bend radii, width gradient and any weight restrictions on approach roads. Discussions are also likely to continue with the local electricity utility concerning the connection to the distribution network and the export of the wind farm power.

The planning application will require the preparation of an Environmental Statement and the scope of this is generally agreed, in writing, with the civic authorities during the project feasibility assessment.

9.1.6 Public consultation

Prior to the erection of the site anemometer masts the wind farm developer may wish to initiate some form of informal public consultation. This is likely to involve local community organisations, environmental societies and wildlife trusts. It may also be appropriate to keep local politicians informed. Of course, the erection of meteorological masts does not necessarily imply that the wind farm will be constructed but, as they are highly visible structures, careful consultation is required to ensure that unnecessary public concern is allayed.

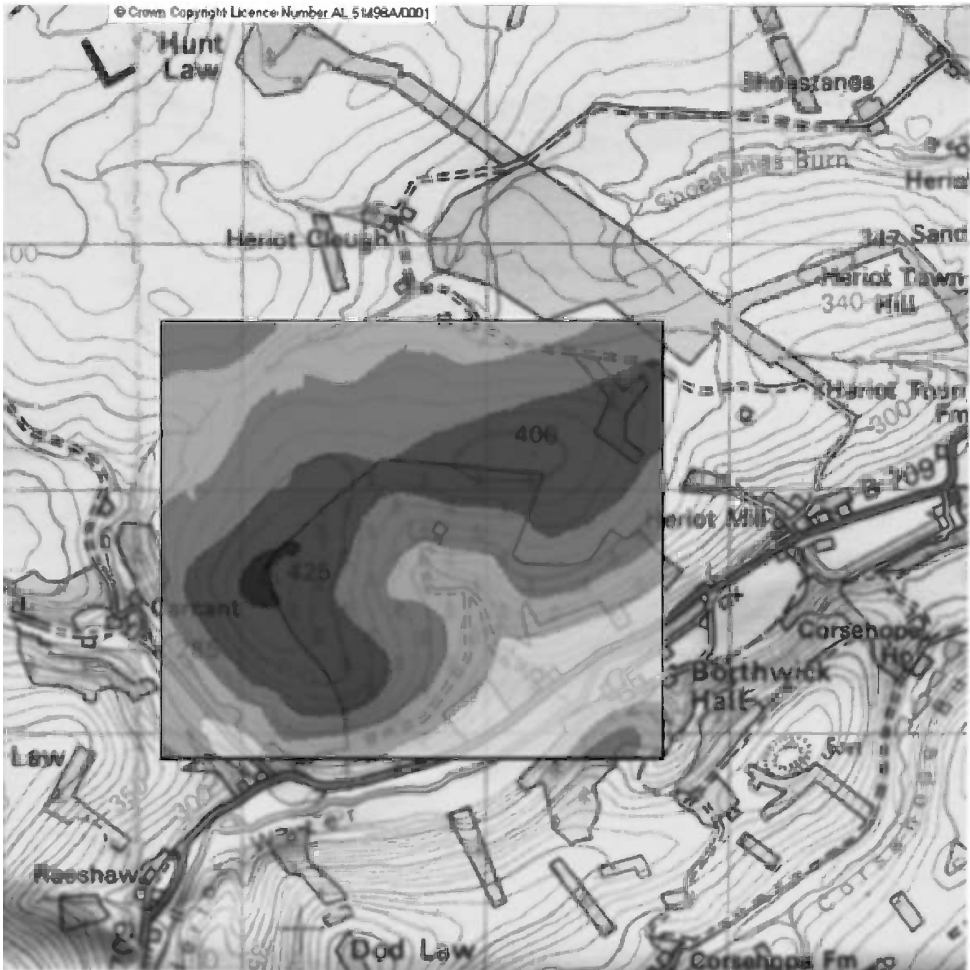


Figure 9.2 Example of Energy Map of a prospective wind farm site created with wind farm design software. Reproduced by permission of GL Garrad Hassan. Based upon the Ordnance Survey mapping (c) Crown Copyright and/or database right. All rights reserved. Licence number 100050309

9.1.7 Preparation and submission of the planning application

Wind farms are recognised as having significant environmental impact and it is usual for an Environmental Statement to be required as a major part of the application for planning permission. The preparation of the Environmental Statement is an expensive and time-consuming

undertaking and usually requires the assistance of various specialists. The purpose of a wind farm Environmental Statement may be summarised as to:

- Describe the physical characteristics of the wind turbines and their land-use requirements.
- Establish the environmental character of the proposed site and the surrounding area.
- Predict the environmental impact of the wind farm.
- Describe measures which will be taken to mitigate any adverse impact.
- Explain the need for the wind farm and provide details to allow the planning authority and general public to make a decision on the application.

Topics covered in the Environmental Statement will typically include (BWEA, 1994):

Policy framework – the application is placed in the context of national and regional policy.

Site selection – the choice of the particular site that has been selected is justified.

Designated areas – the potential impact of the wind farm on any designated areas (e.g. National Parks) is evaluated.

Visual and landscape assessment – This is generally the most important consideration and is certainly the most open to subjective judgement. Hence, it is usual to employ a professional consultancy to prepare the assessment. The main techniques which will be used include: Zones of Theoretical Visibility ZTV, (known formerly as Zones of Visual Influence, ZVI) to indicate where the wind farm will be visible from, wireframe analysis which shows the location of the turbines from particular views and production of photomontage which are computer generated images overlaid on a photograph of the site.

Noise assessment – After visual impact, noise is likely to be the next most important topic. Hence, predictions of the sound produced by the proposed development are required with special attention being paid to the nearest dwellings in each direction. It may be necessary to establish the background noise at the dwellings by a series of measurements so that realistic assessments can be made after the wind farm is in operation.

Ecological assessment – The impact of the wind farm, including its construction, on the local flora and fauna needs to be considered. This may well require site surveys at particular seasons of the year.

Archaeological and historical assessment – This is an extension of the investigation undertaken during the site selection.

Hydrological assessment – Depending on the site it may be necessary to evaluate the impact of the project on water courses and supplies.

Interference with telecommunication systems – Although wind turbines do cause some interference with television transmission this is normally only a local effect and can usually be remedied at modest cost. Any interference with major point-to-point communication facilities (e.g. microwave systems) is likely to be a much more significant issue.

Aircraft safety and interference with radar – The proximity to airfields or military training areas needs to be considered carefully. The effects of wind farms on aircraft radar, both

civil and military, has become a most important issue in several countries and requires early attention.

Safety – An assessment is required of the safety of the site including the structural integrity of the turbines. Particular local issues may include highway safety and shadow flicker.

Traffic management and access construction – The Environmental Statement addresses all phases of the project and so both the access tracks and the increase in vehicle movements on the public roads need to be considered.

Electrical connection – There may be significant environmental impact associated with the electrical connection (e.g. the construction of a substation and new circuit). Any requirement to place long high voltage circuits underground will be very expensive.

Economic effects on the local economy, global environmental benefits – It is common to emphasise the benefit that the wind farm will bring both to the local economy and to reduction in gaseous emissions.

Decommissioning – The assessment should also include proposals for the decommissioning of the wind farm and the removal of the turbines at the end of the project. Decommissioning measures are likely to involve the removal of all equipment which is above ground and restoration of the surface of all areas.

Mitigating measures – It is obvious that the wind farm will have an impact on the local environment and so this section details the steps that are proposed to mitigate and adverse effects. This is likely to emphasise the attempts that have been made to minimise visual intrusion and control noise.

Non-technical summary – Finally a non-technical summary is required and this may be widely distributed to local residents.

9.2 Landscape and visual impact assessment

Modern wind turbines are large structures, sometimes more than 100 m to the blade tip, and must be sited in exposed locations of high mean wind speed to operate effectively. Within a wind farm, the individual turbines are spaced at up to six rotor diameters and hence large wind farms are extensive. There is generally a trade-off between energy yield and visibility of wind turbines and so landscape and visual impact assessment is a key aspect of the Environmental Statement required for any wind farm development. The assessment of landscape and visual effects of a wind farm is normally undertaken by a professional consultant specialising in such work who will attempt to quantify the impacts but it is generally recognised that a degree of subjective interpretation is required.

The assessment process is, of course, iterative and will influence the design and layout of the wind farm. However, it may be divided roughly into a number of areas:

- Landscape character assessment (including landscape policy and designation).
- Design and mitigation.
- Assessment of impacts (including visibility and viewpoint analysis).
- Shadow flicker.



Figure 9.3 Wind farm of six 660 kW turbines in flat terrain. Reproduced by permission of Cumbria Wind Farms, Ltd. Paul Carter

The landscapes within which wind farms are constructed vary widely and the turbine layout is chosen to take account of the landscape character as well as wind conditions. Thus, in Figure 9.3 the turbines are located along the field boundaries while linear arrangements are shown in Figures 9.4–9.6 either along ridges or the coast line.

It is also essential to consider the sociological aspects of the development. An individual's perception of a wind farm development will be determined not only by the physical parameters (e.g. wind turbine size, number, colour etc.) but also by his or her opinion of wind energy as part of the energy supply (Taylor and Rand, 1991; Devine-Wright, 2005).

9.2.1 Landscape character assessment

The fundamental step in minimising the visual impact of a wind farm is to identify an appropriate site and ensure that the proposed development is in harmony with the location. Many exposed upland areas are likely to be of high amenity and have been designated as areas of significant landscape value or even as National Parks.

Development of wind farms within a National Park is likely to be extremely difficult and it is also important to recognise that any view of the site from inside a specially designated area may be considered to be important by the planning authorities.



Figure 9.4 Wind farm of 600 kW turbines, Tarifa, Spain. Reproduced by permission of NEG MICON



Figure 9.5 Wind farm of 700 kW wind turbines located along coast. Reproduced by permission of PowerGen Renewables/Wind Propsect Ltd.



Figure 9.6 Cameron Ridge in California, USA. Reproduced by permission of Renewable Energy Systems Ltd.

Stanton (1994) suggests some characteristics of the wind farm image which are considered to be desirable. The development should be simple, logical and avoid visual confusion. Although one landscape type is no more appropriate for a wind farm development than another this author considers that their suitability for different types of development varies greatly. Five different landscape zones are identified and discussed for the most appropriate type of development. Flat agricultural land is considered suitable for either a small number of wind turbines or large wind farms of similar regularly spaced machines. Coastal areas are considered appropriate for large numbers of wind turbines but the development should relate to the linear quality of the coastal space. There is some opportunity to install a small number of wind turbines in industrial areas. In areas of high aesthetic value, such as mountains and moorland the turbines should be arranged either along ridge lines or as a grid layout within flat land. Finally in areas of variable and rolling relief it is proposed that only small wind turbines or single wind turbines are appropriate. Such views are, of course, open to debate but

they do illustrate that it is essential to consider the character of the landscape before a wind farm development is proposed.

Cumulative effects are becoming increasingly important and the impact of more than one wind farm being visible from a viewpoint can be an important consideration. Local features such as buildings and hedges, which restrict views, may mean that a landscape has the capability to accommodate additional wind turbines but detailed investigation is required to establish this.

9.2.2 Design and mitigation

Wind turbine designers have recognised for some years that the overall form of the structure of a large wind turbine has to be pleasing and this aspect of industrial design is considered early in the development of a new machine. It is now generally recognised that for aesthetic reasons three-bladed turbines are preferred (see Section 6.5.6, Chapter 6). Two-bladed rotors sometimes give the illusion of varying speed of rotation, which can be disconcerting. In addition, for a similar swept area, a two-bladed rotor will operate faster than one with three blades. There is considerable evidence (Taylor and Rand, 1991; Gipe, 1995) that a slower speed of rotation is more relaxing to the eye. This effect works to the advantage of the large modern wind turbines, which operate typically at less than 30 rpm. There is some speculation that, in the future, very large wind turbines designed specifically for remote offshore installation, where visual impact and blade noise is less important, may revert to two-bladed rotors to realise the engineering benefits of this arrangement.

Lattice steel towers are now rarely used for wind turbines. There are clearly engineering benefits associated with lattice towers particularly in terms of the costs of the foundations (see Section 7.10.4, Chapter 7); although the upper, tapered section of the tower must be such as to provide adequate clearance for the blades. From a distance and in certain light conditions lattice towers can disappear leaving only the rotors visible and this effect is generally thought to be undesirable. In Europe, only solid tubular towers are generally considered to be acceptable. Tower heights can vary greatly in response to both wind conditions and planning constraints. In parts of Germany, very high towers (more than 80–100 m high) are used in order to maximise energy generation. Conversely, in western parts of the UK, which have high mean wind speeds, tower heights have been reduced in order to minimise the area from which the wind farm is visible.

The appearance of the turbines can be influenced to some extent by their colour. In the upland sites of the UK, the turbines will generally be seen against the sky and so an off-white or mid-grey tone is generally considered to be appropriate. Where the wind farm is seen against other backgrounds a colour to blend in with the ground conditions may be more suitable. There is general agreement that the outer gel-coat of the blades should be a matt or semi-matt finish to minimise reflections. In the early development of wind energy in the USA there were wind farms that used a variety of wind turbines sometimes of quite different design (i.e. with differing numbers of blades, direction of rotation, tower types, etc.). It is unlikely that such developments of large wind turbines would now be acceptable.

The layout and design of the wind farm is important in determining if planning consent is to be received. The preliminary wind farm layout may be determined by engineering considerations (e.g. local wind speeds, turbine separation, noise, geo-technical considerations, etc.) but will then be modified to take account of visual and landscape impacts. Usually the

developer will wish to maximise the number of turbines on the site, and hence the revenue, within all the environmental constraints. On open, flat land the turbines are often arranged in a regular layout in order to provide a simple and logical visual image with maximum power output. Alternatively on hill sites, or where there are significant hedges or field boundaries, the turbine layout is arranged around these features. When viewed from up to 1–2 km, wind turbines are considered to dominate the field of vision and so it is desirable that views within this distance can be minimised (e.g. by moving turbines to make use of local screening features or by tree planting). When turbines are seen one behind another there is an increase in visual confusion and this ‘stacking’ effect is considered to be undesirable. Some viewpoints are likely to be particularly important and it may be appropriate to arrange the turbines so that these views of the wind farm are as clear and uncluttered as possible.

In addition to the wind turbines, there are a number of associated ancillary structures required, including the wind farm meteorological mast(s). With smaller wind turbines, the local transformers may be located adjacent to the towers, often in an enclosure to provide protection against the weather and vandalism. However, the towers of many large turbines are wide enough to accommodate the transformers inside and this obviously reduces visual clutter on the wind farm. There is often a requirement for a main wind farm substation and some form of local control building. Engineering considerations would indicate that this substation should be located in the middle of the wind farm but, in order to reduce visual impact, it is often located some distance away where it gives minimal visual intrusion. Within European wind farms all power collection circuits are underground and it may also be appropriate, if rather expensive, to use underground cable to make the final connection to the local utility system. Roads are required within the wind farm for construction and it is a common requirement of the planning consent that the hard surface is removed after commissioning. This of course can lead to very considerable expense if the road has to be reinstated to allow a large crane to be used for maintenance or repair.

9.2.3 Assessment of impact

A major part of the Environmental Statement is the assessment of landscape and visual impact. Landscape and visual impact assessment of wind farms is now an important specialist subject with its own specialist practitioners and literature, for example, University of Newcastle (2002), Horner + maclellan and Envision (2006). Two main techniques are used: (1) visibility analysis using Zones of Theoretical Visibility (ZTV) – formerly known as Zones of Visual Influence (ZVI) and (2) viewpoint analysis using wire frames and photomontages.

Zones of theoretical visibility show those areas of the surrounding country, usually up to 10–20 km radius, from which a wind turbine, or any part of a wind turbine, in a wind farm is visible. The ZTV is generated using computer methods based on a digital terrain model and shows how the local topology will influence the visibility of the wind farm. Usually ZTV techniques ignore local landscape features such as screening from trees and buildings. Also, weather conditions are not considered and clear visibility is assumed. Figure 9.7 shows an example of a ZTV generated using commercially available wind farm design software. The cumulative impact of a number of wind farms may be calculated in a similar manner.

Viewpoint analysis is based on selecting a number of important locations, from which the wind farm is visible and applying professional judgement using quantitative criteria to assess the visual impact. The viewpoints are selected in consultation with the civic planning authorities and for a large wind farm more than 20 locations may be chosen. Although

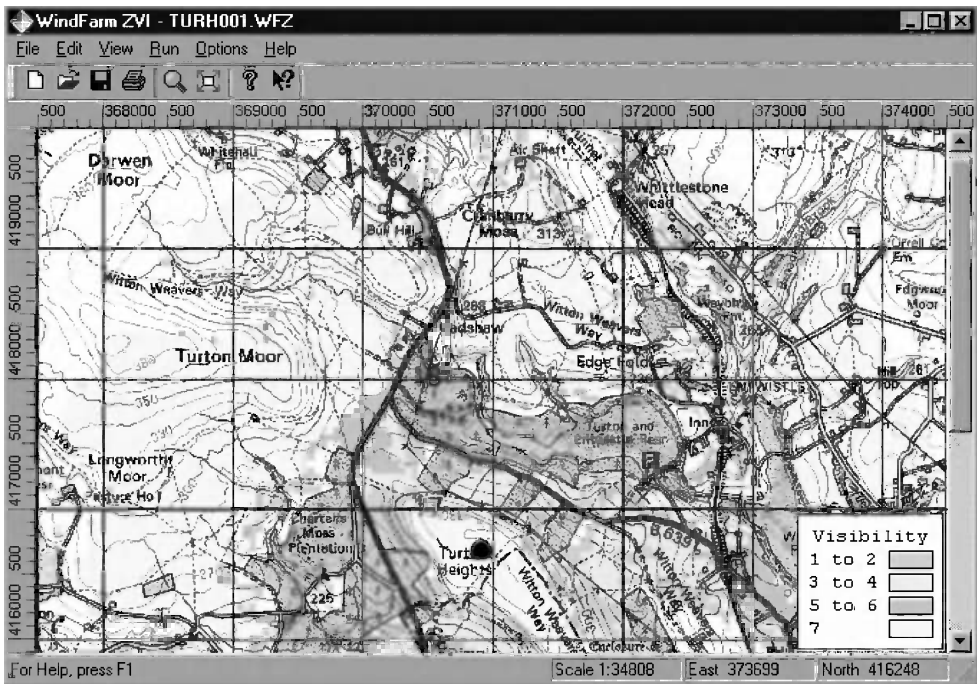


Figure 9.7 Example of Zone of Theoretical Visibility (Zone of Visual Influence) of a wind farm. Reproduced by permission of ReSoft, based upon the Ordnance Survey mapping (c) Crown Copyright and/or database right. All rights reserved. Licence number 100050309

approaches vary, the assessment may involve consideration of two aspects: (1) the sensitivity of the viewpoint and (2) the magnitude of the change of view. Thus, for example, a viewpoint where the land use is residential or has high recreational value (e.g. a scenic viewpoint inside a National Park) may have a 'high' sensitivity. Conversely, a viewpoint, which is used only for employment (e.g. a local industrial estate), might be considered to have a 'low' sensitivity. The magnitude of the impact can be described in a similar manner depending, for example, on the number of turbines visible, the distance to the wind farm and the prominence of the development. The overall significance of the impact is then assessed, again using quantitative terminology (such as substantial, moderate, slight, negligible, etc.), combining these factors. Where a substantial impact is identified, acceptability will depend on whether it is considered that the wind farm will have a detrimental effect on the landscape quality.

Figure 9.8 shows a wireframe image generated from a viewpoint from which three wind farms are visible. As wind energy develops, the planning authorities are becoming increasingly concerned about such cumulative impacts. Figure 9.9 shows a photomontage. Both of these images were generated using wind farm design software. Wireframe representations provide an accurate impression of turbine position and scale while photomontages are the normal tool used to give an overall impression of the visual effect of a wind farm. Video-montages are used in order to give an impression of the movement of turbine blades as well as 3-D 'virtual reality' techniques.

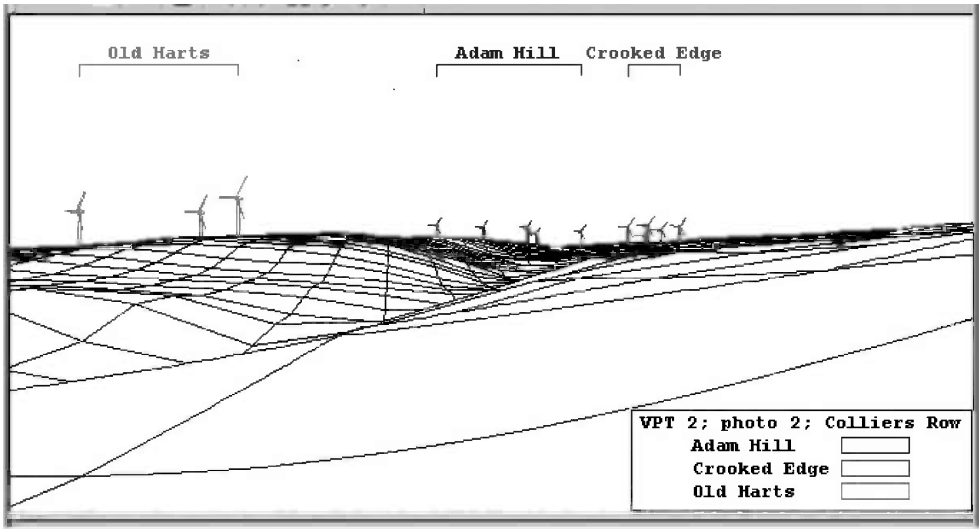


Figure 9.8 Example of wire-frame showing visibility of three wind farms. Reproduced by permission of ReSoft

9.2.4 Shadow flicker

Shadow flicker is the term used to describe the stroboscopic effect of the shadows cast by rotating blades of wind turbines when the sun is behind them. The shadow can create a disturbance to people inside buildings exposed to such light passing through a narrow window (EWEA, 2009; AWEA 2008). The frequencies that can cause disturbance are between 2.5–20 Hz.



Figure 9.9 Example of photomontage. Reproduced by permission of ReSoft

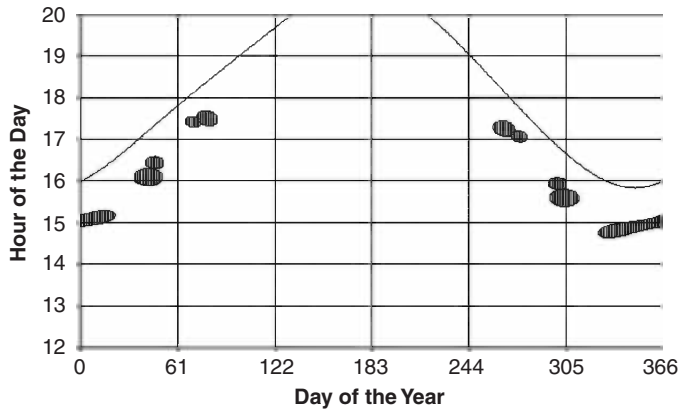


Figure 9.10 Example of shadow flicker prediction – continuous line shows time of sunset. Reproduced by permission of ReSoft

The effect on the human eye-brain is the same as that caused by changes in intensity of an incandescent electric light due to variations in network voltage from a wind turbine (see Section 10.2.1, Chapter 10). In the case of shadow flicker the main concern is variations in light at frequencies of 2.5–3 Hz which have been shown to cause anomalous EEG (electroencephalogram) reactions in some sufferers of epilepsy. Higher frequencies (15–20 Hz) may even lead to epileptic convulsions. Of the general population, some 10% of all adults and 15–30% of children are disturbed to some extent by light variations at these frequencies (Verkuijlen and Westra, 1984).

Large modern three-bladed wind turbines will rotate at less than 30–35 rpm, giving blade-passing frequencies of 1.5–1.75 Hz, which is below the critical frequency of 2.5 Hz. The wind farm SCADA system can be arranged to stop some of the wind turbines temporarily at times that a shadow flicker nuisance may be created. Shadow flicker, noise constraints and avoidance visual domination all need to be addressed when considering how close a turbine should be located to a dwelling. Figure 9.10 shows the calculation of when shadow flicker might cause disturbance in a dwelling close to a wind turbine. It may be seen that this occurs during late afternoon (from around 15:00–17:30 hrs) of the spring and autumn.

9.2.5 Sociological aspects

There are a number of computer based tools available for quantifying visual effects and landscape architects and planners have developed techniques to place quantitative measures on visual impact using professional judgement. However, public attitudes, which ultimately determine whether a wind farm may be constructed, are influenced by many more complex factors. Public attitudes to wind farms have been studied on a number of occasions (e.g. ETSU reports W/13/00300/REP, 1993 and W/13/00354/038/REP, 1994) and Gipe discusses this subject in considerable detail. In general, the large majority of people approve of wind farms after they have been constructed although a significant minority remains opposed to them. In particular, there is the difficult issue that some local residents consider they are paying a high cost for a benefit, either financial or environmental, which accrues to others.

The financial benefits may be shared with the community in a number of ways, including by the development of co-operative or community owned wind farms, while the environmental issue has to be addressed by consultation. It is suggested that stationary wind turbines are less acceptable than those rotating and so maintaining high availability with a low cut-in wind speed is likely to improve public perception.

9.3 Noise

Noise from wind turbines is often perceived as one of the more significant environmental impacts (Wagner *et al.*, 1996). During the early development of wind energy, in the 1980s, some turbines were rather noisy and this led to justified complaints from those living close to them. However, since then, there has been very considerable development both in techniques for reducing noise from wind turbines and in predicting the noise nuisance a wind farm will create.

It is common for any wind farm Environmental Statement to be accompanied by details of the proposed turbines and predicted noise levels. This is likely to include:

- Predicted noise levels at specific properties close to the wind farm over the most critical range of wind speeds.
- Measured background noise levels at the properties at these wind speeds.
- A scale map showing the proposed wind turbines, the prevailing wind conditions and nearby existing developments.
- Results of independent measurements of noise emission from the proposed wind turbines including the sound power and narrow band frequency spectrum.

IEC 61400–11 (2006) describes how tests to determine the noise emissions from a wind turbine may be conducted and IEC 61400–14 (2005) details how the results of these tests should be declared so that they are likely to be representative of a number of turbines of the same design. These test results are provided by the wind turbine supplier to the wind farm noise consultant to undertake part of the Environmental Impact Assessment. Figure 9.11 shows an example of predicted sound pressure levels from a small wind farm. Noise contours of 40, 50 and 60 dBA are superimposed on a map of the wind farm site and adjacent dwellings.

9.3.1 Terminology and basic concepts

Two distinctly different measures are used to describe wind turbine noise. These are the *sound power level* L_W of the source (i.e. the wind turbine) and the *sound pressure level* L_P at a location. Because of the response of the human ear, a logarithmic scale is used based on reference levels that correspond to the limit of hearing. The units of both L_P and L_W are the decibel (dB).

A noise source is described in terms of its sound power level, L_W

$$L_W = 10 \log_{10} \left(\frac{W}{W_0} \right) \quad (9.4)$$

where W is the total sound power level emitted from the source (in Watts) and W_0 is a reference value of 10^{-12} W.

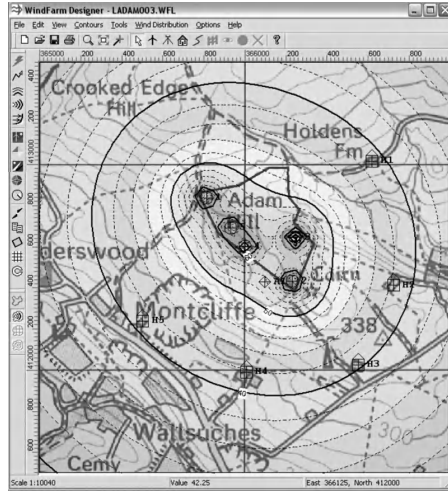


Figure 9.11 Noise contours around a small wind farm. Reproduced by permission of ReSoft, based upon the Ordnance Survey mapping (c) Crown Copyright and/or database right. All rights reserved. Licence number 100050309

The sound pressure level L_P is defined as

$$L_P = 10 \log_{10} \left(\frac{P^2}{P_0^2} \right) \quad (9.5)$$

where P is the RMS value of the sound pressure and P_0 is a reference value of $2 \times 10^{-5} \text{ Pa}$.

By simple manipulation it may be seen that the addition of n sound pressure levels (expressed in dB) is carried out as shown:

$$L_{Pn} = 10 \log_{10} \sum_{j=1}^{j=n} 10^{\frac{L_{P(j)}}{10}} \quad (9.6)$$

Thus, adding two sound pressure levels of the same magnitude results in an increase of 3 dB. Sound power levels may be added in a similar manner.

Table 9.3 gives an indication of the typical range of sound pressure levels.

The human ear is capable of detecting sounds between 20 Hz–20 kHz and spectral analysis is typically undertaken over this range. A narrow-band spectrum, with a defined bandwidth of measurement, gives the fullest information of the signal and may be used to detect particular tones. However, it is conventional to use octave and $\frac{1}{3}$ -octave bands for broadband analysis. The upper frequency of an octave band is twice that of the lower frequency while for the $\frac{1}{3}$ -octave band the upper frequency is $\sqrt[3]{2}$ times the lower frequency.

It is common to weight the measurements to reflect the response of the human ear with frequency. This is done by applying the so-called A-weighted filter. Measurements made with this filter are referred to as dBA or dB(A).

Table 9.4 shows the centre frequencies of the octave bands together with the A-weighting in dB. It may be seen that frequencies below 250 Hz and above 16 kHz are heavily attenuated.

Table 9.3 Examples of sound pressure levels

Example	Sound pressure level dB(A)
Threshold of hearing	0
Rural night time background	20–40
In a busy general office	60
Inside a factory	80–100
Jet aircraft at 100 m	120
Wind farm at 350 m	35–45

An equivalent sound pressure level $L_{eq,T}$ is the value of a continuous steady sound that, within the specified time interval (T) has the same mean square sound pressure level as the sound under consideration which varies with time

$$L_{eq,T} = 10 \log_{10} \left(\frac{1}{T} \int_0^T \frac{P^2}{P_0^2} dt \right) \tag{9.7}$$

A similar calculation may be undertaken using A-weighted values to give $L_{Aeq,T}$

$$L_{Aeq,T} = 10 \log_{10} \left(\frac{1}{T} \int_0^T \frac{P_A^2}{P_0^2} dt \right) \tag{9.8}$$

where $L_{Aeq,T}$ is the equivalent continuous A-weighted sound pressure level determined over time T and P_A is the instantaneous A-weighted sound pressure.

The exceedance level, L_{A90} , is defined as the A-weighted sound pressure level which is exceeded for 90% of the time. Some planning authorities prefer the use of the L_{A90} sound pressure level, particularly for measurements of background noise, as the L_{eq} measurement may be heavily influenced by short-term effects such as passing aircraft or traffic. A wind

Table 9.4 Centre frequency of octave bands and A-weighting

Octave band centre frequency (Hz)	A-weighting (dB)
16	–56.7
31.5	–39.4
63	–26.2
125	–16.1
250	–8.6
500	–3.2
1000	0.0
2000	1.2
4000	1.0
8000	–1.1
16000	–6.6

farm is a fairly constant source of noise and its contribution to the L_{A90} sound pressure level (measured over a ten-minute period) is likely to be some 1.5–2.5 dB(A) less than to the ten-minute L_{eq} value (ETSU, 1997b).

The sound intensity I is the power transmitted through a unit area, A ,

$$I = \frac{W}{A} \quad (9.9)$$

and far from the source of sound with a uniform flux

$$I = \frac{P^2}{Z_0} \quad (9.10)$$

where P is the RMS value of the sound pressure level, and Z_0 is the characteristic acoustic impedance.

Choosing a suitable value for I_{ref} (10^{-12} W/m²), the sound pressure level may be expressed in terms of sound intensity as

$$L_P = 10 \log_{10} \left(\frac{I}{I_{\text{ref}}} \right) \quad (9.11)$$

For spherical spreading

$$I = \frac{W}{4\pi r^2} \quad (9.12)$$

where r is the distance from the source.

Hence, under conditions of ideal spherical spreading

$$L_P = 10 \log_{10} \left(\frac{W}{4\pi r^2 10^{-12}} \right) = L_W - 10 \log_{10}(4\pi r^2) \quad (9.13)$$

and similarly, if hemi-spherical spreading is assumed

$$L_P = 10 \log_{10} \left(\frac{W}{2\pi r^2 10^{-12}} \right) = L_W - 10 \log_{10}(2\pi r^2) \quad (9.14)$$

Sound pressure level, from a point source, decays with distance according to the inverse square law under both spherical and hemi-spherical spreading assumptions. Hence, for each doubling of distance the sound pressure level is reduced by 6 dB.

For a line source of noise the sound pressure level is given by

$$L_P = 10 \log_{10} \left(\frac{L_{wl}}{2\pi r 10^{-12}} \right) = L_{wl} - 10 \log_{10}(2\pi r) \quad (9.15)$$

(L_{wl} is the sound power level per unit length of the source). For cylindrical spreading the decay is only proportional to distance, resulting in a 3 dB reduction of sound pressure level for each doubling of distance perpendicular to the line source.

Table 9.5 Sound power levels of mechanical noise of a 2 MW experimental wind turbine (after Pinder, 1992)

Element	Sound power level dB(A)	Air-borne or structure-borne
Gearbox	97.2	Structure-borne
Gearbox	84.2	Air-borne
Generator	84.2	Air-borne
Hub	89.2	Structure-borne
Blades	91.2	Structure-borne
Tower	71.2	Structure-borne
Auxiliaries	76.2	Air-borne

9.3.2 Wind turbine noise

Noise from wind turbines may be divided into

- mechanical; and
- aerodynamic.

Mechanical noise is generated mainly from the rotating machinery in the nacelle, particularly the gearbox and generator although there may also be contributions from cooling fans, auxiliary equipment (such as pumps and compressors) and the yaw system. Mechanical noise is often at an identifiable frequency or tone (e.g. caused by the meshing frequency of a stage of the gearbox). Noise containing discrete tones is more likely to lead to complaints and so attracts a 5 dB penalty in many noise standards. Mechanical noise may be air-borne (e.g. the cooling fan of an air-cooled generator) or transmitted through the structure (e.g. gearbox meshing which is transmitted through the gearbox casing, the nacelle bed-plate, the blades and the tower).

Pinder (1992) gives the values of sound power level shown in Table 9.5 for a 2 MW experimental wind turbine. It may be seen that in this example the gearbox is the dominant noise source through structure borne transmission.

Techniques to reduce the mechanical noise generated from wind turbines include careful design and machining of the gearbox, use of anti-vibration mountings and couplings to limit structure-borne noise, acoustic damping of the nacelle and liquid cooling of the generator. Small changes in the design of gearboxes (e.g. ratio and tooth shape) can have a significant effect on noise generation (IEC 61400–14, 2005).

Aerodynamic noise is due to a number of causes:

- Low frequency noise.
- Inflow turbulence noise.
- Airfoil self noise.

Low frequency noise is caused by changes in the wind speed experienced by the blades due to the presence of the tower or wind shear. Although this effect is very pronounced with downwind turbines it is also significant with upwind machines. The spectrum of the noise is dominated by blade passing frequency (typically up to 3 Hz) and its harmonics (typically

up to 150 Hz). It may be seen from Table 9.4 that the A-weighted filter heavily attenuates these frequencies and so this source does not make a major contribution to audible noise. However, it was reported that experimental, down-wind turbines, constructed in the 1980s excited low frequency vibrations in adjacent buildings. For upwind turbines increasing the clearance between the blades and tower will reduce this effect. Hubbard and Shephard (Spera, 1994) show interesting experimental evidence of low frequency noise from a large 80 m diameter downwind turbine and provide a comparison with a 90 m diameter upwind machine.

Inflow turbulence creates broadband noise as the blades interact with the eddies caused by atmospheric turbulence. It generates frequencies up to 1000 Hz which are perceived by an observer as a 'swishing' noise. Turbulent inflow noise is considered to be influenced by the blade velocity, airfoil section and turbulence intensity. Wagner *et al.* (1996) describe this phenomenon in some detail, remarking that it is not yet fully understood, and quote the results of one field experiment where the sound pressure level actually decreased with increasing turbulence intensity.

Airfoil self-noise is generated by the airfoil itself, even in steady, turbulent-free flow. It is typically broadband although imperfections in the blade surface may generate tonal components. The main types of airfoil self-noise include:

Trailing edge noise. This is perceived as a broadband swishing sound with frequencies in the range 750–2000 Hz. It is due to interaction of the turbulent boundary layer with the trailing edge of the blade. Trailing edge noise is a major source of higher frequency noise on wind turbines.

Tip noise. The literature is not clear as to whether three-dimensional tip effects are a major contributor to wind turbine noise. However, the majority of the blade noise, as well as the turbine power, emanates from the outer 25% of the blade and so there has been very considerable investigation of novel blade tips to reduce noise. Imperfections in the blade shape due to tip-brakes or other control surfaces are another potential source of noise.

Stall effects. Blade stall causes unsteady flow around the airfoil which can give rise to broadband sound radiation.

Blunt trailing edge noise. A blunt trailing edge can give rise to vortex shedding and tonal noise. It can be avoided by sharpening the trailing edge but this has implications for manufacturing and erection.

Surface imperfections. Surface imperfections such as those caused by damage during erection or due to lightning strikes can be a significant source of tonal noise.

The obvious approach to reducing aerodynamic noise is to lower the rotational speed of the rotor, although this may result in some loss of energy.

The ability to reduce noise in low-wind speed conditions is a major benefit of variable-speed or two-speed wind turbines. An informative annex of IEC 61400–14 (2005) gives a relationship between tip speed of a turbine and sound power level as:

$$L_W \approx (50 \text{ to } 60) \log V_{\text{tip}} \quad (9.16)$$

An alternative technique would be to reduce the angle of attack of the blade although again with a potential loss of energy.

Stiesdal and Kristensen (1993) provide a comprehensive description of noise control methods applied to a 300 kW stall regulated wind turbine. The gearbox was identified as an important source of tonal, mechanical noise. Control measures included detailed modifications to the design and manufacture of the gears and an additional insulating covering over the gearbox housing. Air-borne mechanical noise was minimised by total enclosure of the nacelle and careful design of sound baffles in the ventilation openings. Structure-borne noise was reduced by rubber mounting both the gearbox and generator and using a rubber coupling on the high-speed shaft. The long low-speed shaft (2 m) reduced the drive train vibrations transmitted to the rotor. Tip noise was minimised by the use of a 'tip torpedo' to control the tip-vortex while trailing edge noise was controlled by specifying a 1–2 mm trailing edge thickness. Stall noise was reduced by using a turbulator strip on the leading edge of the blades near the tip that initiated stall in a controlled manner. Overall these measures resulted in a reduction of the sound power level of 3–4 dB and the elimination of significant tones. The most important noise control methods were considered to be the tip modification, the controlled stall and the gearbox improvements. The measured sound power level of the improved wind turbine was 96 dB(A) at a wind speed of 8 m/s (measured at 10 m height). This was an impressive improvement as typical sound power levels of a 600 kW wind turbine at that time were up to 100 dB(A) at a wind speed of 8 m/s increasing by 0.5–1 dB(A) per m/s.

9.3.3 Measurement, prediction and assessment of wind farm noise

The sound power level of a wind turbine is normally determined by field experiments. These were originally specified in the, now superseded, IEA Recommended Practice and are described in international standard (IEC61400–11, 2006). Outdoor experiments are necessary because of the large size of modern wind turbines and the necessity to determine their noise performance during operation. The sound power level cannot be measured directly but is found from a series of measurements of sound pressure levels made around the turbine at various wind speeds from which the background sound pressure levels have been deducted. The method provides the apparent A-weighted sound power levels, one-third octave spectra and tonality of a single wind turbine at wind speeds between 6–10 m/s and, optionally, the directivity of the noise source.

The measurements are taken at a distance, R_0 , from the base of the tower

$$R_0 = H + \frac{D}{2} \quad (9.17)$$

where H is the hub height and D is the diameter of the rotor. This distance is a compromise to allow an adequate distance from the source but with minimum influence of the terrain, atmospheric conditions or wind-induced noise. The microphones are located on boards at ground level so that the effect of ground interference on tones may be evaluated. The reference microphone is located downwind of the wind turbine under test.

Simultaneous A-weighted sound pressure level measurements (more than 30 measurements each of not less than one minute duration) are taken with wind speed. All wind speeds are corrected to a reference height of 10 m. The preferred method of determining wind speed, when the turbine is operating, is from the electrical power output of the turbine and the power curve. The main measurement sound pressure level measurement is that of the downwind position while the other microphones located around the turbine are used for determining

directivity. Measurements are taken with and without the turbine operating over wind speeds between 6–10 m/s. A correction for background noise is applied as show in (9.18)

$$L_P = 10 \log_{10} \left(10^{\frac{L_{P+N}}{10}} - 10^{\frac{L_N}{10}} \right) \quad (9.18)$$

where

L_P is the sound pressure level of the wind turbine.

L_{P+N} is the sound pressure level of the wind turbine and the background sound.

L_N is the sound pressure level of the background.

At each of the wind speeds, the apparent A-weighted sound power level of the turbine is then calculated from

$$L_W = L_{PAeq} + 10 \log_{10} (4\pi R_i^2) - 6 \quad (9.19)$$

R_i is the slant distance from the microphone to the wind turbine hub.

It may be seen that the calculation of sound power level assumes spherical radiation of the noise from the hub of the turbine. The subtraction of 6 dB is to determine the free field sound pressure level from the measurements and to correct for the approximate pressure doubling that occurs with the microphone located on a reflecting board at ground level.

One of the Appendices of the, now superceded, IEA document provided a method to estimate the sound pressure level of a single turbine or group of turbines at a distance R provided that they are located in flat and open terrain.

The calculation was based on

$$L_P(R) = L_W - 10 \log_{10}(2\pi R^2) - \Delta L_a \quad (9.20)$$

The correction ΔL_a was for atmospheric absorption and calculated from $\Delta L_a = R\alpha$ where α was a coefficient for sound absorption in each octave band and R the distance to the turbine hub. An alternative approach is to use a similar equation to 9.20 but with α as 0.005 dB/m (as suggested in the Danish Statutory Order on Noise from Windmills) and L_W specified as a single, broadband sound power level.

If there are several wind turbines which influence the sound pressure level the individual contributions are calculated separately and summed using

$$L_{1+2+\dots} = 10 \log_{10} \left(10^{\frac{L_1}{10}} + 10^{\frac{L_2}{10}} + \dots \right) \quad (9.21)$$

The IEA Method for determining sound pressure levels at a point was based on hemi-spherical spreading with a correction for atmospheric absorption. The assumption of hemi-spherical spreading gives a reduction of 6 dB per doubling of distance. Under some conditions, particularly downwind, this may be an optimistic assumption and a reduction of 3 dB per doubling of distance is more realistic. The IEA method also ignored any effects of meteorological gradients (Wagner *et al.*, 1996). Under normal conditions, air temperature decreases with height and so the sound speed will decrease with increasing height and cause the path of the sound to curve upwards. However, under conditions of temperature inversion, for example as might prevail on cold winter nights, the temperature increases with height

Table 9.6 Noise limits for sound pressure levels LAeq, in dBA, in different European countries (after Gipe, 1995)

Country	Commercial	Mixed	Residential	Rural
Germany				
Day	65	60	55	50
Night	50	45	40	35
Netherlands				
Day		50	45	40
Night		40	35	30
Denmark			40	45

Note the definitions of location vary from country to country. Further details may be found in ETSU (1997b)

causing the sound to curve downwards. Wind speed will have a similar effect. In the down-wind direction the sound will be bent downwards while a shadow zone is formed upwind. The effect of the upwind shadow zone is more pronounced at higher frequencies.

A more comprehensive method for calculating the attenuation of sound outdoors is given in ISO 9613-2 (1996). This standard applies generally to outdoor noise propagation, not specifically from wind turbines, and calculates the sound pressure level at a point due to a source taking into account geometrical divergence, atmospheric absorption, ground effect, reflection from surfaces and screening by obstacles. The calculation is undertaken using octave frequency bands.

Permitted noise levels vary widely from country to country and even within countries according to local planning conditions. International practice was reviewed by the UK 'Working Group on Noise from Wind Turbines', (ETSU, 1997b). In Germany, the Netherlands and Denmark the limits are expressed in terms of a maximum permitted value of the sound pressure level at differing locations e.g. Table 9.6.

In contrast, the proposals of the UK Working Group (ETSU, 1997b) was to base the permitted noise level of a wind farm on an increase of 5 dB(A) of the $L_{A90,10 \text{ min}}$ sound pressure level above background noise. The 5 dB(A) limit was selected as being a reasonable compromise between protecting the internal and external environment while not unduly restricting the development of wind energy itself which has other environmental benefits. In addition, it is suggested that a limit of less than 5dB(A) would be difficult to monitor. BS4142 (1997), which is a general standard for industrial noise and may not be directly applicable to wind farms, states that a difference of +10 dB or more indicates that complaints are likely while a difference of around +5 dB is of marginal significance.

A lower fixed limit of 35–40 dB(A) during daytime and 43 dB(A) during night-time is also applied. The selection of the daytime limit of either 35 or 40 dB(A) is made by considering:

- The number of dwellings in the neighbourhood of the wind farm.
- The effect of noise limits on the number of kWh generated.
- The duration and level of exposure.

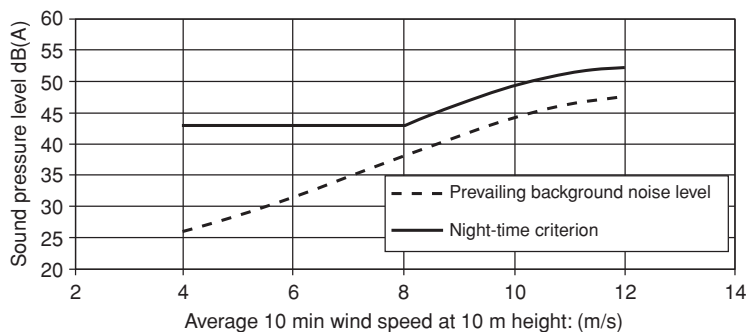


Figure 9.12 Example of noise criterion proposed by the UK Working Group on Noise from Wind Turbines. Night-time criterion (ETSU, 1997b). Reproduced by permission of ETSU on behalf of the DTI

The night-time lower limit of 43 dB(A) is derived from a 35 dB(A) sleep disturbance criterion, an allowance of 10 dB(A) for attenuation through an open window and with 2 dB subtracted for the use of $L_{A90,10\text{ min}}$ rather than $L_{Aeq,10\text{ min}}$. Examples of the noise criteria are shown in Figures 9.12 and 9.13. There is also a penalty for audible tones which rises to a maximum of 5 dB. The UK regulations governing noise from wind turbines are presently under review (2010) and may change in the near future.

9.4 Electromagnetic Interference

Wind turbines have the potential to interfere with electromagnetic signals that form part of a wide range of modern communication systems and so their siting requires careful assessment in respect of Electromagnetic Interference (EMI). In particular, wind energy developments often compete with radio systems for hilltops and other open sites that offer high energy outputs from wind farms and good propagation paths for communication signals. The types of system that may be affected by EMI, and their frequency of operation, include VHF radio systems (30 MHz–300MHz), UHF Television broadcasts (300 MHz–3GHz) and Microwave

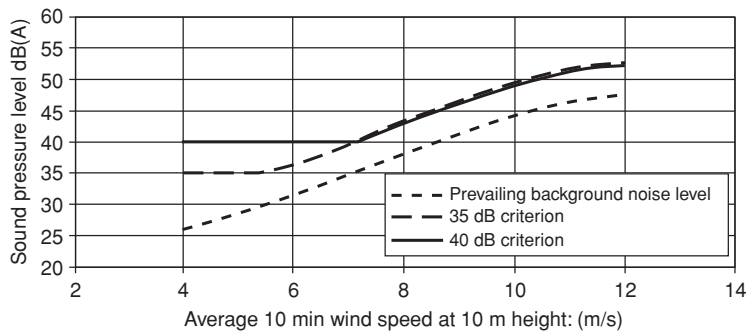


Figure 9.13 Example of noise criterion proposed by the UK Working Group on Noise from Wind Turbines. Daytime criterion (ETSU, 1997b). Reproduced by permission of ETSU on behalf of the DTI

links (1GHz–30GHz). The interaction of wind turbines with defence and civilian radar used for air traffic control is the subject of continuing investigation (ETSU, 2002). Concern over the effect of wind farms on civil and military radar systems has led to major difficulties and delays in a number of wind farms with, at one time, the UK not allowing the development of wind farms within a radius of 74 km of any air defence radar site.

Hall (1992) reported that tests on a fixed speed 400 kW wind turbine showed that no radio transmissions attributable to a wind turbine could be detected at 100 m. The electrical generator and associated control gear and electronics can produce radio frequency emissions but these may be minimised by appropriate suppression and screening at the generator. Rather than behave as an aerial the tubular tower had a substantial screening effect on all emissions. If the nacelle is metallic this will also screen the emissions from the nacelle itself. Although additional precautions are necessary with the power electronic converters of variable speed wind turbines, measures to deal with conducted or radiated EMI are standard with any power electronic equipment and electromagnetic emission from wind turbines is not a common problem.

Scattering is, however, an important electromagnetic interference mechanism associated with wind turbines. An object exposed to an electromagnetic wave disperses incident energy in all directions and it is this spatial distribution that is referred to as scattering.

The problem of the interaction of wind turbines and radio communication systems is complex as the scattering mechanisms due to wind turbines are not readily characterised and the signal may be modulated by the rotation of the blades. There is also a large, and increasing, number of types of radio systems with quite different requirements for their effective operation. ETSU, (1997c) indicates that it is expected that the electromagnetic properties of wind turbine rotors will be influenced by:

- rotor diameter and rotational speed;
- rotor surface area, planform and blade orientation including yaw angle;
- hub height;
- structural blade materials and surface finish;
- hub construction;
- surface contamination (including rain and ice); and
- internal metallic components including lightning protection.

ETSU (1997c) includes information of the practices of various radio authorities in determining if a wind farm is likely to give rise to EMI problems. For microwave fixed links it is important that there is a clear line of sight between the transmitter and receiver and that a proportion of the first Fresnel zone must be free from obstructions. The first Fresnel zone is an ellipsoid region of space that makes the major contribution to the signal received and within which component parts of the signal will be in phase (Figure 9.14). At least 60% of the first Fresnel zone must be free from obstruction in order to ensure ‘free space’ propagation conditions. The radius of the first Fresnel zone (R_F) is given by

$$R_F = \sqrt{\frac{\lambda d_1 d_2}{d_1 + d_2}} \quad (9.22)$$

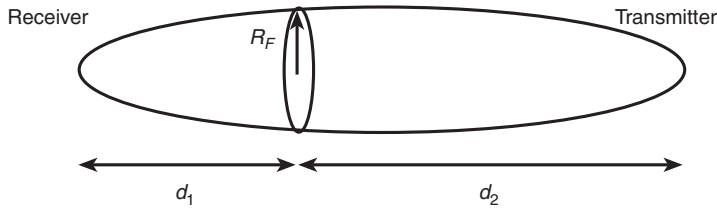


Figure 9.14 Illustration of first Fresnel zone (Fresnel Ellipsoid)

where

d_1 and d_2 are the distance from the two microwave terminals to the point of reference
is the wavelength

In practice it is usually considered necessary to be completely outside the first Fresnel zone of the link with an additional exclusion zone of either 200 m or 500 m in order to avoid unwanted reflections. These additional allowances for reflections are rather conservative rules-of-thumb and may be reduced following detailed studies.

For UHF television relay links, if the wind farm is outside 60° of the relative direction of the receive antenna no major problems are anticipated. If the site is within 15° – 60° then some problems may be anticipated but may be overcome with a customised antenna. If the site is within 15° major problems may be anticipated. Similar considerations apply to domestic TV reception. No problems are anticipated if the wind farm is outside 60° of the relative direction of the domestic receive antenna, a good quality antenna being required for 20° – 60° and a new source required if the wind farm is within 20° . It is emphasised by the radio authorities that each wind farm proposal needs to be considered individually.

9.4.1 Modelling and prediction of EMI from wind turbines

There are two fundamental interference mechanisms for EMI from wind turbines, back-scattering and forward-scattering (Moglia *et al.*, 1996). These are shown in Figure 9.15. Forward-scattering occurs when the wind turbine is located between the transmitter and receiver. The interference mechanism is one of scatter or refraction of the signal by the wind turbine and, for TV signals, it causes fading of the picture at the rotational speed of the blades. Back-scattering occurs when the turbine is located behind the receiver. This results in a time delay between the wanted signal and the reflected interference and gives rise to ghost or double images on a TV screen.

Moglia *et al.* (1996) (using the earlier work by van Kats and van Rees, 1989) provide an analysis of the electro-magnetic interference caused by a wind turbine.

The useful carrier signal received, C , is given by

$$C = P_T - A_{TR} + G_{TR} \quad (9.23)$$

where:

P_T is the transmitter power (dB).

A_{TR} is the attenuation between the transmitter and receiver (dB).

G_{TR} is the receiver antenna gain in the direction of the required signal (dB).

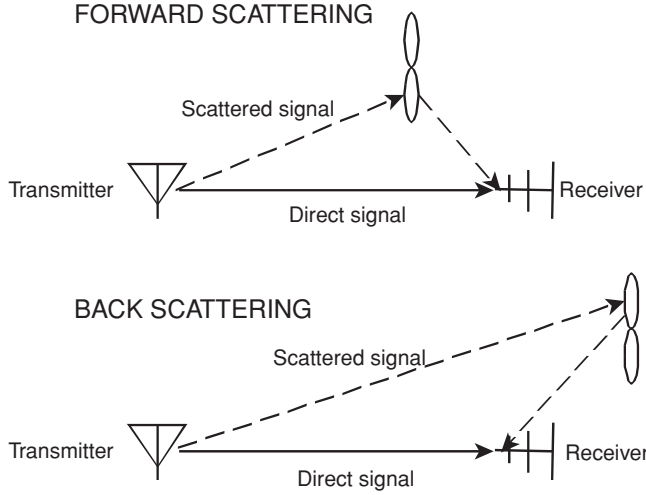


Figure 9.15 Interference mechanisms of wind turbines with radio systems

The interfering signal, I , is given by

$$I = P_T - A_{TW} + 10 \log_{10} \left(\frac{4\pi\sigma}{\lambda^2} \right) - A_{WR} + G_{WR} \quad (9.24)$$

where A_{TW} is the attenuation between the transmitter and wind turbine (dB), A_{WR} is the attenuation between the wind turbine and receiver (dB), G_{WR} is the receiver antenna gain in the direction of the reflected signal (dB), $10 \log_{10}(4\pi\sigma/\lambda^2)$ is the contribution to scattering of the wind turbine, σ is the radar cross section and may be understood as the effective area of the wind turbine and λ is the wavelength of the signal.

It may be seen that the ratio of useful signal to interference is:

$$\frac{C}{I} = A_{TW} - 10 \log_{10} \left(\frac{4\pi\sigma}{\lambda^2} \right) + A_{WR} - A_{TR} + G_{TR} - G_{WR} \quad (9.25)$$

Assume the distance between the transmitter and receiver is much greater than the distance of the wind turbine to the receiver denoted as r , then $A_{TW} = A_{TR}$. Assume the free space loss is $A_{WR} = 20 \log_{10} 4\pi r/\lambda$ and define the antenna discrimination factor as $\Delta G = G_{TR} - G_{WR}$.

Then C/I reduces to

$$\frac{C}{I} = 10 \log_{10} 4\pi + 20 \log_{10} r - 10 \log_{10} \sigma + \Delta G \quad (9.26)$$

Thus, the ratio of the useful carrier signal to interference may be improved by:

- increasing the distance of the turbine to the receiver, r ;
- reducing the radar cross section, σ ;
- improving the discrimination factor of the antenna, ΔG .

The carrier to interference ratio C/I defines the quality of a radio link. For example, a fixed microwave link may have a C/I requirement of 50–70 dB while for a mobile radio service the requirement may be only 15–30 dB (ETSU, 1997c).

Hence Equation 9.25 may be rearranged to define a ‘forbidden zone’ within which a wind turbine may not be located if an adequate carrier to interference ratio is to be maintained.

$$20 \log_{10} r = \frac{C}{I_{\text{required}}} + 10 \log_{10} \sigma - \Delta G - 11 \quad (9.27)$$

It may be seen that the ‘forbidden zone’ is critically dependent on the radar cross section, σ .

Determination of the radar cross section of a wind turbine is not straightforward and a number of approaches are described in the literature. van Kats and van Rees, 1989 undertook a comprehensive series of site measurements on a 45-m diameter wind turbine. They estimated a radar cross section in the back-scatter region of 24 dBm² and a worst case value in the forward-scatter region of 46.5 dBm².

Where measured results are not available, simple predictions may be made based on approximating the turbine blades to elementary shapes (Moglia *et al.*, 1996). For example, for a metallic cylinder the radar cross section is given by

$$\sigma = \frac{2\pi a L^2}{\lambda} \quad (9.28)$$

where:

a is the radius of the cylinder.

L is the length of the cylinder.

λ is the signal wavelength.

while for a rectangular metallic plate

$$\sigma = \frac{4\pi l^2 L^2}{\lambda} \quad (9.29)$$

where l is the width of the plate.

It is suggested that, in the back-scatter region, reflection is only caused by the metallic parts of the blades and so only these dimensions are used in the simple formulae. However, in the forward-scattering region the entire blades make a contribution although this is reduced because of the blade material (GRP) and shape. Hence, Moglia *et al.* (1996) apply the simple formulae but with a 5 dB correction for blade material and a 10 dB correction for blade shape (when using the rectangular approximation).

Hall (1992) quotes a radar cross section model used by a number of researchers as:

$$\sigma = 20 \log_{10} \left[\frac{AX(\alpha)}{\lambda} \right] + 11 - C$$

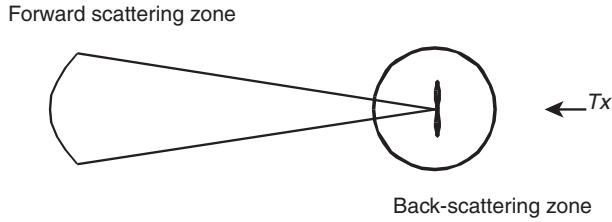


Figure 9.16 Example of simple calculation of interference regions of a wind turbine

where:

A is the area of one blade.

$X(\alpha)$ is a function describing the amplitude of the scattered signal in direction α .

λ is the signal wavelength.

C is a calibration constant, which may be assumed to be 15 dB.

Dabis and Chignell in (ETSU, 1997c) dispute the use of these simple approaches to the determination of the radar cross-section and suggest that more accurate computation is necessary. Their rather complex model is based on a physical optics formulation and assumes a conducting flat plate representation of the blades. It ignores any effects due to non-metallic blade materials and complex blade shapes. Although Dabis and Chignell provide interesting illustrative results of their method they consider significant further work is required to develop the technique further and to validate it against measured field data.

The simple approaches of Moglia *et al.* and van Kats and van Rees allow the calculation of forbidden zones as indicated in Figure 9.16.

The back-scattering region (for a given required C/I value) has a much smaller radius than the forward-scattering region. This is because the radar cross section is much smaller (only the metallic parts near the root of the blades were considered) and also it is possible to take advantage of the directivity of the receiving antenna (ΔG). Equation 9.26 defines only the radii of the two regions and it is necessary to determine the angle over which the two regions extend. van Kats and van Rees remark that some uncertainty exists as to where the back-scat region changes into the forward scatter region. Sengupta and Senior (in Spera, 1994) suggest that 80% of the region around the turbine should be in the back-scat zone with the remaining 20% in the forward scatter region. However Moglia *et al.* use the formula

$$\Phi_{border} = \pi - \frac{\lambda}{L}$$

where

λ is the wavelength.

L the blade element radius.

the back-scat region extends between $0 < |\Phi| < \Phi_{border}$ while the forward-scat region extends between $\Phi_{border} < |\Phi| < \pi$.

For the 45 m diameter turbine studied by van Kats and van Rees, back-scat region radii of 100 m (C/I contour of 27 dB) and 200 m (C/I contour of 33 dB) were determined. The forward scattering-region was much larger with radii of 1.3 km (C/I contour of 27 dB) and

2.7 km (C/I contour of 33 dB). They suggest that a C/I value of 33 dB should result in no visible TV interference. Moglia *et al.* applied their method to medium sized wind turbines (33 m and 34 m diameter) to give a back-scatter radius of approximately 80 m and a forward-scatter radius of 450 m for a 46 dB C/I contour. Their calculations were supported by measured site results, which gave reasonable agreement with the predictions. In these two examples back-scattering is unlikely to be a significant problem as other constraints (e.g. noise and visual effects) will ensure that any dwelling is outside the ‘forbidden zone’.

In summary, it may be concluded that accurate analytical determination of the way wind turbines may interfere with radio communication systems remains difficult. In particular, techniques to determine the effects of irregular terrain and the details of the blade shape and materials have yet to be developed and validated. The approaches that rely on simplified assumptions to estimate the radar cross section can only be considered approximate. However, they do provide a useful qualitative understanding of the problem. In practice a dialogue with the local telecommunications authorities is required in order to determine if a wind turbine or wind farm development is likely to cause electromagnetic interference.

9.4.2 Aviation radar

The potential impact of wind turbines on aviation radar, both civil and military, is of major concern and has resulted in a number of wind farm projects not proceeding (ETSU, 2002). A radar system works by emitting a pulse of electromagnetic energy at radio frequency, receiving and amplifying the reflected signal and then processing it. The distance to the target object is calculated from the time difference between the pulse being emitted and the signal received. A large reflected signal, such as may be caused by the large radar cross-section of a wind turbine, can result in amplitude limiting and distortion within the receiving or signal processing circuits. In addition, aviation primary radars use the Doppler effect to distinguish between fixed and moving objects and can have difficulty in distinguishing between moving wind turbine blades in a wind farm and a moving aircraft.

The main impacts of a wind farm on radar are (ETSU, 2002):

- *Masking.* Air defence radars operate at high frequency and so require a clear ‘line of sight’. Wind turbines can cause shadowing effects and the creation of areas where targets cannot be detected. Masking is also of concern for meteorological radars that look at a very narrow angle just above the ground.
- *Radar Clutter.* Unwanted radar returns are known as ‘clutter’ and rotating wind turbines can lead to a large number of, sometime intermittent, returns that may be interpreted by the radar as a moving object.
- *Scattering.* Scattering occurs when the radar signal is reflected or refracted by rotating wind turbine blades and detected by the emitting radar system. This can lead to multiple false radar returns being displayed or even returns from genuine aircraft being assigned a wrong location.

Figure 9.17 is an example of the calculation of the radar ceiling of three wind farm sites and shows the areas where masking of each of the radars can occur. It uses a digital terrain model with the radar line-of-sight to determine where wind turbines would be likely to mask radar systems. The calculation is based on the radar line-of-sight and so includes the curvature of the earth as well as local topography.

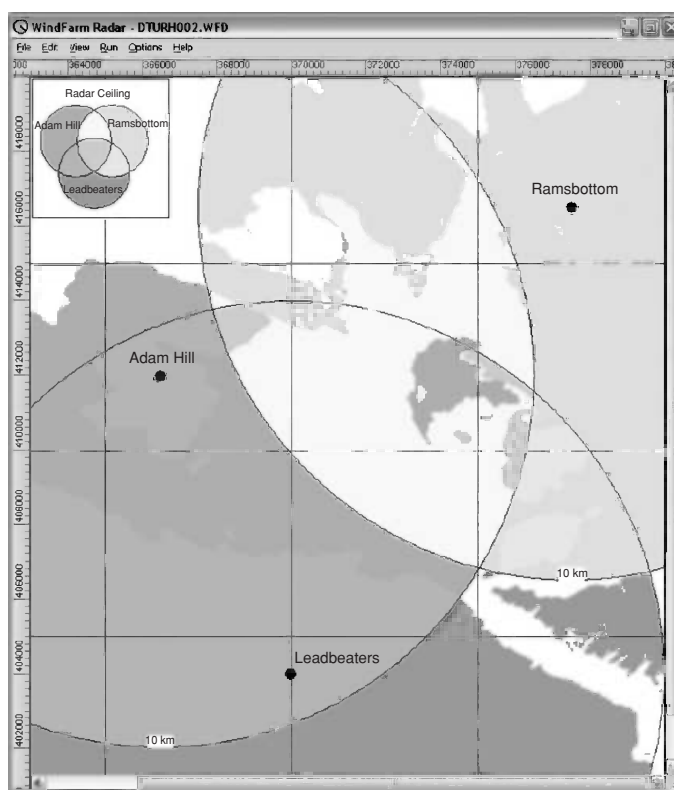


Figure 9.17 Example of radar ceiling calculated for three radar sites. Reproduced by permission of ReSoft

There is continuing interest in reducing the impact of wind farms on aviation radar (ETSU, 2003a, 2003b). Options include modifying the operation of the radar system, changing the geometric layout or location of the wind farm as well as reducing the radar cross section of the wind turbines, particularly the blades. A number of research projects have been undertaken to investigate the use of aircraft stealth technology but to date such approaches have not entered widespread commercial production due to their cost and the increased precision that would be required in the manufacturing process.

9.5 Ecological assessment

Wind farms will often be constructed in areas of ecological importance and the Environmental Statement will include a comprehensive assessment of the local ecology, its conservation importance, the impact of the wind farm (during construction and operation) and mitigation measures. A study of the site hydrology is also likely to be included because of its importance for the ecology. It was suggested by English Nature (1994) that when considering the ecological impact of renewable energy schemes the following categories of effects should be considered:

- Immediate damage to wildlife habitats during construction.
- Direct effects on individual species during operation.

- Longer term changes to wildlife habitats as a consequence of construction or because of changed land use management practices.

Thus, the scope of the ecological assessment is likely to include:

- A full botanical survey including identification and mapping of plant species on the site.
- A desk and field survey of existing bird and non-avian fauna.
- An assessment of how the site hydrological conditions relate to the ecology.
- Evaluation of the conservation importance of the ecology of the site.
- Assessment of the potential impact of the wind farm.
- Proposed mitigation measures, including which part(s) of the site should be avoided.

Gipe (1995) suggests that the main impact on plants and wildlife (excluding birds) is from road building and the disturbance of habitat. Direct loss of habitat due to a wind farm construction is small (approx. 3%) and this can be reduced further by reinstating roads and construction areas once the wind farm is built. However, during construction there is likely to be considerable disturbance including frequent traffic of heavy vehicles and it may be necessary to timetable major work to avoid sensitive periods (e.g. breeding seasons).

9.5.1 Impact on birds

The impact of wind energy developments on birds was particularly controversial in the 1990s because of concern over raptors colliding with wind turbines in the USA and in the south of Spain at Tarifa. Raptors are rare and long-lived birds and so collisions are of particular concern. Elsewhere in Europe the main concern was not collisions of birds with turbines but issues associated with disturbance and habitat loss (Colson, 1995; ETSU, 1996). Since then, there has been considerable research into the impact of wind farms on birds and it would appear that by careful assessment and monitoring, before construction and during operation of the wind farm, the environmental impact on birds may be managed effectively. Guidance on how surveys to assess the impact of onshore wind farms on birds should be conducted is given in Scottish National Heritage (2005).

The presence of a wind farm may affect bird life in one or more of the following ways:

- collision;
- displacement due to disturbance;
- barrier effect; and
- habitat change and loss.

Drewitt and Langston (2006) conducted an extensive literature survey to assess the impact of wind farms on birds. They found very variable annual collision rates reported varying between 0.01–23 birds/turbine. The higher numbers are corrected for scavenger removal and are from a coastal site and relate to gulls, terns and ducks. They also found that none of the barrier effects identified had resulted in significant impact on bird populations while the studies of displacement due to disturbance were largely inconclusive. Typical habitat loss due to a wind farm was approximately 2–5% of the total development area.

Table 9.7 Bird strikes at wind farms in the UK (Lowther S. in ETSU, 1996)

Wind farm	Number of turbines	Bird strikes/turbine year
Burgar Hill, Orkney	3	0.15
Haverigg, Cumbria	5	0
Blyth Harbour, Northumberland	9	1.34
Bryn Titli, Powys	22	0
Cold Northcott, Cornwall	22	0
Mynydd y Cemmaes, Powys	24	0.04

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Lloyd (ETSU, 1996) states that, because the removal of habitat is comparatively small in a wind farm development, the main issues are the effect on bird behaviour either from construction or operation and direct mortality from birds colliding with wind turbines.

A number of important studies reported collisions of raptors with wind turbines in California (Committee on Environmental Impacts of Wind Energy Projects, 2007). The wind farms of the Altamont pass in the early 1990s were not typical of European projects or more recent developments in the USA. The turbines were very numerous, some 7000 turbines in 200 km², rather small in size (typically 100 kW), often on lattice steel towers and with close spacing along the rows facing the prevailing winds. Lloyd (ETSU, 1996) suggests that a number of factors may have led to the high collision rate among raptors. The Altamont turbines were often located on low rises and ridges in the pass to exploit local acceleration of the wind speed. A large number of the collisions appeared to be at the end of the turbine rows where the birds may be attempting to fly around the densely packed turbines. In both Tarifa and Altamont there are few trees and it is suggested that some species used the lattice steel towers as perches and even for nesting.

The bird strike rate for early UK wind farms is quoted by Lowther (ETSU, 1996) and is shown in Table 9.7.

The three experimental turbines on Burgar Hill including a 3 MW 60 m diameter prototype were adjacent to the habitats of a number of bird populations of national significance, including 4% of the UK breeding population of the hen harrier. Studies were undertaken over a nine-year period and during that time four mortalities (three black-headed gulls and one peregrine) were noted as possibly being the result of turbine strike.

The Blyth Harbour wind farm (Still *et al.*, 1994) consists of nine, 300 kW wind turbines positioned along the breakwater of a harbour which has been designated a site of special scientific interest. As it has the highest density of birds of any UK wind farm site (110 varieties identified with more than 1100 bird movements per day) it was the subject of a monitoring programme. The main species that inhabit the area are the cormorant, eider, purple sandpiper and three types of gull. There was particular concern over the purple sandpiper, which winters in the harbour, and so special measures were taken to improve its roosting habitat by providing additional shelter. Over a three-year period, 31 collision victims were identified. Mortality by collision was mainly in eider and among the gulls. It was calculated not to have a significant adverse impact on the local populations. In common with studies undertaken on Dutch and Danish coastal wind farms it appeared that most species had adapted to the wind turbine structures. With respect to disturbance and habitat loss, the study indicated that there was no

significant long-term impact. The purple sandpiper was not adversely affected by the wind farm and, although temporary displacement of cormorant occurred during construction, the population returned to its precious haunts once construction was completed.

The Bryn Titli wind farm is adjacent to a site of special scientific interest that holds important, breeding communities of buzzard, peregrine, red grouse, snipe, curlew and raven. The wind farm was the subject of a bird impact study which showed no statistically significant impact on breeding birds and a bird strike study undertaken in 1994/95 indicated that it is unlikely that there were any collision mortalities during that time.

Clausager (ETSU, 1996) undertook an extensive review of both the American and European literature of the impact of wind turbines on birds. He concluded that, of the mainly coastal locations studied, the risk of death by collision with wind turbine rotors is minor and creates no immediate concern of an impact on the population level of common species. Drawing on 16 studies, and using a multiplier of 2.2 for birds not found, he estimates the highest number of mortalities due to collisions to be six to seven birds/turbine year. With approximately 3500 wind turbines in Denmark this led to the maximum number of birds dying from collision being 20,000–25,000. This number is compared with at least one million birds being killed from traffic in Denmark each year. He dismisses the direct loss of habitat as being small and of minor importance but draws attention to the issues of changes in the area due to the wind farm construction particularly draining of low-lying areas. He also notes that some species of birds temporarily staying in an area may be adversely affected and that an impact has been recorded within a zone of 250–800 m, particularly for geese and waders. The need for further studies, particularly for offshore wind farms and developments on uplands, where only limited number of investigations have been reported, is emphasised.

The Committee on Environmental Impacts of Wind Energy Projects (2007) has undertaken a comprehensive review of the literature on the impacts of wind turbines on birds and bats. Although clearly recognizing the limitations of that data they estimated that in 2003 wind turbines accounted for 0.003% of bird deaths in the USA. They also quote the results of 14 studies on raptors to give an average fatality rate from wind turbines of 0.03 birds per turbine year.

Both Lloyd (ETSU, 1996) and Colson (1995) propose mitigation measures that may be taken to protect important bird species while allowing wind energy development to continue. These include:

- Baseline studies should be undertaken at every wind farm site to determine, which species are present and how the birds use the site. This should be a mandatory part of the Environmental Statement for all wind turbines.
- Known bird migration corridors and areas of high bird concentrations should be avoided unless site specific investigation indicates otherwise. Where there are significant migration routes the turbines should be arranged to leave suitable gaps, (e.g. by leaving large spaces between groups of wind turbines).
- Microhabitats, including nesting and roosting sites, of rare/sensitive species should be avoided by turbines and auxiliary structures. (It may be noted that meteorological masts as well as wind turbines can pose a hazard for birds.)
- Particular care is necessary during construction and it is proposed that access for contractors should be limited to avoid general disturbance over the entire site. If possible,

construction should take place outside the breeding season. If this is not possible then construction should begin before the breeding season to avoid displacing nesting birds.

- Tubular turbine towers are preferred to lattice structures. Consideration should be given to using unguyed meteorological masts.
- Fewer large turbines are preferred to larger numbers of small turbines. Larger turbines with lower rotational speeds are probably more readily visible to birds than smaller machines.
- Within the wind farm the electrical power collection system should be underground.
- Turbines should be laid out so that adequate space is available to allow the birds to fly through them without encountering severe wake interaction. A minimum spacing of 120 m between rotor tips is tentatively suggested as having led to minimum collision mortalities on UK wind farms. Turbines should be set back from ridges and avoid saddles and folds which are used by birds to traverse uplands.

Drewitt and Langston (2006) suggest wind farm development should be avoided where there is

- a high density of waterfowl and waders;
- a high level of raptor activity; or
- a high level of breeding and wintering activity.

It is interesting to note that a 30 MW wind farm development in Scotland was only able to proceed after an extensive study of raptors and the conversion of 450 ha of coniferous forest to heather dominated moorland and the exclusion of sheep from a further 230 ha (Madders and Walker, 1999). It is anticipated that this extension of the moorland habitat will increase the amount of prey away from the wind farm and so reduce the risk of collisions by golden eagles and other raptors.

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Wind energy and the electric power system

10.1 Introduction

As the wind is a diffuse source of energy, wind farms and individual wind turbines are often distributed over wide geographical areas. Hence the public electricity networks, which were originally constructed to supply customer loads from conventional generators (fossil and hydro) are used and extended to gather the electrical energy. Smaller wind farms and individual turbines are connected to the distribution system and are described as distributed generation (Jenkins *et al.*, 2010). Larger wind farms are connected directly to the transmission system and their output displaces energy from conventional central generators.

Traditionally distribution systems have been designed for a uni-directional flow of power from the high voltage transmission network to the customers. Significant generation was not considered in the initial design of the distribution networks and alters the way they operate. Transmission systems were designed for bulk power transfer from fossil or hydro generation to large load centres and the output of wind generation has to be balanced by varying the output of other generators.

In some countries, the output of wind farms now has a significant impact on the technical and commercial operation of central generation. At times of low load and high wind power output, the wholesale market price of electrical energy can drop to a very low value. It may also be necessary to constrain off wind farms in order to allow conventional generation to operate and so provide the ancillary services (e.g. voltage and frequency control) that are necessary for power system operation.

10.1.1 The electric power system

Figure 10.1 is a diagrammatic representation of a typical modern electric power system. The electrical power is generated by large central generating sets and is then fed into an

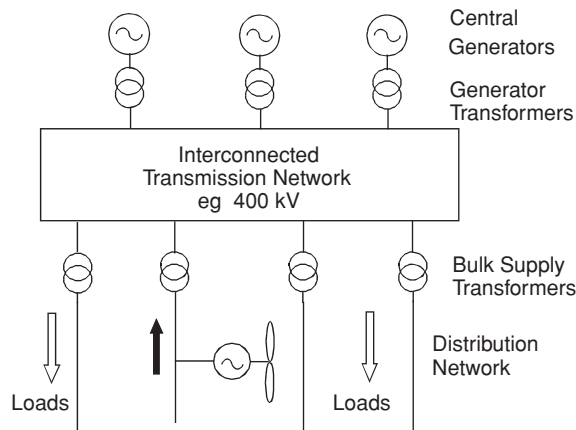


Figure 10.1 Large utility power system

interconnected high voltage transmission system. The generating units may be fossil-fuel, nuclear or hydro sets and will have capacities of up to 1000 MW. The generation voltage is rather low (typically around 20 kV) to reduce the insulation requirements of the machine windings and so each generator has its own transformer to increase the voltage to that of the transmission system. The transmission network is interconnected, or meshed, and so there can be many paths for the electrical power to flow from the generator to the bulk supply transformers.

The bulk supply transformers are used to extract power from the transmission network and to provide it to the distribution networks at lower voltages. Practice varies from country to country but primary distribution voltages can be as high as 150 kV. Distribution networks are normally operated as radial networks with a single path between the bulk supply transformers and the loads. In urban areas with high loads the distribution networks use large underground cables and transformers and so have a high capacity. However, in rural areas the customer loads are often small and so the distribution circuits, using overhead lines, have been designed to have only a limited capability to transport power while maintaining the voltage within the required limits. Small wind farms (distributed generation) are often connected to rural, overhead distribution lines. The design of these networks tends to be limited by consideration of voltage drop or rise rather than thermal constraints of the conductors and this severely limits the ability of these circuits to accept wind generation.

10.1.2 Electrical distribution networks

The conventional function of an electrical distribution network is to transport electrical energy from a transmission system to customers' loads. This is to be done with minimum electrical losses and with the quality of the electrical voltage maintained. The voltage drop is directly proportional to the current while the series loss in an electrical circuit is proportional to the square of the current. Therefore, the currents must be kept low which implies that the network voltage level must be high. However, high voltage plant (e.g. lines, cables and transformers) is expensive due to the cost of insulation and so the selection of appropriate distribution network voltage levels is an economic choice.

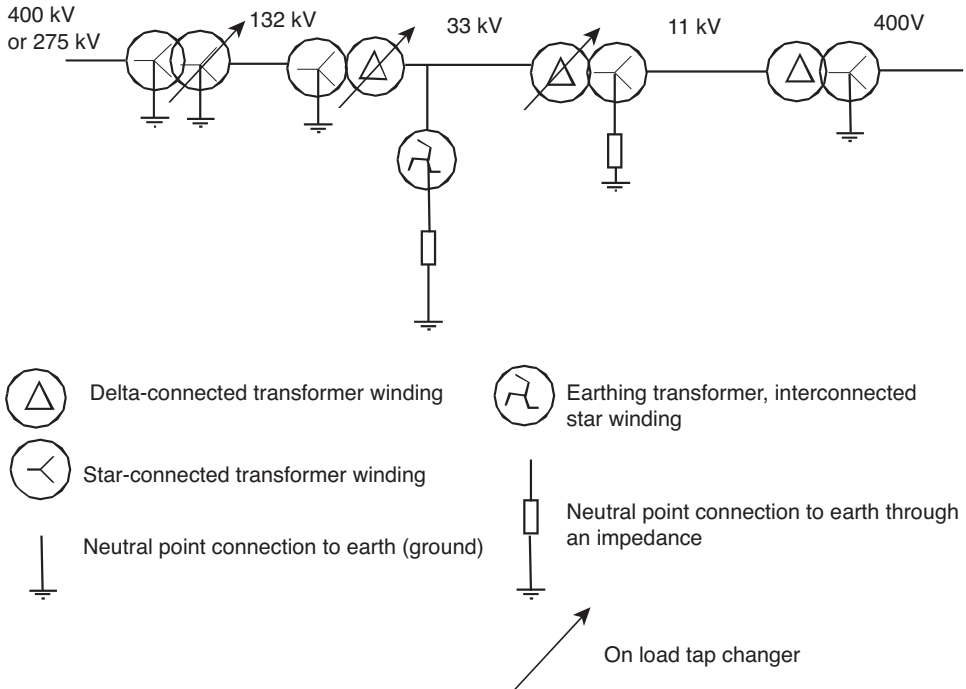


Figure 10.2 Typical UK distribution voltage transformer and earthing arrangements

Figure 10.2 is a schematic representation of a typical UK distribution system although most countries have similar networks. Power is extracted from the interconnected transmission grid and then transformed down to the primary distribution voltage (132 kV in this case). The electrical energy is then transported via a series of underground cable and overhead line circuits to the customers. Most domestic or small commercial customers typically receive electrical energy at 400 V (for a three-phase connection) or 230 V (for a single-phase connection). As the power required to be transmitted in a distribution circuit reduces, the circuit voltage level is lowered by transformers.

The majority of distribution circuits are three-phase although in some rural areas, particularly in the USA, long single-phase circuits are used. Only balanced three-phase networks are suitable for the connection of wind generators larger than about 10 kW. Three-phase transformer windings may be connected either in star or in delta and the winding arrangement chosen varies from region to region and generally follows historical precedent. One advantage of the star connection is that the transformer neutral point is directly accessible and so can easily be grounded or earthed. Where a delta winding connection is used an accessible star point may be created by an earthing transformer. UK practice is to earth the neutral points of each voltage level at one point only, although in certain continental European countries some distribution networks are operated with the neutral point isolated.

Current passing through a circuit leads to a change in voltage and this is compensated for by altering the ratio (or taps) of the transformers. The 11 kV/400 V transformers have fixed taps which can only be changed manually when the transformer is de-energised. However,

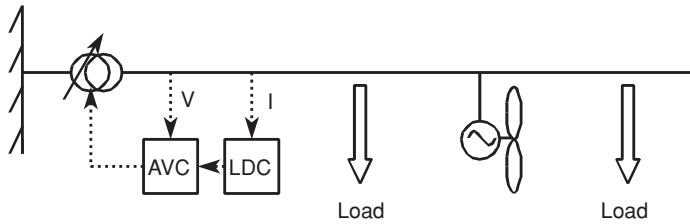


Figure 10.3 Voltage control of a distribution circuit (after Lakervi and Holmes, 1995); AVC – Automatic Voltage Control, LDC – Line Drop Compensation

higher voltage transformers have on load tap changers that can be operated automatically when load current is passing. The simplest control strategy is to use an automatic voltage controller (AVC) to maintain the lower voltage terminals of the transformer close to a set-point (Figure 10.3). The AVC operates by measuring the voltage on the low voltage transformer busbar, comparing it to a set point value and then issuing an instruction to the on load tap changer to alter the ratio of the transformer. Control systems of this type are unaffected by the presence of wind generators on the lower voltage network. Even if the power flow through the transformer is reversed and real or reactive power flows from the lower voltage to the higher voltage network this control system will work satisfactorily. This is because the network source impedance is much less than the equivalent impedance of the wind generators and so the distribution network voltage is controlled predominantly by the transmission network voltage and the tap-changer. Some designs of tap changer have a reduced current rating if power flows from the low voltage side to the high voltage side of the transformer. However, it is rare for distributed wind generation to require the full reverse power flow capacity of a transformer on the distribution network as some of the locally generated power will be taken by local loads.

In some distribution networks, more sophisticated control techniques are used to control the voltage at a remote point along the circuit (Lakervi and Holmes, 1995). This is achieved by adding a signal proportional to the voltage drop on the network to the local voltage input of the AVC relay. This technique is known as line drop compensation (LDC). The LDC element measures the current flowing in the circuit and applies this value to a simple model of the distribution circuit (i.e. the resistance and inductance of the circuit up to the point at which the voltage is to be controlled). The resulting voltage is then subtracted from the local voltage measurement input. If wind generators are connected to the network then the current flow in the circuit will be changed, the effective impedance to the point at which the voltage is to be controlled will alter, and so the LDC technique will no longer work correctly. LDC is sometimes known as positive reactance compounding and another technique, negative reactance compounding, also may not function correctly with wind turbines connected to the distribution circuits (Thomson, 2000).

10.1.3 Electrical generation and transmission systems

Electrical transmission systems are used for the transfer of bulk power from generation to loads. They are interconnected grids operating at voltage levels from 220 kV to 400 kV in Europe and up to 765 kV in large countries (e.g. USA and China). Because of their strategic

importance, transmission grids are normally operated with multiple circuits in parallel so that if an electrical fault occurs and one circuit is isolated, the grid continues to operate satisfactorily. This results in quite low utilization of the circuits during normal operation and so the electrical energy losses within a transmission grid are low, less than 2% of the energy transmitted, while in a distribution network the energy losses are typically 6%. The largest wind farms (greater than, say, 200 MW) are connected to the transmission network directly through wind farm transformers. The voltages within an interconnected transmission grid are determined primarily by the flows of reactive power and so the reactive power output of a large wind farm has to be controlled in response to the requirements of the transmission system operator.

The frequency of an AC power system is determined by the balance of real power being generated and load demanded. If the load demand exceeds the generated power then the frequency will drop while if the power being generated exceeds demand the frequency will rise. The speed with which the frequency changes for a given mismatch of supply and demand power depends on the inertia of all the spinning machines connected to the power system. Some isolated electrical power systems (e.g. Great Britain with a peak demand of around 50 GW) can be considered effectively as a single node for frequency control but large interconnected systems such as that in mainland Europe and in the USA have considerable restrictions on the flows of power from region to region through their transmission circuits. Thus, within each control area, generation and load must be balanced at all times and as wind generation output varies with wind speed, other forms of generation must change their output to ensure the load is met and major interconnecting circuits not overloaded.

10.2 Wind farm power collection systems

Large wind turbines are designed to operate only when they are connected to a stable three-phase AC network. They cannot operate in isolation and the network gives a reference voltage and frequency which is used for control of the turbines and ensures stability of the generator systems. The network also provides power for auxiliaries when the wind speed is low. For fixed speed wind turbines the network provides the reactive power that is necessary for the operation of the induction generators while for variable speed generator systems, the converters are synchronised to the network frequency through their control systems. The wind turbines inject power into the network that varies with wind speed and the stability of the network (both frequency and voltage) is provided mainly by the large conventional (fossil or hydro) generators through the transmission network.

The voltage level chosen for the main power circuits within a wind turbine is usually less than 1000 V and is selected to be one of the internationally standard voltages, in Europe typically 690 V between phases. This voltage is surprisingly low for a large electrical machine and leads to rather high currents. For example a 1.2 MW wind turbine generating at 690 Volts has a current of over 1000 Amps. However, it has been found to be cost-effective to restrict the voltage of the wind turbine generator to less than 1000 V as, in many countries, the safety requirements become very much more severe for voltages above this value and special precautions are required including the provision of dedicated equipment to earth the circuits before work on them is permitted. Further reasons for this low voltage include a wider choice of switchgear and flexible cables for the pendant as well as higher production volumes and hence lower costs of the generators.



Figure 10.4 Transformer and switchgear of 1.5 MW wind turbine, MV switchgear is on the right LV switchgear on the left. Reproduced by permission of NEG MICON

However, this low generator voltage leads to a requirement for a transformer located either in the nacelle, in the tower base or immediately adjacent to it (Figure 10.4). In some early wind farms a number of small wind turbines were connected to one transformer (Figure 10.5a) but as turbine ratings have increased, with consequently higher currents, this approach leads to excessive voltage variations and electrical losses in the low voltage cables. It is only with rather small wind turbines located close together that the connection of multiple turbines to one transformer is likely to be cost-effective. Hence, larger wind turbines all have one transformer per turbine (Figure 10.5b).

The Point of Connection is defined as the point where the wind farm power collection system joins the utility distribution network. The Point of Common Coupling is the nearest point to the wind farm on the utility network to which other customers are (or can be) connected. In many circumstances these points may coincide but the Point of Connection defines the ownership boundary while the impact of the wind turbines on power quality is evaluated at the Point of Common Coupling.

The layout of the wind turbines within the wind farm is fixed by the wind energy resource, local topography and the requirements of the planning authorities. The power collection network is then designed to transport the power to the local public electricity system.

The choice of medium voltage (MV) level for the wind farm power collection system usually follows the practice of the local distribution utility. In this way cable and switchgear is readily available. Thus, in the UK the choice is generally between 11 kV and 33 kV while in continental Europe wind farm collection circuits are likely to be between 10–30 kV. Larger wind farms may include a transformer (e.g. 33/132 kV) to increase the voltage for onward transport of the power.

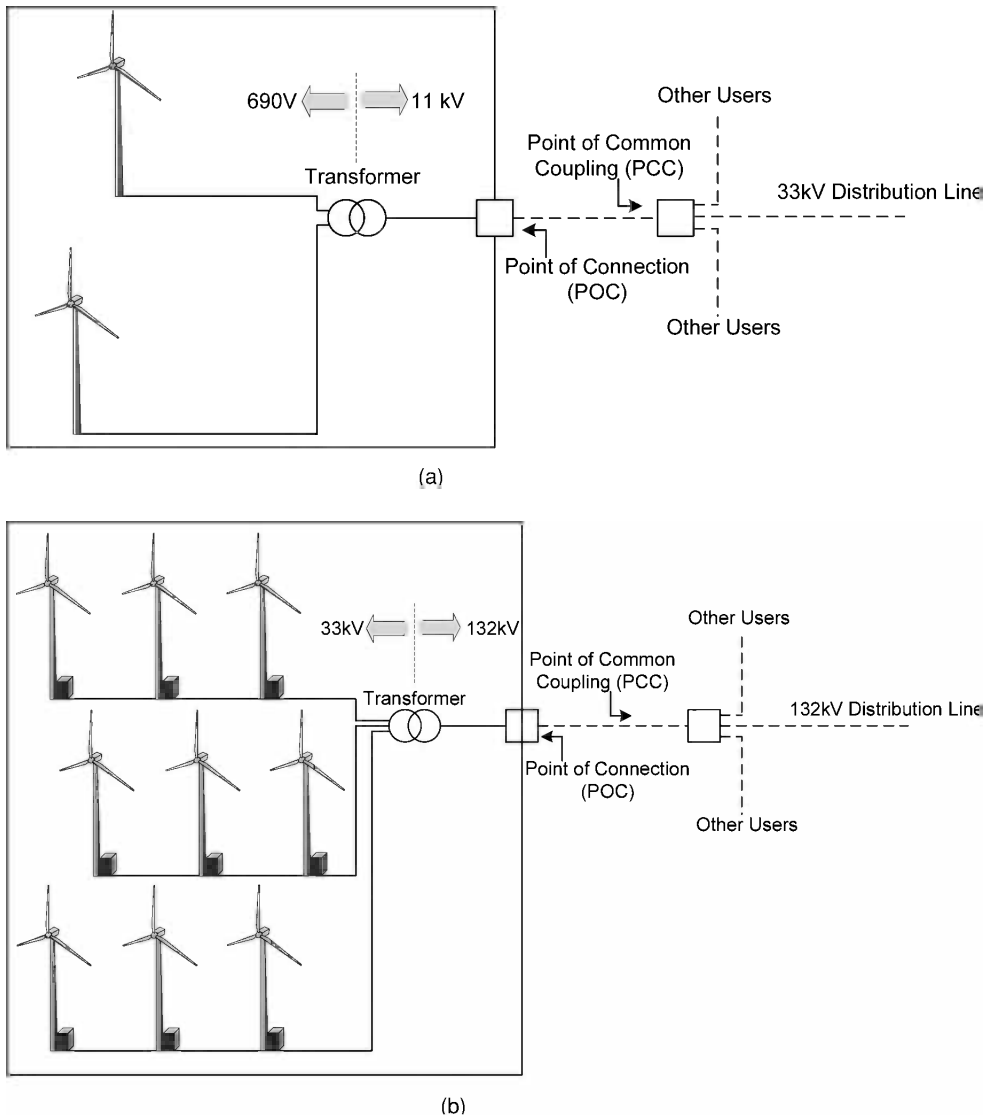


Figure 10.5 (a) Electrical power collection system of small wind turbines. (b) Electrical power collection system of a large wind farm

Local utility practice also influences the choice of the neutral earthing arrangement on the wind farm MV circuits (Figure 10.2). In the UK, 11 kV circuits tend to have their neutral point either solidly earthed, or connected to ground through an impedance typically a low value resistor (often 6.35 ohms to allow 1000 Amps to flow into a phase/ground fault). Solid neutral earthing is cheaper as it requires no extra equipment, but it can lead to high earth fault currents which may cause damage and high step or touch voltages (ANSI/IEEE Std 80–2000). In contrast, on some continental European wind farms the neutral point of the

medium voltage system is left unearthed. This allows operation of the wind farm to continue with a single-phase earth fault but with the voltages of the healthy phases raised by a factor of $\sqrt{3}$.

The design of the power collection is very similar to that of any public utility MV power network but with one or two particular aspects. It is generally not appropriate to provide any redundancy in the circuits to take account of failures of the MV electrical equipment. Both operating experience and reliability calculations have shown that the power collection system of a wind farm is so much more reliable than the individual wind turbines that it is generally not cost-effective to provide any duplicate circuits. This also applies to the single circuit or transformer linking the wind farm to the utility network where duplication cannot usually be justified. If part of the wind power collection circuit does fail then the only consequence is the lost revenue of the energy output of that part of the wind farm. This is easily estimated and is generally modest. In contrast, if a public distribution utility circuit fails then the consequence is that the customers do not have electrical power. The social and commercial value of this loss of supply is difficult to quantify but is generally considered to be several orders of magnitude higher than the price of the electrical energy. Thus, MV public distribution circuits usually have duplicate supplies while wind farm power collection networks tend to consist of simple radial circuits with limited switchgear for isolation and switching. The wind turbine transformers are connected to the radial circuits through simple switchgear.

In Europe almost all power collection circuits within wind farms use underground cable. This is for reasons of visual amenity and for safety as large cranes are required for the erection of the turbines. In some parts of the world, overhead MV lines have sometimes been used within the wind farm to reduce costs. If underground cables are used, then their low series inductive reactance leads to small variations of voltage with current and so the circuits are dimensioned primarily on considerations of current rating and electrical losses. Any wind farm project which reaches the stage of requiring a detailed design of the electrical system should also have a good estimate of the wind resource and hence energy output. Thus, it is straightforward to use this data to calculate the energy losses in the electrical equipment at various wind turbine output powers using a load flow program, sum these over the life of the project and using discounted cash flow techniques choose the optimum cable size and transformer rating. In countries where wind generated electricity attracts a high price then it is likely to be cost-effective to install cables and transformers with a thermal rating in excess of that required at full wind farm output in order to reduce losses. It may also be worthwhile to consider low-loss transformer designs.

10.3 Earthing (grounding) of wind farms

The power system neutral points as well as the metallic enclosures and external conducting parts of all electrical equipment require a connection to the general mass of earth in order to:

- Establish a low impedance path for earth fault currents and, hence, satisfactory operation of electrical (over-current) protection.
- Minimise shock hazards to personnel and animals, including control of step and touch potentials.
- Ensure satisfactory discharge of lightning currents and control the associated voltages.

- Prevent large potential differences occurring which are potentially hazardous to both personnel and equipment.

In the UK this subject is referred to as ‘earthing’ while in the US it is called ‘grounding’. The terms are synonymous.

The neutral points of the three-phase power circuit are usually connected to earth at each voltage level to provide a path for earth fault current. Also the metallic enclosures of electrical equipment are earthed to ensure they do not rise in potential if an internal insulation failure occurs. High structures such as wind turbines are earthed to ensure there is a low impedance path for lightning current. These three rather different objectives are usually achieved with a common earthing and bonding system of the wind farm.

Because earthing has obvious safety implications there is considerable guidance available in both national and international standards for conventional electrical plant. For earthing of AC substations, the US standard (ANSI/IEEE 80, 2000) is widely applied and Charlton has written an informative guide (Copper Development Association, 1997) which is mainly focused on UK practice.

Wind farms, however, have rather unusual requirements for earthing. They are often very extensive, stretching over several kilometres, are subject to lightning strikes because of the height of modern wind turbines and are often on high resistivity ground being located on rocky terrain and on the tops of hills. Thus, normal earthing practice tends not to be easily applicable and special consideration is required. IEEE 1094 (1991), which has now been withdrawn, recommended ‘that the entire wind-farm installation have a continuous metallic ground system connecting all equipment. This should include, but not be limited to the substation, transformers, towers, wind turbine generators and electronic equipment’. This practice of a single, wind farm wide earthing system for both 50/60 Hz fault currents and lightning protection is recommended by IEC 61400–24 (2010) and is generally implemented with bare conductor being laid in the cable trenches above the power collection cables. These conductors provide both bonding of all parts of the wind farm and a long horizontal electrode to reduce the impedance to ground of the earthing system.

A wind farm earthing system is required to operate effectively for both power frequency 50 Hz (60 Hz in the USA) currents and lightning surges which have rise-times typically of less than ten microseconds. Although it is conventional to use the same physical earthing network for power frequency currents and lightning surges the response of the earthing system to the high frequency components of the lightning current is quite different to that at 50/60 Hz.

The performance of a wind farm earthing system may be understood qualitatively by considering Figure 10.6.

At each wind turbine a local earth is provided typically by placing a ring of bare conductor around the foundation at a depth of about 1 m (sometimes known as a counterpoise earth) and by driving vertical rods into the ground. It is common to bond the steel reinforcing of the wind turbine foundation as well as the tower into this local earthing network. The purpose of this local earth is to provide equi-potential bonding against the effects of both lightning and power frequency fault currents, and to provide one element of the overall wind farm earthing system. These local turbine earths are shown as R_{turbine} in Figure 10.6. As the turbine earthing network consists of a ring of only, say, 20 m in diameter and vertically driven rods it may be considered as resistive to ground.

However, the long horizontal electrodes linking one turbine to the next have a more complex behaviour (similar to a transmission line) and are represented in Figure 10.6 as a

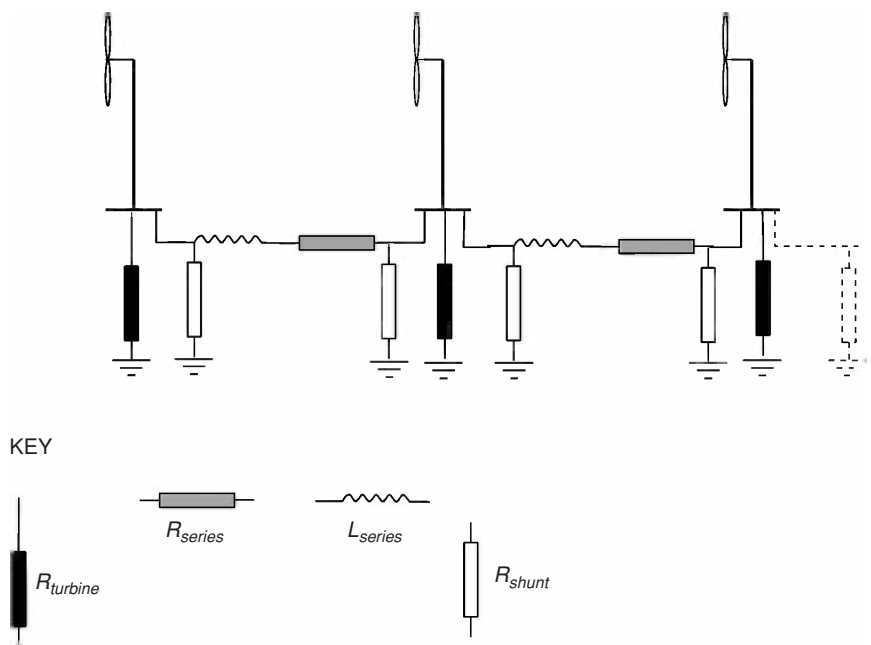


Figure 10.6 Schematic of a wind farm earthing system

π equivalent circuit. Thus, the resistance to ground is indicated by R_{shunt} while the series impedance is the combination of R_{series} and L_{series} . R_{series} comes about simply from the longitudinal resistance of the earth wire while L_{series} is the self-inductance of the earth circuit. On the long earthing networks found on large wind farms this series impedance cannot be ignored. It may be seen immediately that for the high frequency of a lightning strike on a wind turbine the series inductance acts effectively to reduce the earthing network to only the local turbine earth. Even with 50 Hz fault current the series impedance leads to significantly higher earthing impedances than would be expected with geographically small earthing systems where series impedance may be neglected. In this discussion shunt capacitance and soil ionisation have been ignored, although these may become significant at high frequencies and currents.

This behaviour of wind farm earthing systems has been confirmed by site measurements and Table 10.1 shows the measured values for the earth impedance at the main substations of two UK wind farms.

Table 10.1 Measured earth impedance at substations of two wind farms

Capacity of wind farm (MW)	No. of turbines	Route length of horizontal earth conductor (km)	Earth impedance at main substation (ohms at 50 Hz)
7.2	24	6.7	$0.89 + j0.92$
33.6	56	17.7	$0.46 + j0.51$

It may be seen that even at 50 Hz the impedance to earth consists of almost equal resistive and inductive parts (i.e. the X/R ratio of the impedance of the earthing system to ground is almost unity). This has important implications for the design and testing of wind farm earthing systems. Simple conventional calculation techniques which were developed for small, fully resistive, earth networks may give optimistic results and the impedance of the long horizontal conductors at lightning frequencies needs to be considered. Sophisticated computer programs to do this are available and these may be used to calculate both 50 Hz and lightning surge behaviour. Similarly, simple testing methods such as those developed by Tagg (1964) are only intended to measure resistance to earth and will give optimistic results on large wind farms. An effective measurement method to determine the 50 Hz earth impedance of a large wind farm is by a current injection test. In this test a current (typically 10–20 Amps) is injected into the earth electrodes at the wind farm and the potential rise measured with respect to a remote ‘true’ earth. However, the return path of the injected current must be remote from the wind farm (usually some 5–10 km away) to ensure that it does not affect measurements at the wind farm. Hence, it is conventional to use the de-energised circuit connecting the wind farm to the main power network as the path of the test current. The rise in potential of the wind farm earthing systems is measured against a remote earth which classically is transferred to the site over a metallic telephone circuit. Ideally the route of the telephone circuit should be orthogonal to the power circuit to avoid induced effects. It may be seen that tests of this type are difficult and expensive to undertake as they involve taking the wind farm out of commission and arranging a temporary earth at the remote end of the current injection circuit. Also it is becoming increasingly difficult to use a telephone circuit to provide the remote earth reference as telecommunication companies increasingly use non-metallic media. However, it is very questionable whether simpler tests give useful results on large earthing networks with significant series inductance.

10.4 Lightning protection

It is now recognised that lightning is a significant potential hazard to wind turbines and that appropriate protection measures need to be taken. Authoritative guidance on lightning protection of wind turbines is given in IEC 61400–24 (2010).

Some years ago it was thought that if the blades of wind turbines were made from non-conducting material (i.e. glass reinforced plastic (GRP) or wood-epoxy) then it was not necessary to provide explicit protection for these types of blades provided they did not include metallic elements for the operation of devices such as tip brakes. However, there is now a large body of site experience to show that lightning will attach to blades made from non-conducting materials and can cause catastrophic damage if suitable protection systems have not been fitted. If lightning current enters a blade and is not controlled by a metallic lightning conductor the arc may cause an increase in pressure that will damage the blade catastrophically. If carbon fibre, which is conducting but with a rather high resistance, is used to reinforce the structure of the blades then additional precautions are necessary (Cotton *et al.*, 2000; IEC 614200–24, 2010).

Lightning is a complex natural phenomenon often consisting of a series of discharges of current. The term ‘lightning flash’ is used to describe the sequence of discharges which use the same ionised path and may last up to a second. The individual components of a flash are called ‘strokes’.

Table 10.2 Effects of various aspects of lighting on a wind turbine

Parameter	Effect on wind turbine
Peak current (A)	Heating of conductors, shock effects, electromagnetic forces
Specific energy (Joules/ohm)	Heating of conductors, shock effects
Rate of rise of current (A/s)	Induced voltages on wiring, flashovers, shock effects
Long duration charge transfer (Coulombs)	Damage at arc attachments point or other arc sites (e.g. bearing damage)

Lighting flashes are usually divided into four main categories:

- Downward inception, negative and positive polarity.
- Upward inception, negative and positive and polarity.

Generally flashes which start with a stepped leader from the thunder cloud and transfer negative charge to earth (downward inception – negative polarity) are the most common. Downward negative flashes typically consist of a high amplitude burst of current lasting for a few microseconds followed by continuing current of several hundred Amps. Then, following the extinction of the initial current transfer between cloud and earth there may be a number of re-strikes. However, in some coastal areas there may be a majority of positive flashes of downward inception which consist of a single long duration strike with high charge transfer and specific energy content. Upward inception is normally associated with very high objects (e.g. communication towers and large wind turbines) and the lightning has very different characteristics. Although the maximum peak current is low (some 15 kA) the charge transfer can be very high and, hence, has significant potential for damage. The top of wind turbine blades can now be over 100 m above ground and so there is growing concern over the effect of upward negative lightning flashes. Upward positive flashes are rare.

Table 10.2 shows the parameters normally used to characterise lighting and some aspects of their potential for damage in wind turbines. The peak current of a single lightning stroke may be over 200 kA but the median value is only approximately 30 kA. Corresponding values for charge transfer are 400 C (peak), 5 C (median) and specific energy 20 MJ/ohm (peak) and 55 kJ/ohm (median). The very large range of these parameters implies that the initial step in any consideration of lightning protection of a wind farm or wind turbine is to undertake a risk assessment (IEC 614200–24, 2010). The risk assessment will include consideration of the location of the turbines as the frequency and intensity of lighting varies considerably with geography and topology. On flat terrain a wind turbine is assumed to attract the same frequency of lightning as a circular area with a radius of three times the height of the wind turbine.

Table 10.3 shows some historical data of the frequency of lightning damage (Cotton *et al.*, 2000). Although this must be interpreted with care, as both wind turbine design and lightning protection systems were evolving rapidly during this period, the data does indicate the scale of the problem. A large number of the faults shown in Table 10.3 were due to indirect strokes affecting the wind turbine and wind farm control systems. Generally the number of faults was dominated by incidents affecting the control and electrical systems while blade damage

Table 10.3 Wind turbine lightning damage frequency

Country	Period	Turbine years	Lightning faults	Faults per 100 turbine years
Germany	'91 to '98	9204	738	8.0
Denmark	'91 to '98	22000	851	3.9
Sweden	'92 to '98	1487	86	5.8

(Cotton *et al.*, 2000)

gave the highest repair costs and loss of turbine availability, and hence reduction in wind farm revenue.

Figure 10.7 shows some of the techniques that have been used to protect blades against lightning. The main distinction is whether a limited number of receptors are used to intercept the lightning (Types A and B) or if an attempt is made to protect the entire blade (Types C and D). Type A shows how, using one or two receptors at the tip, the steel control wire of the tip brake may be used as the down conductor. In a blade without a moveable tip (Type B) then an additional conductor is installed. Type C has the down conductor located on the leading and trailing edges although there are considerable practical difficulties in firmly attaching a suitable conducting path to the leading edge. Type D shows the use of a conducting mesh on

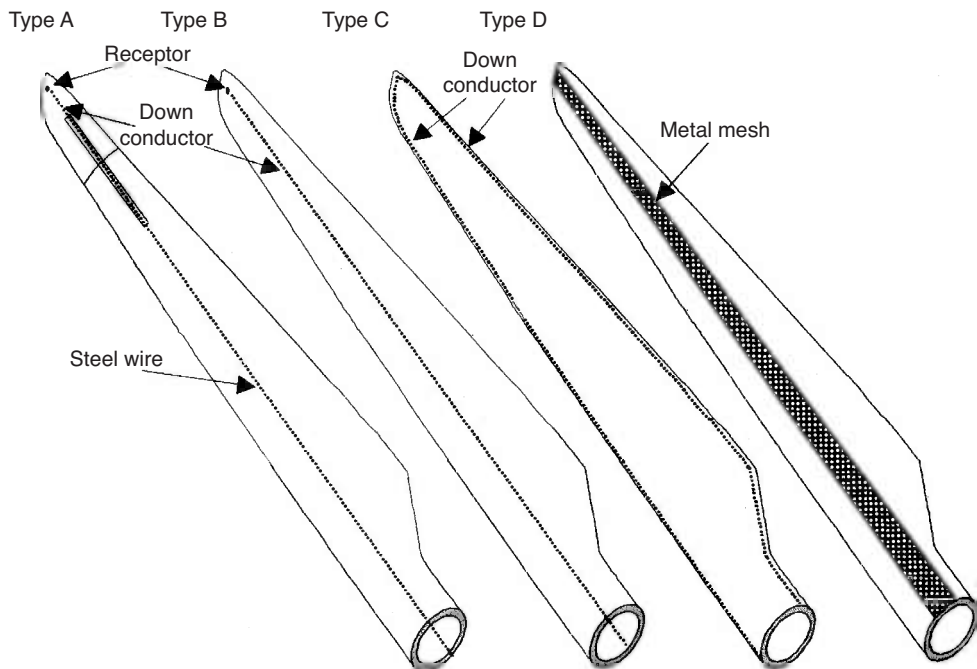


Figure 10.7 Methods of lightning protection for blades. Cotton *et al.*, 2000, by permission of Dr R Sorenson, DEFU

each side of the blade. The main mechanism of damage is when the lightning current forms an arc in air inside the blade. The pressure shock wave caused by the arc may explode the blade or, less dramatically, cause cracks in the blade structure. Thus, for effective protection it is essential that the lightning attaches directly to the protection system and then is conducted safely down the length of the blade in a metallic conductor of adequate cross-section.

Once the lightning current has been conducted to the root of the blade there remains the problem of ensuring its safe passage to the outside of the tower and, hence, to earth. This is not straightforward as it is necessary for the current to pass across the pitch, shaft and yaw bearings while not damaging the generator and sensitive control equipment in the nacelle. The present understanding of how wind turbine bearings may be damaged by the passage of lightning current is summarised in IEC 614200–24. Generally large heavily loaded bearings are unlikely to be catastrophically damaged although there may be a reduction in their service life. There is, so far, no effective means of shunting lightning current around a large bearing as the bearing itself is the lowest inductance path and so the preferred route for the high frequency current.

The control and electrical systems are protected against lighting by dividing the turbine into zones depending on whether direct attachment of lightning is possible and the magnitude of the current and, hence, electro-magnetic field expected in each zone, the Lightning Protection Zone approach (IEC 614200–24). Within each zone, components are protected to withstand the anticipated effects of lighting. The principal damage mechanisms are direct conduction and magnetic coupling while the main protective measures are good bonding, effective shielding and the use of appropriate surge protection devices at the zone boundaries.

10.5 Connection of wind generation to distribution networks

Distribution utilities have an obligation to operate the electrical distribution networks in such a way as to provide power to their customers at an agreed power quality. At present, power quality requirements are based on national standards although a common European position is emerging based on the document (BS) EN 50160 (2007). This document gives the characteristics of the voltage that can be expected across the public distribution network but does not apply to emission limits from network users (i.e. wind turbines and wind farms).

The main impacts of wind turbines on distribution systems are:

- Slow (or steady state) voltage variations.
- Rapid voltage changes (leading to flicker).
- Waveform distortion (i.e. harmonics).

The operation of wind turbines is effected by:

- Voltage Unbalance (negative phase sequence voltage).
- Transient voltage variations (dips and sags).

Consideration of whether a wind generation scheme may be connected to a distribution circuit is based on its impact on other users of the network. Similar considerations apply to the connection of any load. The steady state voltage variations are generally considered assuming conditions of minimum (or zero) network load with maximum wind generation and maximum network load with minimum generation. These are rather onerous and conservative

Table 10.4 Indication of possible connection of wind farms to distribution networks

Location of connection	Maximum capacity of wind farm (MW)
Out on 11 kV network	1–2
At 11 kV busbars	8–10
Out on 33 kV network	12–15
At 33 kV busbar	25–30
Out on 132 kV network	30–60

assumptions to ensure that steady state voltage limits will never be violated. However, they do ensure that the wind farm output can always be exported to the network (other than when network faults occur). In some cases, local agreements have been reached between the network and generator operators so that some wind turbines are constrained off at times of low network load or abnormal network conditions. This then allows a larger wind farm to be connected than would otherwise be the case. However, the risk of not being able to generate is then transferred to the wind farm operator.

The voltage change permitted in 11 kV circuits is small (typically $\pm 1\%$ or 2%) as any voltage variation is passed directly through the fixed tap (11/0.4kV) transformers to the customers. In contrast, 33 kV and 132 kV circuits are allowed to operate over a wider voltage range (e.g. up to $\pm 6\%$) as the automatic on-load tap changers supplying the lower voltage networks and, hence, customers can compensate for variations in the higher voltage networks. Therefore, it is common to find that only very limited capacities of wind generation can be connected at 11 kV as the high impedance of the circuits and relatively large currents flowing from the generators result in unacceptable variations in voltage. Determining the size of a wind farm which may be connected to a particular point in the distribution network requires a series of calculations based on the specific project data. However, Table 10.4 gives some indication of the maximum capacities which experience has indicated may be connected. Table 10.4 assumes that the wind farms are made up of a number of turbines and so the connection assessment is driven by voltage rise effects and not by power quality issues due to individual large machines.

For wind farms of relatively large capacity connected to weak distribution networks the calculations required can be involved and use computer based power system studies. The expense of these calculations may not be justified for small wind farms and so some countries have adopted an assessment approach to allowing connection based on the ratio of wind farm capacity (in MW) to symmetrical short-circuit level, without the wind farm connected (in MVA).¹ Typical values chosen range from 2–5% based on studies and on experience

¹ The electrical ‘strength’ of a circuit is described by its short circuit or fault level. The short circuit level is the product of the pre-fault voltage and the current which would flow if a three-phase symmetrical fault were to occur. Clearly, this combination of current and voltage cannot occur simultaneously but the fault level (expressed in MVA) is a useful parameter which gives an immediate understanding of the capacity of the circuit to deliver fault current and resist voltage variations. In the per unit system the fault level is the inverse of the magnitude of the source impedance (Weedy and Cory, 1998).

Table 10.5 Ratio of site capacity to connection short-circuit level for two large (Fixed Speed Induction Generator) wind farms

Site capacity MW	Number of turbines	Short-circuit level of connection MVA	Ratio of wind farm capacity to short-circuit level (%)
21.6	36	121	18
30.9	103	145	21

(Gardner, 1996). However, such simple rules can be too restrictive and lead to refusal of permission to connect or to excessive reinforcement of the distribution system. Table 10.5 gives data on two large wind farms in successful commercial operation in the UK with much higher ratios of wind farm capacity to short-circuit level. Both wind farms have operated successfully for some years but it should be noted that the number of turbines on each site is large and so the impact of any individual machine is small. Both sites are connected to 33 kV circuits from which other customers are supplied only through transformers fitted with automatic tap-changers.

Figure 10.8 shows the calculated variation of voltage with wind farm output at one site. The generator voltage of the turbines was 690 V and each turbine had a local 690/33 kV transformer. The 33 kV curve refers to the voltage at the point of connection of the wind farm to the public distribution network. With zero wind farm output the system voltage is close to nominal as the distribution network is lightly loaded. As the wind farm output increases the voltage rises but, once rated power is reached, then the reactive power drawn by the generators increases rapidly and so the voltage drops. At 140% output power the load flow calculation

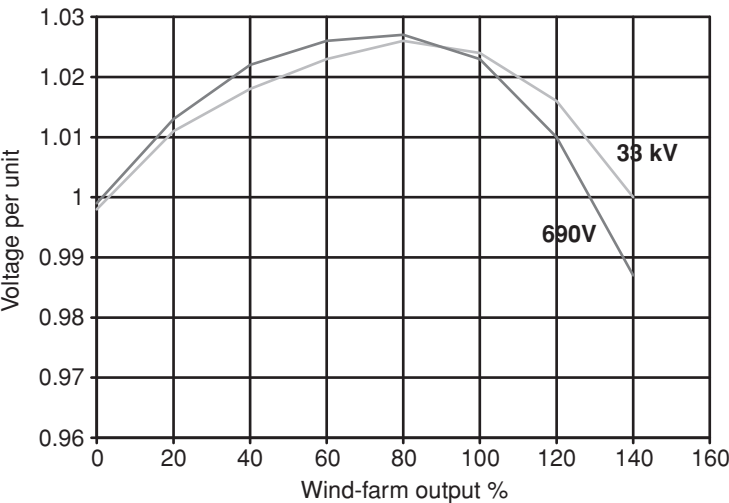


Figure 10.8 Variation of wind farm voltage with output power (generator voltage 690 V, connection voltage 33 kV)

fails to converge² indicating that the network voltage is likely to collapse. Although such overloads are unlikely on wind farms with many turbines, individual (pitch regulated, fixed speed) machines can produce up to 200% of their rated output and so this phenomenon of voltage instability can be significant when considering the connection of large fixed speed induction generator wind turbines to weak networks.

10.6 Power system studies

Power system studies are required in order to assess the impact of large wind farms on the network and to ensure that the network conditions are such as to allow the wind generators to operate effectively. In the past simple manual calculations were used but sophisticated computer programs are now available and are always used on large projects.

Load flow (or power flow) programs take as inputs the network topology and impedances of the lines, cables and transformers. The customers' load and generator outputs are then used to calculate the steady state performance of the network in terms of voltages, real and reactive power flows and network losses. Generally a balanced three-phase system is assumed. For wind energy applications it is important that the load flow program includes a good model of the tap-changers on the network transformers and an effective representation of the wind turbine generators. A simple load flow gives the state of the network at a single instant but some commercially available programs allow the use of daily load and generator profiles to repeat the calculations many times (sequential steady state analysis) to show the network performance over a period of time.

Fault calculators calculate the fault currents in the network at one instant. Both balanced and unbalanced faults may be investigated. They use the method of symmetrical components (Wagner and Evans, 1933) to calculate the effect of unbalanced short-circuits on the power system. Fault calculators that are used to examine the connection of wind turbines require good models of the induction generators or power electronic converters. Fault calculators give as output the fault current flows and network voltages at particular times after the fault occurs. Some of the more sophisticated programs use analytical methods to estimate the decay of fault currents with time but the main analysis uses a steady-state technique.

Transient stability programs allow investigation of how wind generators will respond to disturbances on the network. A balanced network is assumed as the calculation is based on an equivalent single-phase representation using fundamental frequency phasors with an integration algorithm to calculate how the network conditions change over time. Traditionally, transient stability programs were used to investigate the angle stability of synchronous generators and a good model of the wind turbine generators is required for useful results.

Electro-magnetic programs are among the more sophisticated tools and unlike the previous programs do not assume perfect, fundamental frequency waveforms. Detailed representation of both the induction generators and power electronic converters is possible. These programs use detailed time-domain simulations which allow reconstruction of the distorted wave forms generated by power electronic converters as well as investigation of high frequency transients such as those due to lightning. They may be used to investigate almost any condition on

²The failure of a load flow program to converge does not predict voltage collapse accurately but is an indication that the system may be close to instability.

the power system but at the cost of considerable complexity and computation time. Electro-magnetic simulations are not routinely used for wind farm design but only to investigate particular problems.

There are of course many more specialised calculation techniques and associated tools including probabilistic power flow, optimal power flow, harmonic load flow and various earthing/grounding codes. However, their use requires specialist knowledge and so they are not in common use for small wind farm projects.

10.7 Power quality

Power quality describes how closely the electrical power delivered to customers corresponds to the appropriate standards and so operates their end-use equipment correctly (Dugan *et al.*, 1996). Thus, it is essentially a customer focused measure that is greatly affected by the operation of the distribution and transmission network. Wind turbines, and other forms of distributed generation, can reduce power quality by distorting the voltage supplied to customers but also increase power quality by raising the network short-circuit level and so stabilising the voltage.

There are a large number of ways in which the electrical supply (i.e. voltage or frequency) can deviate from the specified values. These range from transients and short-duration variations (e.g. voltage sags or swells) to long-term waveform distortions (e.g. harmonics or unbalance). Sustained complete interruptions of supply are generally considered an issue of network reliability rather than power quality. The importance of power quality is due to the increasing use of sensitive load equipment including computer based controllers and power electronic converters as well as the awareness of customers of the commercial consequences of equipment mal-operating due to disturbances originating on the power system.

The issues of power quality are of particular importance to distributed wind turbines connected to rural distribution networks. Individual units can be large, up to 2 MW, feeding into distribution circuits with a high source impedance and with customers connected in close electrical proximity. For variable speed wind turbines, which use power electronic converters, harmonic distortion of the network voltage may be important while the connection of fixed speed induction generator turbines to the network needs to be managed carefully if excessive voltage transients are to be avoided.

During normal operation wind turbines produce continuously varying output power. The higher frequency power variations are mainly caused by the effects of turbulence, the wind shear, tower shadow and the operation of the turbine control system. These effects lead to periodic power pulsations at the frequency which the blades pass the tower (typically around 1 Hz for a large turbine), which are superimposed on the slower variations in power caused by changes in wind speed. There may also be power variations (at a few Hz) caused by the dynamics of the turbine. Variable speed operation of the rotor has the advantage that many of the faster power variations are not transmitted to the network but are smoothed by the flywheel action of the rotor. However, fixed speed operation, using a low slip induction generator, will lead to cyclic variations in output power, at rotor rotational frequency (1P) as well as blade passing frequency (3P), and hence network voltage changes at these frequencies. The connection of large induction generators on to weak networks may also lead to transient voltage changes.

It is essential that wind turbines do not degrade the power quality of the distribution network as otherwise permission for their connection or continued operation will be refused by the distribution utility. In addition to national standards describing the limits of voltage variations that are caused by the connection of any equipment (e.g. Electricity Networks Association, 1989), the particular importance of the influence of wind turbines on power quality has been recognised in the standard ‘Measurements and assessment of power quality characteristics of grid connected wind turbines’ (IEC 614200–21, 2008). This standard lists the voltage quality indicators that should be provided by the turbine manufacturer to describe the voltage quality of wind turbine power output. It also details the testing procedures required to determine the wind turbine’s response to a network voltage drop, as well as the operation of its power control and grid protection systems.

The main aspects considered in IEC 614200–21 are:

Voltage Fluctuations

- flicker coefficient for continuous operation as a function of network source impedance phase angle and annual average wind speed;
- maximum number of wind turbine starts within 10 minute and 120 minute periods;
- flicker step factor and voltage change factor at start up as a function of network source impedance phase angle.

Current harmonics and inter-harmonics

- harmonic currents, up to the fiftieth, during continuous operation but excluding harmonics during wind turbine start-up.

Response to network voltage drops

Active power control

- maximum output power (ten minute average, 60 second average and 200 minutes average values);
- ramp-rate limitation after wind turbine start-up;
- active power set-point control for network frequency support.

Reactive power capability and set-point control

Grid protection

- settings of under/over voltage and frequency relays and reconnection time.

In general the main power quality issue when connecting wind farms with a large number of generators (more than ten wind turbines) is steady state power flow and voltage rise while for individual large turbines on weak networks the limiting factor is often transient voltage changes. Power quality is an important consideration in the design and development of new large wind turbines and is one factor in the increasing popularity of variable speed operation and active stall control for fixed speed machines as both of these features reduce the transient variations in output power.

Figure 10.9 shows how power quality issues may be viewed with respect to wind generation. Figure 10.9a shows the various effects which may be considered to originate in the

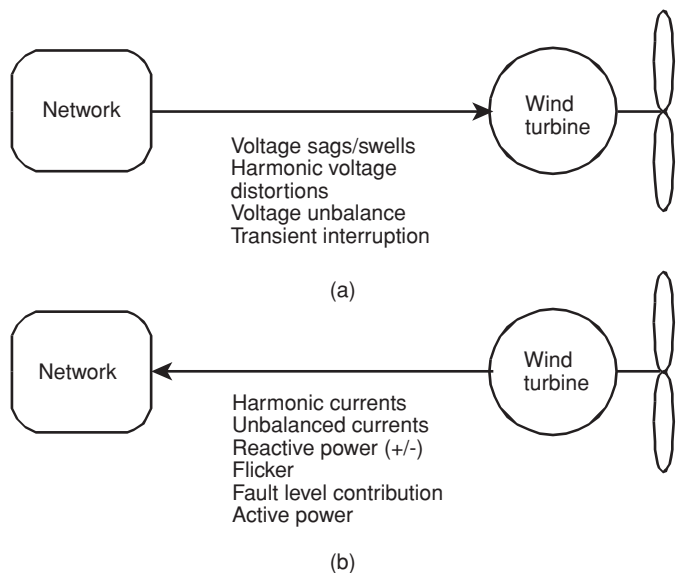


Figure 10.9 Origin of power quality issues

transmission and distribution networks and which can affect the voltage to which the wind turbines are connected.

Voltage sags, are a decrease to between 10% and 90% of nominal voltage for a period of up to one minute, usually caused by faults on the transmission or distribution network. They are of particular concern as they will cause fixed speed induction generator wind turbines to over-speed as the load on the generator is removed. This in turn can lead to a high demand for reactive power which further depresses the network voltage. The depressed voltage may also cause contactors to open and voltage sensitive control circuits to operate.

The control systems of variable speed wind turbines may mal-operate during voltage sags as they rely on Phase Locked Loop(s) to synchronise the operation of the power electronic switches with the network voltage. The voltage of the DC links of variable speed wind turbines will rise during network voltage sags as the wind turbine continues to generate power that cannot be injected into the grid due to its low voltage. Solutions include the use of a crow-bar resistor across the DC link of a DFIG wind turbine or allowing the aerodynamic rotor to speed up and store kinetic energy during the voltage sag.

Voltage swells (an increase in voltage over 110%) are less common and tend not to be a major problem for wind turbines. Ambient harmonic voltage distortion is increasing in many power systems, due to proliferation of power supplies of electronic equipment (TVs and PCs) and in the UK it is not uncommon to find levels at some times of the day in excess of that which is considered desirable by network planners (Electricity Association, Engineering Recommendation G5/4–1). Harmonic voltage distortions will lead to increased losses in the generators of wind turbines and may also disturb the operation of the control systems and harmonic current performance of power electronic converters. It is common practice to use power factor correction capacitors with induction generator, fixed speed wind turbines and these will have a low impedance to harmonic currents and the potential for

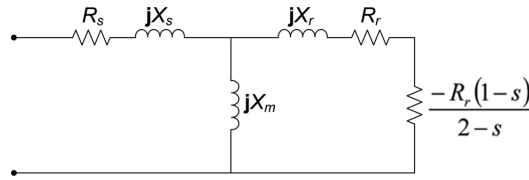


Figure 10.10 Negative phase sequence equivalent circuit of an induction machine. s = slip (negative for generator); R_s = stator resistance; X_s = stator reactance; X_m = magnetising reactance; X_r = rotor reactance (referred to the stator); R_r = rotor resistance (referred to the stator); j is the imaginary operator

harmonic resonances with the inductive reactance of other items of plant on the network. Transient interruptions (caused by the use of auto-reclosure on distribution networks) can potentially be very damaging due to the possibility of out-of-phase reclosure on to induction generators which are still fluxed, and, hence, developing a voltage. It is usual to apply fast acting loss-of-mains protection against this condition so that the turbine is isolated once the supply is interrupted.

Network voltage unbalance will affect fixed speed induction generators by increasing losses and introducing torque ripple (Electricity Networks Association, 2005; Engineering Recommendation P29). Voltage unbalance can also cause power converters to inject unexpected harmonic currents into the network unless their design has included consideration of an unbalanced supply. Voltage unbalance is usually described in terms of ‘negative phase sequence’ voltage. Using the method of symmetrical components (Weedy and Cory, 1998) the three-phase voltage phasors are split into a set describing balanced (positive phase sequence) and unbalanced operation (negative phase sequence).

Figure 10.10 shows the negative phase sequence equivalent circuit of an induction machine (Wagner and Evans, 1933). It may be seen that during normal operation (i.e. when the slip (s) is close to zero) the effective rotor resistance to negative sequence (unbalanced) voltages is only $R_r/2$. Thus, any small unbalance in the applied voltage will lead to high unbalanced currents. This has been found to be a particular problem for wind turbines connected to 11 kV distribution networks in the UK. It is common on rural 11 kV public supply networks to connect transformers and small spur lines between two phases only, to supply farms and other small loads. These single-phase connections tend to unbalance the voltages (up to 2% voltage unbalance is not uncommon) and so generate high unbalanced currents (up to 15–20%) in wind turbines connected to 11 kV three-phase circuits. The wind turbine control systems will detect excessive unbalanced current and cause a shut-down. Nuisance tripping often occurs in the night when heating loads in the farms are switched in automatically to take advantage of reduced price electrical energy tariffs.

Unbalance and single-phase voltage distortion can also lead to difficulties with variable speed wind turbines. The network side converters of variable speed generator systems synthesise a three-phase AC voltage from the DC voltage. This three-phase AC voltage is then used to inject current through a coupling reactance connected to the network. Hence, if the network voltage is distorted or all three voltages not balanced, unbalanced currents may be injected and the current waveform distorted. One solution is to use independent devices to detect the network voltages of each phase (i.e. an independent phase locked loop for each phase).

Figure 10.9b indicates how a distributed wind generator might introduce disturbances into the distribution network and so cause a reduction in power quality. Thus, a variable speed turbine using a power electronic converter may inject harmonic currents into the network. Unbalanced operation will lead to negative phase sequence currents being injected into the network which, in turn, will cause network voltage unbalance. Variable speed wind turbine generators can either produce or absorb reactive power while exporting active power and, depending on the details of the network, load and generation, this may lead to undesirable steady state voltage variations. Voltage flicker refers to the effect of dynamic changes in voltage caused by output power variations due to blade passing or other transient effects. Fixed speed wind generators will increase the network fault level and so will affect power quality, often to improve it. There is considerable similarity between the power quality issues of wind turbines and large industrial loads and, in general, the same standards are applied to both.

10.7.1 Voltage flicker

Voltage flicker describes dynamic variations in the network voltage which may be caused either by wind turbines or by varying loads (Bossanyi *et al.*, 1998; Tande, 2003; Tande in Ackerman, 2005). The origin of the term is the effect of the voltage fluctuations on the brightness of incandescent lights and the subsequent annoyance to customers (IEC 614000–4–15). Human sensitivity to variations of light intensity is frequency dependent and Figure 10.11 indicates the magnitude of sinusoidal voltage changes which laboratory tests have shown are likely to be perceptible to observers. It may be seen that the eye is most sensitive to voltage variations around 10 Hz. The various national and international standards for flicker on networks are based on curves of this type. The use of voltage flicker as an indicator of acceptable dynamic voltage variations on the network is rather unusual as the assessment is based on the experimentally measured effect of changes of the intensity of incandescent lamps on the human eye and brain. Incandescent lamps are now falling into disuse and are being replaced with more efficient types fed through power electronics. However, the flicker

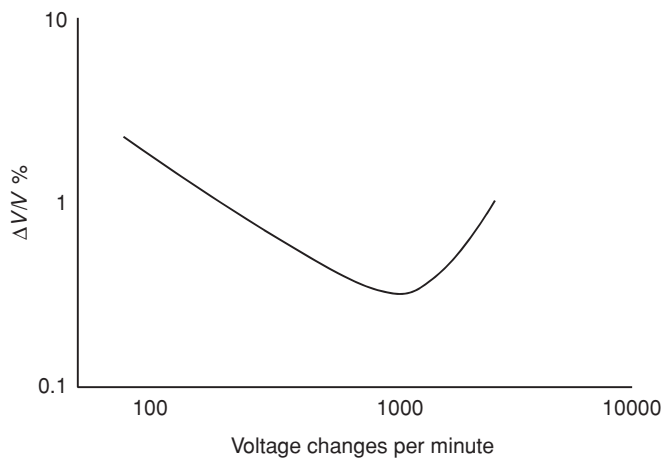


Figure 10.11 Influence of frequency on the human perception of sinusoidal voltage changes

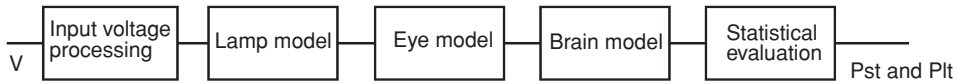


Figure 10.12 Principle of flicker measurement

standards are generally used to characterise transient voltage variations and are of considerable significance for distributed wind generation which (1) often uses relatively large individual fixed speed induction generator wind turbines, (2) may start and stop frequently and (3) is subject to continuous variations in input power from the fluctuating wind energy source.

Flicker is usually evaluated over a ten minute period to give a ‘short-term severity value’ Pst. The Pst value is obtained from a ten minute time series of measured network voltage using an algorithm based on the nuisance perceived by the human eye in fluctuating light. This is illustrated in Figure 10.12 which indicates how flicker is measured. Pst is linear with respect to the magnitudes of the voltage change but, of course, includes the frequency dependency indicated in Figure 10.11. Twelve Pst values may then be combined using a root of the sum of the cubes calculation to give a ‘long-term severity value’ Plt over a two hour period (Electricity Association Engineering Recommendation P28) although IEC 61400–21 makes no distinction between Pst and Plt limits.

If a number of wind turbines are subject to uncorrelated variations in torque then their power outputs and effect on network flicker will reduce as

$$\frac{\Delta P}{P} = \frac{1}{\sqrt{n}} \frac{\Delta p}{p}$$

where n is the number of generators, P and p are the rated power of the wind farm and wind turbine respectively and ΔP and Δp are the magnitude of their power fluctuation.

A range of permissible limits for flicker on distribution networks is given in national and international standards. Engineering Recommendation P28 specifies an absolute maximum value of Pst on a network, from all sources, to be 1.0 with a two hour Plt value of 0.6. However, extreme caution is advised if these limits are approached as the risk of complaints increases at between the sixth and eighth power of the change in voltage magnitude once the limits are reached and the approximate assessment method proposed in the same document is based on Pst not exceeding 0.5. BS EN 50160 is significantly less stringent specifying that over a one week period Plt must be less than 1 for 95% of the time. Gardner (1996) describes Pst limits from a number of utilities in the range 0.25–0.5.

10.7.2 Harmonics

Only variable speed wind turbines inject significant harmonic currents into the network. Fixed speed wind turbines, particularly those with power factor correction capacitors, alter the harmonic impedance of the distribution network and, in some circumstance, create resonant circuits. This may be important if fixed and variable speed wind turbines are installed in the same wind farm or in close proximity.

Thyristor soft-starts (Figure 10.13) are commonly used to connect the induction generators used on fixed speed wind turbines. Their mode of operation is initially for the thyristors to be fired late in the voltage cycle and then the firing angle advanced (over several seconds)

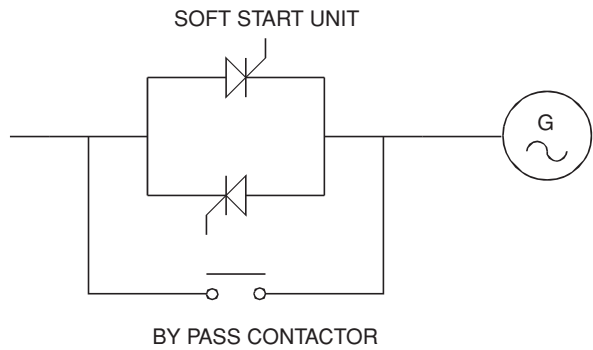


Figure 10.13 Soft-start unit for an induction generator (one phase only shown)

until the entire voltage wave is applied to the generator (Anaya Lara *et al.*, 2009). Thus, the network voltage is applied gradually to the generator and the current drawn by the generators controlled to reduce any voltage variations on the network. When operating, these devices produce varying harmonics as the firing angle of the thyristors is altered. Generally the soft-start units are only used for a few seconds during the connection of the induction generator and for this short period the effect of the harmonics is considered to be harmless and may be ignored (IEC 61400–21). If the anti-parallel thyristors are not bypassed, but left in service, then their harmonic currents need to be assessed. This continuous use of the soft-start unit has been proposed in order to reduce the applied voltage and hence the iron losses in the induction generator at times of low generator output. However, as well as the complication of trying to deal with harmonics which change with the thyristor firing angle there are also potential difficulties with the variation of the applied voltage to the generator altering the mechanical dynamic characteristics of the drive train.

Modern variable speed wind turbines use network side voltage source converters, normally based on the Graetz bridge topology, Figure 10.14. These use Insulated Gate Bipolar Transistors (IGBTs) switching at several kHz to synthesise a sine wave and so eliminate the

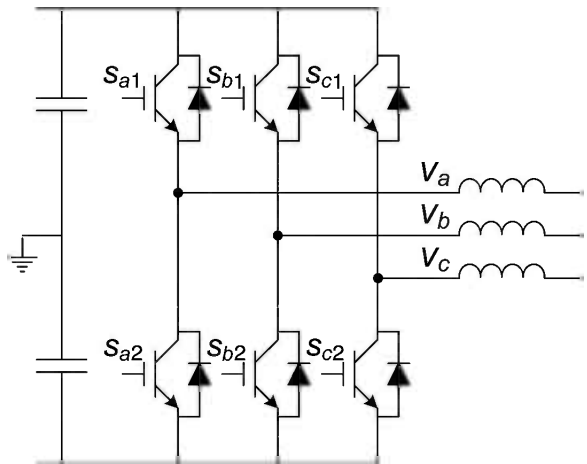


Figure 10.14 Six-pulse two-level IGBT voltage source converter

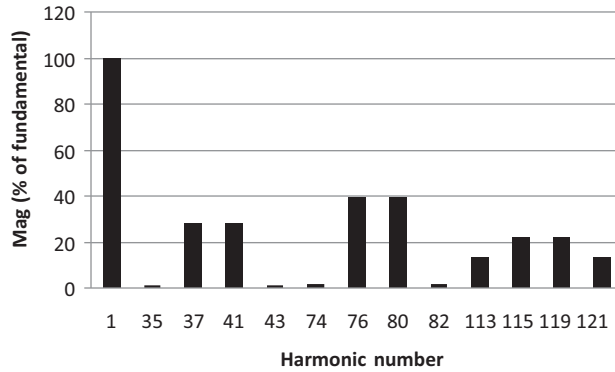


Figure 10.15 Typical harmonic spectrum of three-phase voltage of a PWM inverter. Carrier frequency 1950 Hz (39th harmonic). (After Mohan *et al.*, 2003)

lower order harmonics. The switching techniques used can vary from rather simple carrier modulated techniques which compare a reference signal with a trigger signal to those based on space-vector theory with some form of two-axis transformation (Anaya-Lara *et al.*, 2009). The rapid switching gives a large reduction in low order harmonic currents compared to the classical line commutated converter but the output current will have significant energy at around the switching frequency of the devices (i.e. in the 1–4 kHz region). Figure 10.15 shows the high frequency harmonics of a PWM inverter. Although these high frequency currents are relatively easy to filter they may influence the selection of the size of the coupling reactance which can be used to form part of the filter to block harmonic currents around the switching frequency.

10.7.3 Measurement and assessment of power quality characteristics of grid connected wind turbines

Determination of the power quality of wind turbines and prediction of their performance in service is not straightforward and IEC 614200–21 provides guidance. There are a number of difficulties when assessing the power quality of wind turbines as their performance will depend on:

- The design of the entire wind turbine (including the aerodynamic rotor and control system).
- The conditions of the electrical network to which it is connected.
- The wind conditions in which it operates.

For example, simple measurement of voltage variations at the terminals of a test turbine is not satisfactory as ambient levels of flicker in the electrical network and the X/R ratio of the source impedance at the test site will obviously have a great impact on the outcome. Hence, for evaluating flicker a procedure has been developed where current measurements are made of the output of a test turbine and used to synthesise the voltage variations which would be caused on distribution networks with defined short-circuit levels and X/R ratios

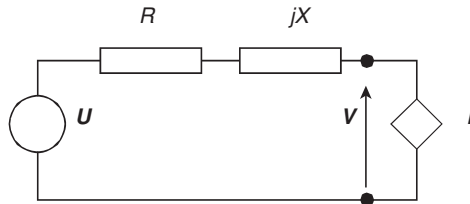


Figure 10.16 Use of ‘fictitious grid’ to establish voltage variations for various potential networks. I – measured current (complex quantity); R – resistance of fictitious grid; jX – inductive reactance of fictitious grid; U – ideal source voltage; V – synthesised voltage $|V|$ passed through flicker algorithm

of their source impedance. This is illustrated in Figure 10.16 and is referred to as simulation using a ‘fictitious grid’. These synthesised voltage variations are then passed through a flicker algorithm to calculate the flicker which the test turbine would cause on the defined networks. When the installation of the particular turbine is considered at a point on the real distribution network these test results are then scaled to reflect the actual short-circuit level and interpolated for the X/R ratio of the point of connection. A weighting factor, based on an assumed Raleigh distribution of wind speed, is also applied to provide flicker coefficients which may be used on sites with various average annual mean wind speeds.

IEC 614200–21 also defines methods to evaluate the impact of wind turbine start-up at cut-in and rated wind speeds and during speed changing of two speed generators. Again currents are measured, combined with the fictitious grid to provide a voltage time series and then passed through a flicker algorithm. For variable speed wind turbines, harmonic currents are measured over ten-minute observation periods.

10.8 Electrical protection

All parts of a high voltage power system are protected by relays which detect abnormal conditions and operate circuit breakers which then open to isolate the faulty circuits. Some lower voltage circuits are protected by fuses, as these are cheaper but do not give the degree of control offered by relays and circuit breakers. However, fuses do have the advantage of operating very rapidly (within a half cycle) and so limit the energy transferred into the short-circuit. On a distribution network the protection system is designed primarily to detect excess currents caused by insulation failure in the circuits, known as over-current protection. Failure of insulation, for example, breakdown of air or of solid insulation, leads to excess currents either between phases or between phases and ground. These high currents are only allowed to persist for up to about a second or so in order to limit the hazards which include: (1) risk to life caused by the arc or excess voltages as the high currents flow into the ground impedance, (2) risk to plant caused by the destructive heating and electro-magnetic effects of the high currents and (3) risk to the stability of the power system.

The electrical protection of wind turbines and wind farms follows the same general principles which are applied to any other items of electrical plant (AREVA, 1987) but there are two significant differences. Because wind farms are often connected to the periphery of

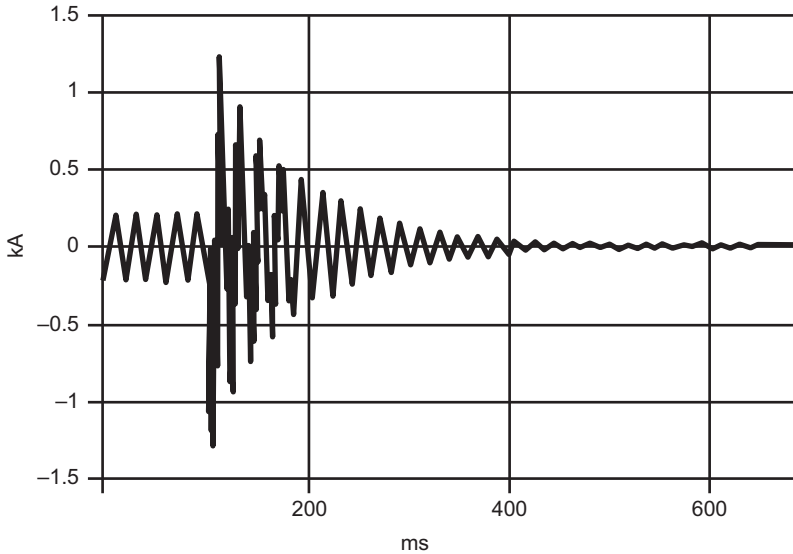


Figure 10.17 Fault current of induction generator with a three-phase fault applied to its terminals (phase shown with minimum DC offset)

the power system it is common to find that the fault currents which will flow in the event of insulation failure are rather small. Although this is desirable from the point of view of reducing hazards, this lack of current can pose significant difficulties for the rapid and reliable detection of faults. In particular, some designs of fuses rely on the energy within the arc for their correct operation. Hence, they cannot be relied on to interrupt small fault currents when the arc energy is low. Secondly, fixed speed wind turbines use induction machines and variable speed wind turbines are interfaced to the network through voltage source converters. Neither induction generators nor voltage source converters are a reliable source of fault current and so their fault current contribution cannot be detected reliably using over current protection. Thus, the faulty circuit is isolated using fault current from the network and then voltage or frequency sensing relays are used to detect the subsequent abnormal conditions, a technique known as sequential tripping.

Figure 10.17 shows the current of an induction generator with a three-phase fault applied to its terminals. It may be seen that the fault current dies away rapidly as the stored magnetic energy in the electrical machine decays. There is no sustained fault current as an induction generator draws its magnetising current from the network or local capacitors and this is not possible with the voltage collapsed by a three-phase fault at the terminals. Some asymmetrical faults (e.g. two-phase faults) can lead to sustained fault currents of two to three times that of full output, but again these are not usually relied on to operate protective relays. However, it may be necessary to ensure that fault current supplied by wind turbines does not lead to mal-operation of relays on the distribution system by altering the flow of fault current in the network and, in effect, providing local support of the network voltage.

One simple approach to the modelling of fixed speed induction generators for short-circuit calculations is to represent them as a one per unit voltage source in series with the

sub-transient reactance. This is a simple approach but may give a slightly low initial current as the internal voltage may be high if the induction machine is generating.

For a simple assessment, the response of a Doubly Fed Induction Generator to a local fault can be divided into two phases (Kanellos and Kabouris, 2009; Morren and de Haan, 2007). During the first phase (approximately two to three cycles), it can be considered to operate as a fixed speed induction generator but with an increased internal voltage particularly if it is operating super-synchronously and exporting power through both the rotor and stator circuits. In the second phase, the current through the rotor-side converter is limited by the control action of the crowbar protection and any rotor resistance that is applied.

In generators using full power converters the short-circuit current into the network is determined by the network side power electronic converter and normally does not significantly exceed 110% of its full load current. The full power converter wind turbine can, therefore, be approximated as a constant current source of approximately rated current, although IEC 60909 (IEC 60909-0:2001) recommends that three times the nominal current is assumed for the fault current contribution of variable speed drives that are able to regenerate.

The fault current contribution of all converter connected generators (DFIG and full power converter types) depends on the detailed design and control system of the converters. Hence test data from the manufacturers showing how the generator system responds to short-circuits and network voltage depressions should be used to construct models for use in fault calculators.

The protection of a wind farm has many similarities to the protection of a large industrial load equipped with machine drives which may back-feed into the network. This is a useful analogy, for the purposes of protection, as a wind farm can be thought of as a collection of large electrical machine drives with torque applied to the machine shafts rather than taken to drive mechanical loads. As in the case of industrial machine drives, the terminal voltages and frequency of the electrical machines are determined by the network. The distribution network provides a reliable source of fault current which can be used to detect insulation failure while there is also the possibility of a fault current contribution from the rotating machines. In addition, with a wind farm there is also the hazard of prolonged generation with abnormal frequency or voltage when the wind farm is disconnected from the rest of the power system. This is known as *islanding* and is an important consideration for all distributed generation schemes.

10.8.1 Wind farm and generator protection

Figure 10.18 shows a typical protection arrangement for a wind farm of fixed speed wind turbines with generator voltages of 690 V and with a collection circuit voltage of 11 kV. The 11 kV circuit is fed from a 33/11 kV Delta/Star wound transformer with the 11 kV neutral grounded either directly or through a resistor. The 11/0.69 kV transformers are also wound Delta/Star and so the 690 volt neutral points of each circuit may be directly grounded. The neutral points of the generators are not connected to ground.

There are a number of zones of protection. At the base of the wind turbine tower a 690 V circuit breaker (usually a moulded case type) will be fitted to protect the pendant cables and the generator. This is shown as zone D.

Zone C in Figure 10.18 is the 690 V cables running from the turbine transformer to the tower base cabinet. Fuses or another moulded case circuit breaker may be fitted to the 690 V side of the turbine transformer to provide protection of the cables and also a point of isolation so that all the electrical circuits of the turbine may be isolated without switching

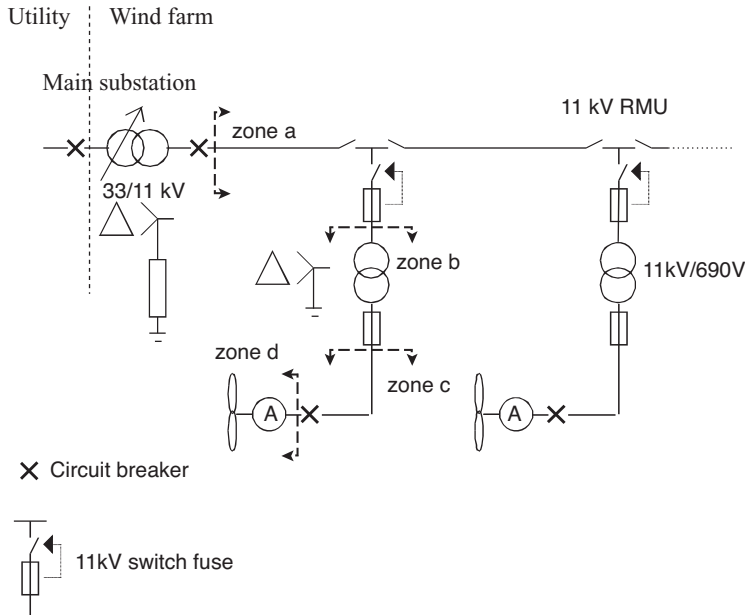


Figure 10.18 Protection of a wind farm with an 11 kV power collection circuit (RMU – Ring Main Unit)

at 11 kV. In some early designs of wind turbine, the main incoming busbar at the bottom of the electrical cabinet at the tower base consisted of exposed conductors. The electrical protection of this area needs careful consideration as there is the possibility of dropping tools or other equipment on to these busbars. Zone B is the 11 kV/690 V transformer including the region around its 690 V terminals. This is a particularly difficult zone to protect as the 11 kV fuses must be robust enough to withstand the magnetising inrush current of the transformer while sensitive enough to detect faults on the 690 V terminals when the fault current will be limited by the impedance of the transformer. This problem is common to the 11 kV/400 V transformers used on the public distribution network and a typical UK solution is to use an 11 kV combination fuse-disconnect (sometimes known as a switch-fuse). Faults on the 11 kV winding of the transformer will lead to high fault currents which are cleared by the 11 kV fuse. However, for faults on the low voltage terminals, the fuse is unable to clear these low fault currents effectively and so, once the fuse operates, a striker pin operates a mechanism to open the disconnect switch to clear all the phases. A more expensive solution adopted on some industrial installations is to use restricted earth fault protection to detect currents leaking to ground from the low voltage winding and terminal area but this requires a, more expensive, 11 kV circuit breaker. There is obvious commercial pressure to reduce the costs of transformer protection on a wind farm as there is a transformer for each wind turbine. However, safety considerations require that all credible faults can be detected and cleared.

Zone A is the 11 kV cable circuit and this is protected in the conventional fashion by over-current and earth fault relays operating an 11 kV circuit breaker. The 33/11 kV transformer is protected in a similar manner to a transformer of the same rating used on the public distribution system.

As wind turbine ratings increase and larger wind farms are constructed, 11 kV power collection circuits cease to be cost-effective and so large wind farms are now constructed with a collection voltage of 33 kV (in the UK). This arrangement poses a number of additional difficulties for the electrical protection.

In the arrangement of Figure 10.18 the 33/11 kV transformer provides two useful features for electrical protection. Its impedance allows easier grading of over-current relays and, as it blocks the passage of zero-sequence current, fast acting earth fault relays can be applied to the wind farm circuit. With direct connection to the utility circuit these desirable features are not available.

Although 33 kV switch fuses are available it can be difficult to provide comprehensive protection of the 33 kV/690 V transformer for the full range of prospective fault currents. Effective protection for a single phase to ground fault on the low voltage terminals is particularly difficult and one approach is to use circuit breakers on both the 33 kV and 690 V sides of the transformers.

Electrical protection of wind farms follows conventional distribution engineering practice with the network as the main source of fault current. In the protection of a public supply network it is important to isolate only the faulty section or circuit and so maintain supply to as many customers as possible. This discrimination of the protection is less important in a wind farm as only some additional loss of generation will occur if correct discrimination is not achieved and so simpler and lower cost protection may be appropriate. The difficulty remains, however, of ensuring that effective protection is installed to detect all credible fault conditions, even with limited prospective fault currents, and to be able to identify the faulted section of circuit.

10.8.2 Islanding and self-excitation of induction generators

Fixed speed wind turbines use induction generators to provide damping in the drive train and, as there is no direct access to the field of an induction generator, the magnetising current drawn from the stator leads to a requirement for reactive power. In order to reduce the reactive power supplied from the network it is conventional to fit fixed speed wind turbines with local power factor correction (pfc) capacitors. As long as the induction machine is connected to a distribution network its terminal voltage is fixed and so the pfc capacitors merely reduce the reactive power drawn from the network. However, once the induction generator is isolated from the network then there is the possibility of a resonant condition, known as self-excitation, leading to large over-voltages (Hindmarsh, 1984). When the generator is isolated from the network it will tend to accelerate as its load is removed. This increase in rotational speed and consequently of frequency adds to the possibility of self-excitation. There have been a number of cases reported of sections of distribution network, to which wind turbines are connected, becoming isolated and over-voltages resulting in damage customers' equipment.

Figure 7.28 showed the conventional equivalent circuit of an induction machine with the pfc capacitors connected at the terminals (McPherson, 1990). At normal running speed and frequency, the slip (s) is small and so the impedance of the rotor branch is high and, to a first approximation, can be considered an open circuit. As the stator impedance ($R_s + jX_s$) is much smaller than the magnetising reactance jX_m then the equivalent circuit may be simplified to a simple parallel connection of the pfc capacitors and the magnetising reactance. Thus, the equivalent circuit of Figure 7.28 may be reduced to that of Figure 10.19. This is a conventional LC parallel circuit but in this case X_m is a non-linear function of voltage



Figure 10.19 Reduced equivalent circuit of an induction generator to illustrate resonance leading to self-excitation

due to magnetic saturation. Energy is added to the circuit from the wind turbine. An indication of the likely voltage which will occur as the wind turbine rotor accelerates can be gained by considering the intersection of the capacitive and inductive voltages as the current is identical in each element of this simplified model. This is shown in Figure 10.20. Point X indicates the self-excitation voltage at frequency f_1 , say 50 Hz, while point Y illustrates the self-excitation voltage at an increased frequency, say 55 Hz. Such simple calculations are indicative only as they are based on a very simple representation of the induction generator and even then it is difficult to obtain reliable data for the saturation characteristic of the machine. However, as self-excitation is a most undesirable condition for grid connected wind turbines, precise calculations of the voltage rise are seldom required.

Self-excitation can be avoided by limiting the power factor correction capacitance, including any capacitance of the distribution network, to a value which will not lead to the resonant condition at any credible frequency which will be experienced during overspeed. Alternatively, the potential for this condition must be recognised and fast-acting over-voltage and over-frequency protection arranged to disconnect the capacitors and stop the wind turbine if islanding occurs.

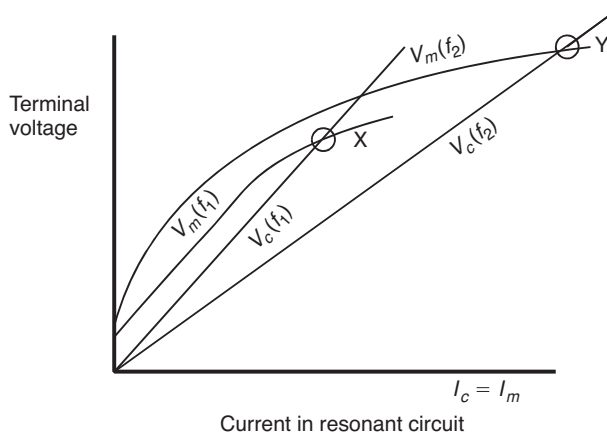


Figure 10.20 Illustration of self-excitation at two frequencies f_1 and f_2 . I_c – current in capacitors; I_m – current in magnetising reactance of induction machine

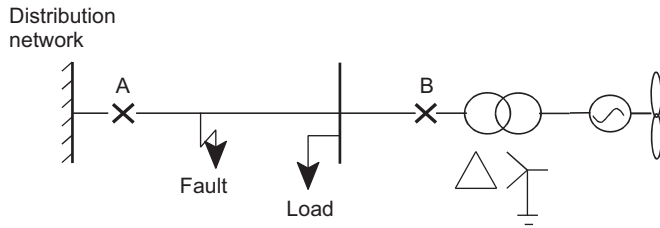


Figure 10.21 Protection of the distribution network from a wind turbine

10.8.3 Interface protection for wind turbines connected to distribution networks

Section 10.8.1 considered the protection of the wind turbines from the effects of insulation failure and subsequent high fault currents, which were supplied by the distribution network. However, protection is also required to ensure that a wind farm does not feed into faults on the distribution network or attempt to supply an isolated section of network.

The problem is illustrated in Figure 10.21. For faults on the network, the difficulty is that wind turbines are not a reliable source of fault current and so circuit breaker B may not be opened by over-current protection. Thus, for the fault shown the current operated protection on the network is used to open circuit breaker A. This then isolates the wind turbine which begins to speed up as the wind input remains but it is no longer possible to export power to the network. In fact, this acceleration begins as soon as the fault occurs, and before circuit breaker A trips, as the fault depresses the network voltage and so restricts the export of power. Circuit breaker B is then opened by over-frequency relays which detect the increase in rotational speed of the wind turbine, and hence the increase in frequency of the output, or by under/over-voltage relays which detect that the connection to the network has been lost by monitoring the change in voltage. It is common for the over/under frequency and voltage relays to be time delayed by, for example 500 milliseconds, to reduce spurious operation. Hence in high winds it is possible that the wind turbine will shut down on mechanical over speed protection which is set to limit the speed of the aerodynamic rotor.

This sequential tripping using voltage and frequency sensitive relays which operate once the generator has been isolated is not generally considered to be good practice in the protection of conventional power systems. However, with the use of induction generators or voltage source converters there is little option and so the arrangement is generally accepted for wind farms.

The issue of *islanding* is the possibility that the output of the wind turbine (in terms of both real and reactive power) matches precisely the local load. In this case even though circuit breaker A opens there will be no change in the voltage or frequency of the wind turbine and so circuit breaker B will not operate. Most distribution utilities are extremely sensitive to possible islanded operation for a number of reasons:

- The possibility that customers may receive supply outside the required limits of frequency and voltage.
- The possibility that part of the network may be operated without adequate neutral earths.
- The danger associated with out-of-phase reclosing.

- That it is against the regulations governing operation of the distribution network.
- The potential danger to staff operating the distribution network.

From the point of view of the wind farm operator the main danger is in out-of-phase reclosing. Many distribution circuits have automatic reclosing to allow transient faults, particularly on overhead lines, to be cleared without prolonged interruption to customers. Thus, circuit breaker A may be arranged to reclose up to a few seconds after opening. If the circuit breaker B is still closed then very high currents and torques will occur as the network voltage is applied out-of-phase to the wind turbine. This is the main technical driver for fast islanding protection as, particularly with wind turbines, it is very unlikely that the required matching of generator output (both real and reactive power) to the load will be sustained for any length of time.

Considerable efforts have been made to devise robust protection systems to detect islanding. However, it may be seen to be very difficult as, by the principle of superposition, if there is no current flowing in circuit breaker A then it makes no difference to the network conditions at the wind turbine whether it is open or closed. The commonly used relays are the so-called rocof (rate of change of frequency) devices or vector-shift relays which measure the jump in the voltage vector when islanding occurs. Both relays rely on some current (and hence real power) flowing in circuit breaker A and can be adjusted to varying levels of sensitivity. However, if they are set too sensitive then they tend to be prone to spurious nuisance tripping due to external disturbances on the power system (e.g. the tripping of a large remote conventional power plant).

A further complication occurs if the fault shown in Figure 10.22 is not between phases but is between a single phase and ground. Faults of this type are particularly common on overhead lines. It can be seen that the turbine transformer (e.g. 33/0.69 kV) has a Delta winding on the High Voltage side. This has no accessible neutral point and no connection to ground is made. Thus, there is no path in which the ground current may flow and so, in principle, the

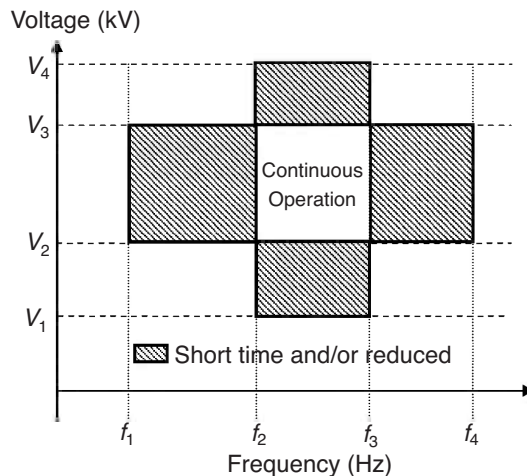


Figure 10.22 Grid Code – continuous operation. Typical shape of continuous and short-time operating regions. (Anaya-Lara, 2009)

single-phase earth fault will remain indefinitely with no current flowing. In practice some stray capacitive currents will flow but these will not be enough to operate conventional earth fault protection and will, in fact, lead to intermittent arcing. The conventional solution is to use a neutral voltage displacement relay to detect that one phase of the circuit is connected to ground and so the neutral is displaced. The disadvantage of this scheme is its cost as a complex (five limb) voltage transformer is required on the high voltage circuit. Although this can be accommodated for a large wind farm the expense is sometimes difficult to justify for individual wind turbines.

The requirements for interface protection vary widely between different countries. Some countries favour the use of transfer tripping whereby when any upstream utility circuit breaker is opened this action is communicated to the wind farm circuit breaker which is then immediately opened. Although this provides a guarantee against islanded operation it can be expensive to implement as communication channels from a number of remote circuit breakers are required. It is interesting to note that in Holland, where almost all the distribution system is underground and so auto-reclose is not used, there is no requirement for loss-of-mains protection. In Denmark, positive phase sequence under-voltage relays are used extensively and appear to be effective in detecting islanded operation. No doubt over time practices will converge but, at present, very considerable national differences remain.

10.9 Distributed generation and the Grid Codes

Wind turbines are connected individually and in small groups to the medium voltage distribution network or in large wind farms to the high voltage distribution and transmission network. The connection rules and performance required by the power system operator depend on the size of the wind farms and the voltage of the network to which they are connected.

Individual wind turbines and small wind farms connected to the distribution network come under regulations such as G59/1 in the UK, for generators up to 5 MW and 20 kV, (Electricity Network Association, 1991) and IEEE 1547, for groups of generators up to 10 MVA, in the USA (IEEE, 2003). The philosophy of these regulations is that the wind generation is too small to have any significant impact on the frequency of the main power system. Thus, these regulations require that the wind turbines disconnect if there is any disturbance in the power system or if a section of the network becomes islanded. When the network voltage or frequency is outside the normal operating range the turbines are tripped and not reconnected until normal network conditions are restored. Sensitive loss of mains protection is used to detect islanding.

The thrust of these regulations reflects the concerns of the distribution network operators to avoid islanded operation and to ensure there is no possibility of out of phase re-closure, back feeding sections of the network with the possible safety hazards or operating an ungrounded system. The stability of the power system is provided by the large central generators and all wind turbines falling under these regulations must operate with power factor control taking their voltage from the network. There is no requirement for real power control and the wind turbines make no contribution to maintaining system frequency.

In contrast, larger wind farms fall under a different set of regulations, known as the Grid Codes. These requirements are imposed by the transmission system operators and reflect their concern that larger wind farms have a significant effect on the main power system. Here the philosophy is that rather than disconnect if the voltage and frequency of the power system

are outside their normal limits, perhaps because of a system fault or loss of a large generator, the wind farms should stay connected to support the power system. In addition the wind farms are required to have the facility to control their terminal voltage and to be able to provide frequency support. These wind farms are considered large enough to influence the transmission system voltage through their output of reactive power and the network frequency by their generation of real power.

The size of wind farm and voltage level at which these two rather different approaches change reflects the relative significance of wind generation compared to the size of the conventional generation system. Thus, the GB Grid Code applies to wind farms above 50 MW in England but 10 MW in parts of Scotland. In the USA, Texas with its large wind farms has a well developed transmission Grid Code while some other states with limited wind energy at present do not. As wind energy continues to increase, it is likely that versions of the Grid Codes will apply to smaller wind farms to ensure they contribute to the operation of the power system. However, operating practice of distribution networks will then have to be revised.

Many countries have their own national standards for the connection of small distributed generation and their own Grid Codes for large wind farms. These are continually evolving as the importance of wind generation and other forms of distributed generation increase and efforts are continuing to harmonise these regulations. The principles of these documents are discussed with examples that are illustrative only.

10.9.1 Grid Code – continuous operation

The Grid Codes follow from similar requirements that apply to the connection of large fossil and hydro central generators. Thus, they do not address individual wind turbines in a wind farm but only the performance of the entire farm at the point it is connected to the transmission network.

Large wind farms are required to be able to operate continuously over the range of frequency and voltage that might be expected on the power system and not to trip during system disturbances. This is to ensure that they continue to provide energy into the power system when the transmission network voltage and frequencies are disturbed. Figure 10.22 shows the typical of the requirements for large wind farms connected to High Voltage networks. The precise values of frequency and voltage vary country by country. There have been recorded examples in both Britain and mainland Europe of a major system frequency drop, caused by the simultaneous loss of central generators or circuits, being made worse by the tripping of distributed generators that were protected against islanded operation by sensitive frequency, voltage or rate of change of frequency relays. Continuous operation over a range of frequencies and voltages ensures that large wind farms contribute to the stability of the power system.

10.9.2 Grid Code – voltage and power factor control

The voltages of a transmission network are determined mainly by the flows of reactive power. Large wind farms are required to be able to operate either by controlling their terminal voltage or exchanging a defined level of reactive power with the transmission network. A typical requirement is shown in Figure 10.23. The wind farm is required to be able to operate anywhere within this area as instructed by the transmission system operator.

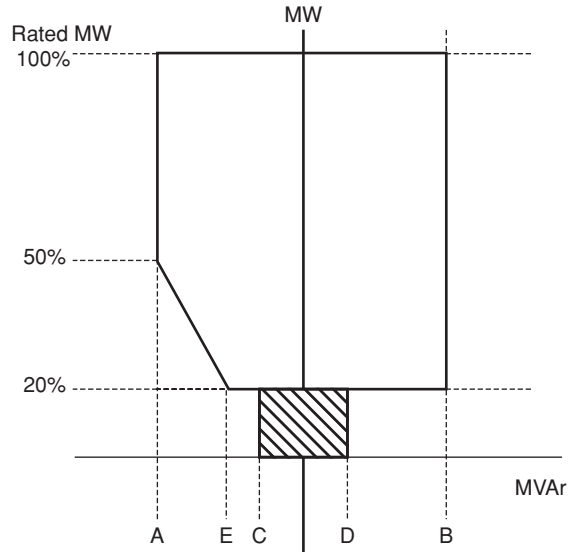


Figure 10.23 Grid Code – voltage and power factor control. Typical reactive power requirement of a grid connected wind farm (National Grid, 2008). Point A is equivalent (in MVar) to: 0.95 leading Power Factor at Rated MW output; point B is equivalent (in MVar) to: 0.95 lagging Power Factor at Rated MW output; point C is equivalent (in MVar) to: -5% of Rated MW output; point D is equivalent (in MVar) to: $+5\%$ of Rated MW output; point E is equivalent (in MVar) to: -12% of Rated MW output.

Variable speed wind turbines can, in principle, operate at any power factor within the rating of their power electronic equipment. In practice it is difficult to ensure rapid compliance with the requirements for the reactive power output of a large wind farm because of the:

- Difficulty of rapidly coordinating the reactive power generation of a large number of geographically dispersed wind turbines.
- Reactive power drawn by the wind farm transformers which varies with both real and reactive power transmitted.
- Increase in costs of the wind turbines if a large reactive power range is specified.

Hence it is common for additional reactive power compensation equipment (e.g. a STATCOM) to be installed at the point of connection of the wind farm to the network to assist in meeting this Grid Code requirement.

10.9.3 Grid Code – frequency response

The frequency of a power system is determined by the instantaneous balance of real power generation and load. If the load exceeds the generation the frequency will drop and if there is an excess of generation the frequency will rise. The rate of change of frequency is determined by the spinning inertia of all generators and spinning loads connected to the power system.

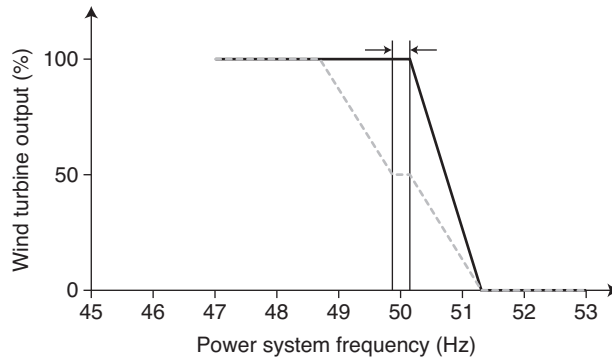


Figure 10.24 Grid Code – frequency response: high frequency response black line, de-loaded turbine grey line

Thus, if the frequency of the power system increases, perhaps because a large load is disconnected, the wind turbines must reduce their real power output (Figure 10.24).

In order to provide low frequency response the wind turbines must be de-loaded either by adjusting their blade pitch angles or by operating at off-optimal rotational speed. Several approaches to active power control including delta control and gradient power control are discussed in *Wind Energy – The Facts* (European Wind Energy Association, 2009). The requirement for de-loaded operation is specified in many of the Grid Codes but is rarely called on by the power system operator as it results in wind energy being wasted.

10.9.4 Grid Code – fault ride through

Short-circuit faults on the transmission network cause the voltage to be depressed over a wide area. Faults are inevitable in any power system, for example due to lightning, and the voltage depressions may extend for hundreds of kilometres. Circuit breakers will clear transmission faults in approximately 100 milliseconds by isolating the faulted section of network and the network voltage then recovers but faults on the lower voltage networks take longer to clear.

A network fault may result in the isolation of a circuit and loss of central generation and it is just at this time that the wind farms should continue to provide energy into the power system. The wind farms should remain stable and connected during any network disturbance and so are required to exhibit an ability to ride through voltage depressions caused by short-circuit faults.

The power that any generator can export is the product of voltage and current and so if its terminal voltage is depressed the generator cannot export its power to the network and will over-speed. The mechanical energy is stored as rotational kinetic energy in the generator and its prime mover. When a fixed speed induction generator wind turbine over-speeds due to a network fault and the voltage is restored it will draw large amounts of reactive power which results in a further voltage depression and possible voltage instability.

A variable speed wind turbine is better able to ride through voltage depressions and a transient reduction in output power by allowing the rotor speed to increase. However, a robust control system is needed to ensure that the converters remain synchronized with the network during the voltage depression.

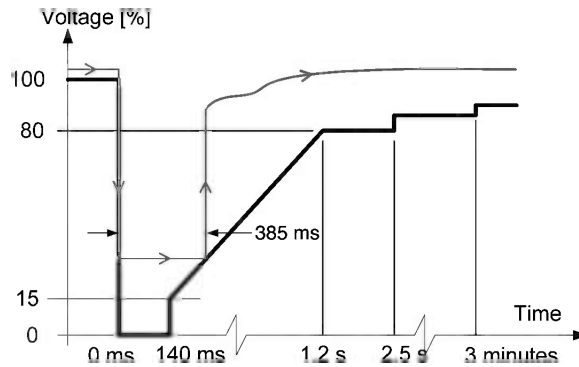


Figure 10.25 Grid Code – fault ride through. Typical fault ride through characteristic

Figure 10.25 shows the voltage time envelope within which a wind farm must continue to operate. The wind farm must withstand a drop to zero volts for 140 milliseconds and a reduction to 80% volts for 1.2 seconds. An example of a voltage depression caused by a fault is also shown. In this example, the fault is cleared within 385 ms and the wind farm continues to operate through this network fault.

10.9.5 Synthetic inertia

The conventional control system of a variable speed wind turbine does not consider the power system frequency. The frequency is used to synchronize the switching of the power electronics of the network side converter but the main control loop measures the rotational speed of the generator and applies a torque so that the wind turbine follows its pre-determined operating characteristic (Figures 6.18 and 6.19). Thus, in the event of a drop in power system frequency, caused, for example by the sudden disconnection of a large central generator, a variable speed wind turbine will not provide any additional energy as the system frequency falls. This is in contrast to a conventional synchronous generator, or a fixed speed induction generator, that will transfer some of its kinetic energy to the power system as the frequency and the speed of rotation of the generator falls.

This lack of response of variable speed wind turbines to a drop in system frequency can be overcome by adding additional control loops as shown in Figure 10.26. The inertia is synthesized by measuring the rate of change of system frequency. The magnitude of the frequency drop may also be used to apply additional torque to the rotor, slow it down and so transfer kinetic energy to the network.

The provision of synthetic inertia is required by some, but not all Grid Codes.

10.10 Wind energy and the generation system

Throughout the world, most electricity is generated using large synchronous electrical machines (up to 1000 MW capacity) driven by turbines from fossil, hydro or nuclear energy. Until a few years ago, the amount of wind energy being generated was so small that

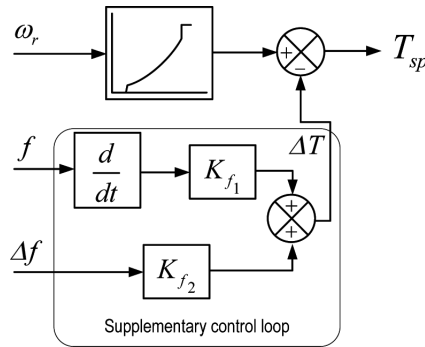


Figure 10.26 Additional control loops to provide synthetic inertia to a variable speed wind turbine T_{sp} is the set point of the torque controller of the power electronic converters. (Anaya-Lara, 2009)

it acted merely to reduce the electrical load seen by these generators and had no significant impact on either the planning or operation of the generation system. This is now changing rapidly with wind energy becoming significant both for the development (planning) of the generation system and its operation.

In many countries, the electricity system has been liberalized and separated into companies dealing with generation, energy retail, transmission and distribution. Some parts of a liberalized electricity system operate as a market (generation and energy retail) while others are natural monopolies (transmission and distribution) and are regulated by the state for the benefit of customers. Although the market influences the way generators and energy retailers behave in the short term the underlying operation of the electrical power system remains unchanged and the market and regulation can be considered as merely mechanisms to discover the most efficient way for the various actors to behave.

10.10.1 Capacity credit

All generating systems (and indeed all aspects of the power system) are designed assuming a probability of failure. All pieces of plant (e.g. generators, transformers, overhead lines and cables) have a finite probability of failure and any design for a completely reliable system would be infinitely expensive. Traditionally the probability that load would exceed the available generation has been defined in terms of generation adequacy. Before liberalisation, the England and Wales generating system was designed to a Loss of Load Probability (LOLP) of 0.09. This meant electrical demand was expected to exceed available generation on average nine times a century. A LOLP of 0.09 with the mainly coal generating system of England and Wales in the 1980s required a generation planning margin of 24% (i.e. the generating capacity built would exceed the peak load by 24%). The French system was designed to a Loss of Load Expectation (LOLE) of three hours per year while the Eire system assumed demand would exceed generation for eight hours per year. In North America a LOLP of 0.1 or LOLE of 2.4 hours/year were common. Similar criteria were used by other major generating utilities.

This principle of planning for a defined level of adequacy is illustrated in Figure 10.27 where the probability distributions of the load and generation are shown. The risk of

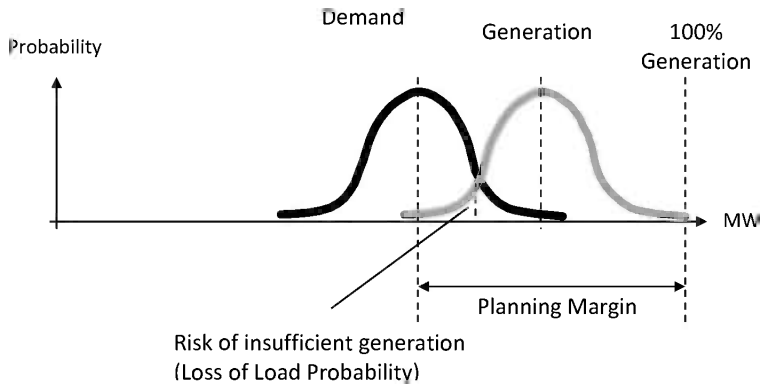


Figure 10.27 Generation adequacy. (Jenkins, Ekanayake and Strbac, 2010)

insufficient generation is shown by the intersection of these probability distributions. All generators have a probability of not being available due to breakdown but wind generation has a higher probability of not generating than fossil generation as it may not generate because of low wind speeds as well as equipment failure. Thus, adding wind energy to the generation portfolio has the effect of widening the probability distribution of generation and, hence, increasing the generation planning margin.

Calculations to establish the contribution of wind energy to generation adequacy of a power system are undertaken in a similar manner to those for fossil generators (Billinton and Allan, 1996) except that rather than consider only one of two states of a conventional generator (available or not available) wind generators are represented by a number of states to reflect the probability of generation at various wind speeds. These states and their associated probabilities are used to construct a Capacity Outage Probability Table and so calculate the probability of generation being available. As the wind generation is added, conventional generation is removed until the same adequacy is achieved. This approach then allows the capacity value of wind energy to be calculated.

The capacity value of wind generation is described by its *capacity credit* (Milligan and Parsons, 1999) which is defined as the reduction in the requirement for capacity of conventional plant, made possible by the introduction of wind energy, while maintaining reliable supplies of electricity. The capacity reduction of conventional generation is expressed as percentage of the wind generating capacity in the system. In summary, the capacity credit of wind is assessed through quantification of the generation adequacy of the generation system with and without wind power under a standard reliability criterion (e.g. LOLP or LOLE).

Figure 10.28 shows how the capacity credit of wind energy varies with the penetration of wind energy on the Great Britain power system. It may be seen that at low penetrations the capacity credit is close to the wind energy capacity factors and then reduces with increasing penetration. The benefit of geographical diversity is shown clearly with geographically diverse wind farms giving a higher capacity credit than if they are all located in the same area.

10.10.2 Wind power forecasting

The electrical power system is operated by balancing supply and demand on a second-by-second basis. Thus, as well as an appropriate capacity of generation being constructed it needs

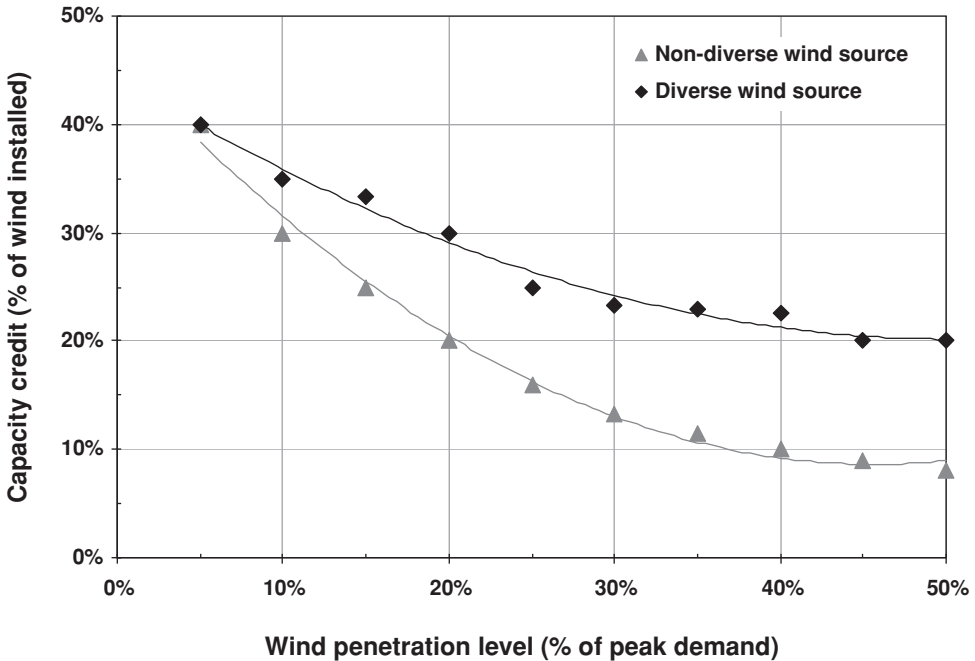


Figure 10.28 The capacity credit of wind generation versus the level of wind penetration in the Great Britain electricity system. (Jenkins, Ekanayake and Strbac, 2010)

to be operated to meet the load. This is known traditionally as Unit Commitment and is the selection of the optimum (minimum cost or minimum emission) choice of generators to meet the load within constraints. The constraints include the generator operational constraints (e.g. minimum up and down times of thermal generators), network constraints (e.g. loading of circuits) and the requirement to maintain adequate reserves (e.g. partially loaded spinning generators). Once the generators to be connected to the network are selected, their most economic output is determined through Economic Despatch. Unit Commitment and Economic Despatch together are often referred to as generation scheduling.

In a traditional vertically integrated electricity system, the scheduling of the generation system was optimised centrally by the system operator. In a market based system each generating company will optimise their own portfolio of generators with the objective of meeting the contracts they have struck with energy retailers. However, the principles of having to commit plant, often up to eight hours in advance for large thermal generators, and then optimise its operation remains.

The objective is always to meet the demand at minimum cost (or emissions) and so knowledge of the demand is critical. If historic demand and factors such as ambient temperature and TV programs are taken into account it is possible to forecast electrical load demand with considerable accuracy (typically within 3%). However, the introduction of large quantities of wind energy into the power system significantly increases uncertainty of the level of residual load that will be needed by other generations. Hence, forecasting of wind power output becomes extremely important for power system operation (Boyle, 2009).

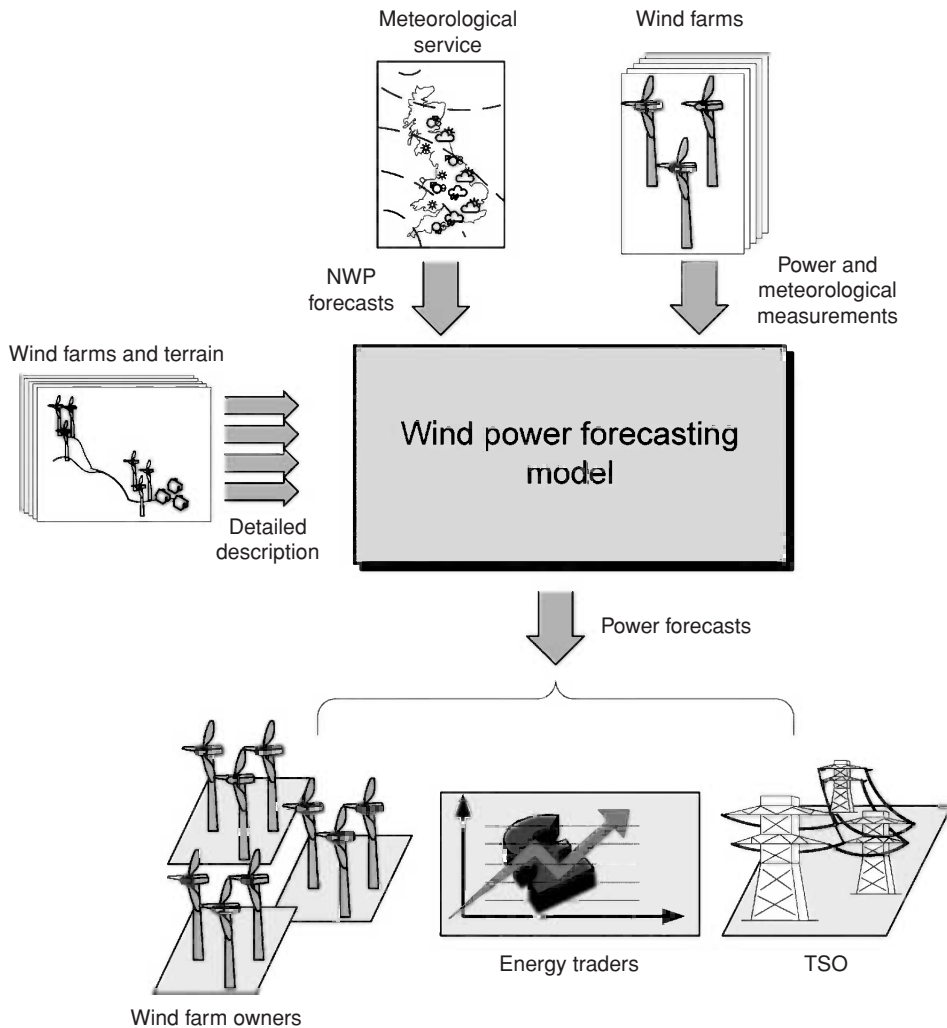


Figure 10.29 Flows of information in a wind power forecasting system. (After, Fox *et al.*, 2007)

Effective forecasting of wind power output is required for:

- The scheduling of generators and balancing supply and demand.
- Determining the reserve generation needed, either part-loaded generators or fast-start gas turbines. This requires a prediction of the accuracy of the forecast.
- Effective operation of the transmission system and managing congestion on circuits.

Forecasts of up to four to six hours ahead may be based on measured data (wind speeds or wind farm power output) using analysis of the previous time series. A large number of techniques have been investigated and modest improvements over persistence can be achieved. The persistence model assumes that the forecast is the same as the last measured value.

All predictions with a longer time horizon are based on Numerical Weather Prediction (NWP) using weather forecasts. Global NWP models are operated by a number of national weather services. They use meteorological observations from weather stations and satellites together with a numerical model to predict the future state of the earth's atmosphere. They are very computationally intensive and so use a rather coarse spatial mesh. Local area NWP models take as their input the results of the global model to provide higher resolution but over only a part of the earth. These local forecasts, which may be purchased from the national weather service, are used as an input of the wind power forecasting model (Figure 10.29).

The wind speed forecasts from the local NWP model are provided on a grid perhaps 20 km square, every eight hours over a 48-hour time horizon. These are then translated (downscaled) to the wind farm site at the height of the turbines. In flat terrain this may be done by spatial interpolation but in more complex terrain a meso-scale model is used. Using this site wind speed, the wind farm output is then modelled taking into account local topography and loss of energy in wind turbine wakes. Model Output Statistics (MOS) are then used to refine the model by comparing its outputs with measurements from the wind farms of power output and wind speeds.

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Appendix A10 Simple calculations for the connection of wind turbines

A10.1 The Per-unit system

The per unit system (Weedy and Cory, 1998) is simply a technique used by electrical power engineers to simplify calculation by expressing all values as a ratio of:

- actual value (in any unit);
- base or reference value (in the same unit).

Its advantages include: (1) a reduction in the appearance of $\sqrt{3}$ in the calculations (2) that similar per unit values apply to systems of various sizes and (3) by appropriate choice of voltage bases the solution of networks containing several transformers is facilitated.

For simple calculations, such as those required for assessment of a wind turbine or small wind farm on a simple distribution circuit, all that is required is to:

1. Assume an arbitrary base VA (e.g. 10 MVA for a small wind farm connection).
2. Select voltage bases of each voltage level of the network (e.g. 33 kV and 11 kV). These voltage bases should be related by the nominal turns ratios of the transformers.
3. Calculate the appropriate real and reactive power flows at the generator terminals in per unit (i.e. for a Power flow of 5 MW with the base VA chosen as 10 MVA, the per unit value is $P = 5/10$ or 0.5 per unit. Similarly for a Reactive Power flow of 1 MVAR the per unit value is $Q = 1/10$ or 0.1 per unit).
4. If necessary transform the impedances of the circuit from ohmic to per unit values using a base impedance of $Z_{base} = V_{base}^2/VA_{base}$.
5. The base current at any particular voltage level is calculated from: $I_{base} = VA_{base}/\sqrt{3} \times V_{base}$.

A10.2 Power flows, slow voltage variations and network losses

If the output from a distributed wind turbine generator is absorbed locally by an adjacent load then the effect on the distribution network voltage and losses is likely to be beneficial. However, if it is necessary to transport the power through the distribution network then increased losses may occur and slow voltage variations may become excessive (Jenkins, Ekanayake and Strbac, 2010).

If the wind generator operates at unity power factor (i.e. Reactive Power $Q = 0$), then the voltage rise in a lightly loaded radial circuit (Figure A10.1) is given approximately by:

$$\Delta V = V_1 - V_0 = PR/V_0 \quad (\text{A10.1})$$

Operating the generator at a leading power factor (absorbing reactive power) acts to reduce the voltage rise but at the expense of increased network losses. In this case the voltage rise is given by:

$$\Delta V = V_1 - V_0 = (PR - XQ)/V_0 \quad (\text{A10.2})$$

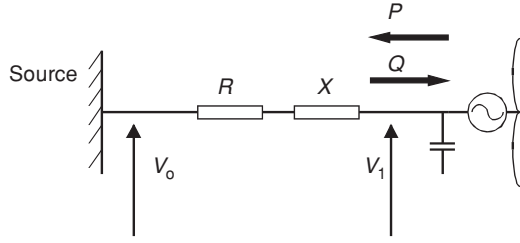


Figure A10.1 Fixed-speed wind turbine on a radial circuit

The impedance of an overhead 11 kV distribution circuit may, typically, have a ratio of inductive reactance to resistance (X/R ratio) of 2. An uncompensated induction generator at rated output, typically, has a power factor of 0.89 leading, that is, $P = -2Q$. Thus, under these conditions, there is no apparent voltage rise in the circuit at full power. However, the real power loss (W) in the circuit is given approximately by:

$$W = (P^2 + Q^2)R/V_0^2 \quad (\text{A10.3})$$

The reactive power drawn by the generator acts to limit the voltage rise but higher real power losses are incurred in the connecting circuit.

Equations A10.1–A10.3 for voltage rise and circuit losses are approximate only and do not apply to heavily loaded circuits. A simple but precise calculation for voltage rise in any radial circuit may be carried out using an iterative technique.

Complex quantities are indicated by bold type.

At the generator busbar of Figure 10.9 the *apparent power* (sometimes known as the *complex power*) $\mathbf{S}_1$ is given by:

$$\mathbf{S}_1 = P - jQ \quad (\text{A10.4})$$

(for a generator operating at lagging power factor, exporting VARs, $\mathbf{S}_1$ would be given by $P + jQ$)

By definition $\mathbf{S} = \mathbf{VI}^*$. * indicates the complex conjugate (i.e. the sign of the imaginary component of the complex number is reversed).

Therefore, the current flowing in the circuit is given by:

$$\mathbf{I} = \mathbf{S}_1^*/\mathbf{V}_1^* = (P + jQ)/\mathbf{V}_1^* \quad (\text{A10.5})$$

The voltage rise in the circuit is given by $\mathbf{IZ}$ and so:

$$\mathbf{V}_1 = \mathbf{V}_0 + \mathbf{IZ} = \mathbf{V}_0 + (R + jX)(P + jQ)/\mathbf{V}_1^* \quad (\text{A10.6})$$

It is common for the network voltage $\mathbf{V}_0$ to be defined and the generator busbar voltage $\mathbf{V}_1$ required. $\mathbf{V}_1$ can be obtained using the simple iterative expression:

$$\mathbf{V}_1^{(n+1)} = \mathbf{V}_0 + (R + jX)(P + jQ)/\mathbf{V}_1^{*(n)} \quad (\text{10.7})$$

where n is the iteration number.

This is a simple form of the conventional Gauss-Seidel load flow algorithm (Weedy and Cory, 1998). Once the calculation converges an accurate solution is obtained. More complex load flow calculations may be carried out using commercially available power system analysis programs. These include models of transformers, with their tap changers, and can solve large interconnected circuits in only a few iterations using more advanced algorithms.

Example: Calculation of voltage rise in a radial circuit

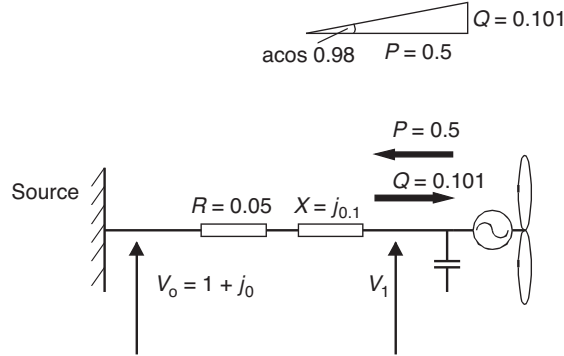


Figure A10.2 Example of calculation of voltage rise on a radial circuit (all values in per unit)

Consider a 5 MW wind farm operating at a power factor of 0.98 leading. The network voltage (V_0) is $(1 + j0)$ per unit and the circuit impedance (Z) is $(0.05 + j0.1)$ per unit on a 10 MVA base (Figure A10.2).

A power factor of 0.98 leading implies a reactive power draw of 1.01 MVar.

Thus, following Equation A10.7 the calculation becomes

$$V_1^{(n+1)} = 1 + (0.05 + j0.1)(0.5 + j0.101)/V_1^{*(n)}$$

For the first iteration ($n = 0$)

Assume

$$V_1^{*(0)} = 1 + j0;$$

then $V_1^{(1)}$ may be calculated to be:

$$V_1^{(1)} = 1.0149 + j0.0551$$

For the second iteration ($n = 1$)

$$V_1^{*(1)} = 1.0149 - j0.0551 \text{ then}$$

$$V_1^{(2)} = 1.0117 + j0.0549$$

For the third iteration ($n = 2$)

$$V_1^{*(2)} = 1.0117 - j0.0549 \text{ then}$$

$$V_1^{(3)} = 1.0117 + j0.0551$$

and the procedure has converged.

Therefore $V_1 = 1.013$ per unit at an angle of 3° , that is, the voltage at the generator terminals is 1.3% above that at the source. The angle between the two voltage vectors is small (3°).

The approximate calculation ($V_1 = V_0 + \text{PR-XQ}$) indicates a voltage V_1 of 1.015 per unit (i.e. a rise of 1.5%). This is confirmed by the results of the first iteration.

The current (I) in the circuit may be calculated from:

$$\begin{aligned} I &= S_1^*/V_1^* = (0.5 + j0.101)/(1.0117 - j0.0551) \\ &= 0.4873 + j0.1264 \text{ per unit} \\ |I| &= 0.503 \text{ per unit} \end{aligned}$$

With a connection voltage of 33 kV the base current is given by

$$I_{base} = V_{Abase}/\sqrt{3} \times V_{base} = 5 \times 10^6/1.732 \times 33 \times 10^3 = 87.5 \text{ Amps}$$

and so the magnitude of the current flowing in the 33 kV circuit is 44 Amps.

The real power loss in the circuit (W) is³:

$$W = I^2 R = 0.0127 \text{ per unit or } 127 \text{ kW}$$

The symmetrical short-circuit level at the generator busbar, before connection of the generators is simply:

$$\begin{aligned} S'' &= 1/|Z| = 1/(0.05^2 + 0.1^2)^{1/2} \\ S'' &= 8.94 \text{ pu or } 89.4 \text{ MVA} \end{aligned}$$

Once the induction generators of a wind farm are connected then they will make a contribution to the short-circuit current seen by a circuit breaker when it closes on to a fault. This is typically some five times the rating of a directly connected fixed speed induction generator, for example some 25 MVA in this case, giving a total short-circuit level of 115 MVA. However, for a three-phase symmetrical fault the fault current contribution from an induction machine decays rapidly and will make only a small contribution to the fault current which must be interrupted by an opening circuit breaker. Detailed guidance on the calculation of fault currents, including the contribution from induction generators is given in IEC 60909.

³ Note when the calculations are carried out using the per-unit system this is the aggregated power loss of all three phases.

11

Offshore wind turbines and wind farms

11.1 Development of offshore wind energy

Offshore wind energy is developing rapidly (Figure 11.1) with a total installed capacity in 2010 of approximately 3000 MW, some 1.5% of worldwide wind farm capacity. In a number of Northern European countries it is increasingly difficult to obtain permission to construct large wind farms onshore and this has stimulated interest in moving offshore, into the North Sea and Baltic. To meet its greenhouse gas emission targets, the UK has ambitious plans for some 15–20,000 MW of offshore wind farm capacity by 2020 with some projects more than 100 km offshore in average water depths of 50 m. The advantages of locating wind farms offshore include:

- Large areas are available for wind farm development with limited environmental impact.
- Mean wind speeds are high and this gives high capacity factors.
- Wind turbulence is low.
- Landfalls of the cables and points of connection to the power network are close to cities and load centres.

Figure 11.1 shows the development of world-wide capacity of offshore wind farms. Until around 2000 the projects were small and located in rather shallow water less than 5 km from shore. Then larger wind farms began to be constructed, in particular the two large Danish wind farms at Horns Rev (160 MW, 2002) and Nysted (166 MW, 2003). Presently a number of wind farms of up to 500 MW are being constructed in Northern Europe, particularly off the UK with smaller installations in the Far East, particularly off the coast of China (Table A11.1).

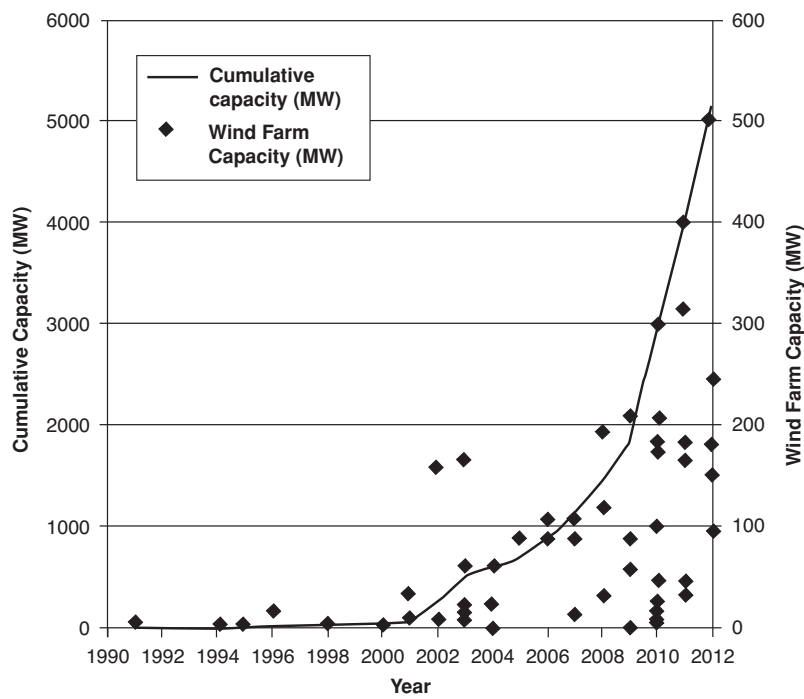


Figure 11.1 Offshore wind farm capacity worldwide (1991–2012)

Offshore installations represent a small fraction of the total wind power market and so the turbines are generally marinised developments of the machines used onshore (rated at up to 3.6 MW) but with some larger turbines (up to 5 MW) used in deep water sites. There is also one installation of two floating wind turbines. In depths of water of up to 20 m, either monopile or gravity foundations have been used while jacket or tripod structures are used for deeper waters. For smaller wind farms close to shore, the wind farm power collection cables at 30–36 kV are extended to shore. Larger wind farms use an offshore transformer station while High Voltage Direct Current transmission will be used for very large remote projects. Access for maintenance is generally by boat although helicopters have been used in some projects.

Table 11.1 shows an indicative breakdown of the capital costs of an offshore wind farm project. The turbines make up a rather lower fraction of the costs than for an onshore project with the support structures, cables and electrical connection a greater proportion. Installation of equipment offshore is, of course, a significant cost.

Table 11.1 Typical breakdown of costs for an offshore wind farm (after EWEA, 2009)

Element of offshore wind farm	% of total cost
Wind turbines	51
Support structures	19
Cables and network connection (including installation)	15
Installation of turbines and support structures	9
Other capital costs	6

The disadvantage of installing wind turbines offshore is the higher capital and operating costs and these have shown a considerable increase over the short history of offshore wind power development. This increase in costs has been studied by Gross *et al.*, 2010. From 2002 to 2004, capital costs for UK offshore wind farms were relatively stable at less than £1.5 m/MW. However, since then capital costs have doubled to £3 m/MW with the levelised costs of electrical energy increasing from £85/MWh (2004) to £150/MWh (2010). This compares with a capital cost for onshore wind energy of between £1.3 m/MW – £1.8 m/MW and a levelised cost of energy of £94/MWh (DECC, 2010). The comparable levelised cost of electrical energy in the UK from Combined Cycle Gas Turbine generators was £80/MWh (2010).

Thus, although onshore wind energy can be considered the most cost-effective low carbon technology for electricity generation, offshore wind energy is expensive requiring considerable additional support.¹

Gross *et al.*, 2010 considered that the factors that drove the escalation in cost of UK offshore wind farms from the mid-2000s to 2009 have been (in descending order of impact):

1. Materials, commodities and labour costs.
2. Currency movements.
3. Increasing prices for turbines over and above the cost of materials, due to supply chain constraints, market conditions and engineering issues.
4. The increasing depth and distance of more ambitious projects, affecting installation, foundation and operation and maintenance (O & M) costs.
5. Supply chain constraints, notably in vessels and ports.
6. Planning and consenting delays.

There has recently been some evidence of a small drop in capital costs of UK wind farms from £3 m/MW (2009) to £2.9 m/MW (2010) with a best guess of a cost of energy of £115/MWh by the mid-2020s. Some projects in the Baltic in rather shallow, sheltered waters close to shore have reported lower costs. Consideration of costs and other commercial matters is difficult and any analysis dates very quickly. However, it would appear that the early experience of UK offshore wind farms in rather shallow waters often close to shore gave a rather optimistic view of costs and that the later larger wind farms in more demanding exposed, offshore environments will lead to a higher cost of electrical energy unless considerable innovation is employed to reduce both capital and operating costs.

Offshore wind energy is in its infancy and experience is being gained as wind farms are constructed further from the shore and in more demanding environments. A number of books and reports on offshore wind energy are now emerging. Offshore climatology and meteorology are discussed by Barthelmie *et al.* and access by Dalen and Jakobsson in the comprehensive book on offshore wind power by Twidell and Gaudiosi (2009). Ackerman (2005) addresses offshore electrical systems while Hau (2006) provides an overview of the subject. After a review of offshore wind meteorology, this chapter considers the design loads of offshore wind turbines, which differ significantly from those onshore. The design of support structures is

¹Electricity generated from UK offshore wind farms attracts two Renewable Obligation Certificates or double the support of electricity generated from onshore wind farms.

described including fatigue calculations for monopile foundations while particular aspects of offshore power collection systems are also addressed.

11.2 The offshore wind resource

11.2.1 The structure of winds offshore

For sites far from shore, and for near-shore sites when the wind is blowing from open sea to land, mean wind speeds are usually higher than on land due to the absence of obstructions to the flow. Similarly, turbulence levels are usually much lower than on land, with low wind shear corresponding to the small surface roughness. However, the surface roughness increases with wave height, so turbulence and wind shear may be much higher during periods of large waves. These are correlated with periods of strong winds, since the waves are generally wind-driven, but there may be a significant time lag, so a wind farm may experience low winds in conjunction with high waves following a storm out at sea. Also the lower frequency swell waves may travel long distances, and may have been caused by high winds elsewhere which may never reach the wind farm. The contribution of the wind to the design loads of offshore turbines is considered in Section 11.3.2.

11.2.2 Site wind speed assessment

Quantification of the long-term wind resource at a potential wind farm site plays a central role in determining the economic feasibility of a project. For onshore wind farms this is typically achieved through the collection of wind data from an onsite meteorological mast and/or analysis of energy production data from neighbouring operational projects (Section 9.1.3, Chapter 9) These datasets are typically analysed in conjunction with historical records from a local meteorological station, regional wind index or modelled hindcast source as well as site wind flow models in order to predict the long-term wind climate at the site.

A similar approach may be adopted for offshore wind projects although there are significantly higher costs associated with the erection of meteorological masts at sea. Alternatively, wind resource estimates may be based on secondary sources such as measured wind data obtained from neighbouring sites or publicly funded measurement stations in the region of interest. In such circumstances, the wind analyst must assess the difference in wind climate between the reference mast and the target site, as well as the uncertainty associated with this assessment.

A commonly held assumption has been that the horizontal spatial variation of long-term mean wind speed in the offshore environment is relatively limited (Barthelmie 1993; Coelingh *et al.*, 1996). Therefore, the uncertainty associated with extrapolation of measurements from one physical location to another has often been perceived as relatively limited. However, in the absence of high quality public domain data from several reliable measurement nodes, there is little evidence for this assumption and so wind climate prediction is subject to a significant degree of uncertainty in the absence of a high quality measurement campaign on the site of interest. This can have a significant impact on the outlook for the financing of some projects.

Various methodologies may be applied in order to assess the variation of wind resource across wide areas of the offshore environment. These include conventional computational flow modelling techniques such as WAsP (Mortensen *et al.*, 1993) as well as more sophisticated

modelling including Computational Fluid Dynamics (CFD) and Mesoscale methods. The latter is an increasingly applied technique which is based on Numerical Weather Prediction models developed by the meteorological research community. A Global Climate Model is used to find the conditions of the upper atmosphere and these results are then either used as boundary conditions of Regional Climate Models (at a scale typically of $50 \text{ km} \times 50 \text{ km}$) or with empirically derived relationships with the sea surface parameters, temperature and wind speed (Barthemie *et al.* in Twidell and Gaudiosi, 2009). The benefit of this approach is the potential for capturing the wide scale synoptic trends as well as more localised thermal and topographic effects through application of a theoretically robust meteorological model, initiated from a large database of historical observations.

Analysis of images recorded by satellite-mounted instrumentation is also becoming a more frequently applied methodology for considering the variation in offshore wind resource. One of the most commonly utilised instruments is a Scatterometer, which is an active microwave device used to derive estimates of wind speed and direction based on returns from a fan beam of short pulses. These returns are received at a variety of incidence angles which respond principally to the surface roughness of the sea. When the wavelength of the incident beam is in phase with the wavelength of the short-crested localised ‘wind waves’ (taking account of the incident beam angle), resonant signals become present in the backscatter return (Aage *et al.*, 1998). The intensity of the resonant backscatter is proportional to the steepness of the wind waves and hence also to the localised wind speed at the time of observation. The relationship between the magnitude of resonant backscatter and wind speed has been derived empirically and also takes into consideration the incidence angle and the azimuth angle of the satellite observation direction with respect to the wind direction (Aage *et al.*, 1998). Whilst the absolute accuracy of the derived wind speed values is limited by the validity of the empirically-derived relationship applied and the availability of the data is limited due to infrequent spatial and temporal sampling, analytical methods have been developed that allow a robust estimate of the spatial variation of wind resource over domains of several hundred kilometres to be determined (Phillips *et al.*, 2008).

In summary, a wide variety of measurement and computer model-based applications are available to determine the wind resource at any given offshore wind site. The uncertainties associated with the source data and analytical methodology adopted play a significant role in determining the level of confidence that may be ascribed to projections of long-term energy production, which in turn has a substantial influence on the economic viability of any given offshore wind farm development.

11.2.3 Wakes and array losses in offshore wind farms

Wake losses in offshore wind farms can lead to a considerable loss of energy particularly at wind speeds below rated. Wake losses are most strongly dependent on wind speed through the turbine thrust coefficient, being high at low wind speeds and then decreasing as wind speeds increase. Above rated wind speed, pitch regulated turbines limit their power production by feathering the blades, which reduces the thrust coefficient and hence the wake losses at downstream turbines, so above a certain wind speed all turbines may be producing rated power despite the wakes. Stall regulated turbines continue to have a high thrust coefficient, which may cause high wake losses at all wind speeds.

The thrust coefficient implies an extraction of momentum from the air flow, but further downstream the momentum is replenished by mixing from the higher-speed flow above the

wind farm. Atmospheric boundary layer models can provide ways of calculating the rate of momentum replenishment, from which the efficiency of the wind farm can be calculated (Bossanyi, 1980). However, a more detailed assessment should consider the development of the individual turbine wakes and their interaction with downstream turbines. This means that the wake losses depend significantly on wind direction as well as turbine spacing. With a regular array of turbines, the wake losses will be high in certain narrow wind direction sectors, and the overall average losses will depend on the joint distribution of wind speed and direction.

Array wake losses can be exacerbated by small separation distances of the turbines used in some projects and may be related to the low levels of ambient turbulence of the offshore wind (Barthelmie *et al.*, 2005). For example, the Lillgrund offshore wind farm (Figure 11.2) was built with 48, 2.3 MW, 93.5 m diameter turbines (Dahlberg and Thor, 2009). It was constructed with rather small separation distances between turbines (3.3 D along each row and 4.4 D between rows). The small separation distances arose as the rotor diameter of the available offshore turbine was increased by the manufacturer during the wind farm permitting process when the base positions had been fixed. An extensive campaign of measurements showed that for wind blowing directly down a row, the second turbine experienced a power deficit wake loss of 80% (i.e. it produced only 20% of the power of the first turbine over

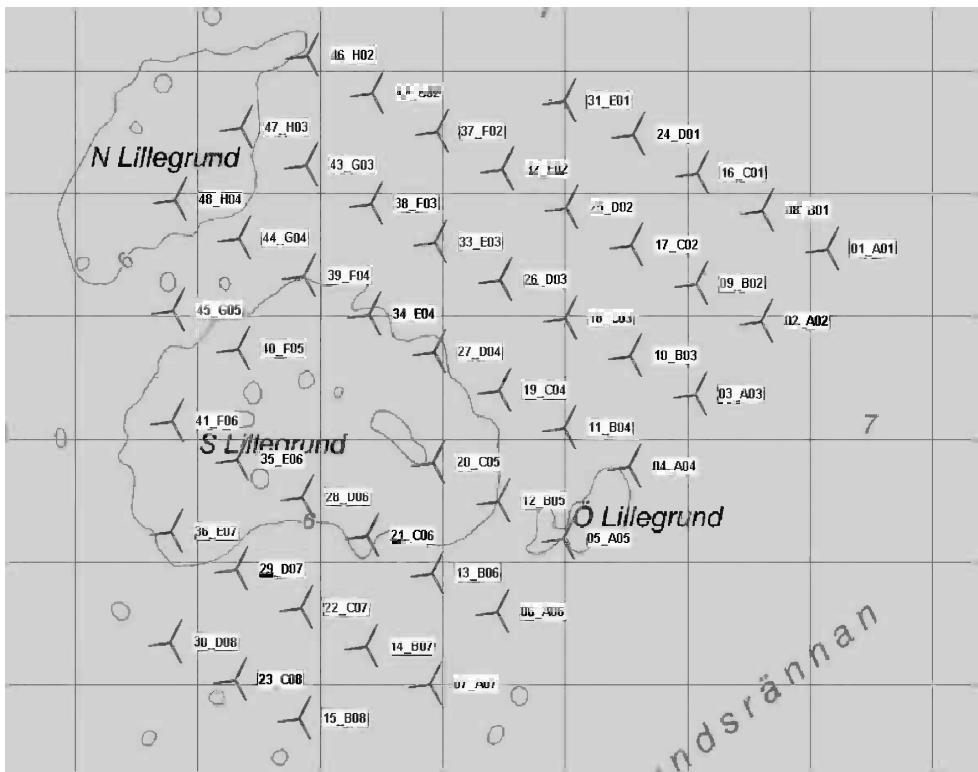


Figure 11.2 Lillgrund wind farm layout. Dahlberg and Thor, 2009. Reproduced by permission of Vattenfall

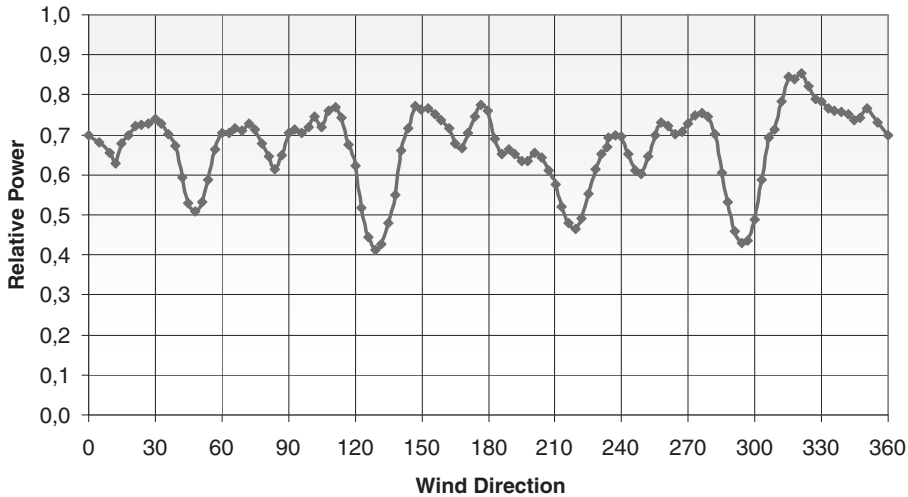


Figure 11.3 Lillgrund wind farm: variation in array efficiency with wind direction for wind speeds below rated. Dahlberg and Thor, 2009. Reproduced by permission of Vattenfall

a range of wind speeds below rated). Similarly at a spacing of 4.4 D the power deficit was around 70%.

Figure 11.3 shows how the wind farm array efficiency varies with wind direction for wind speeds below rated. The array efficiency (relative power) is defined as the average array power generation per turbine at a given wind speed divided by the corresponding output of a turbine not suffering from wake losses – that is, one at the windward edge of the array. Pronounced dips when the wind is blowing along the rows are clearly seen. On the assumption of an isotropic wind direction probability distribution, the overall array efficiency of the wind farm was estimated to be 67% for wind speeds below rated and 77% for a site wind speed distribution assumed to be Rayleigh with an annual mean of 8 m/s. This implies that the power output of this rather dense offshore wind farm was reduced by approximately 23% compared to what might be expected from a single isolated turbine (Lillgrund Pilot Project, 2008). Wake losses at Middelgrunden (curved single line of turbines, spacing 2.4 D) were reported as around 10% in (Barthelmie *et al.*, 2007).

The Lillgrund wind farm was exceptional in its small spacing between turbines. Horns Rev has spacing of 7 D along its rows and 10 D between rows. Its output has been analysed (Hansen K *et al.*, 2010) to show maximum power deficit wake loss of the second wind turbine in the array of around 40% at wind speeds of 5–9 m/s dropping to 30% at 11–13 m/s for a narrow range of directions. The flow deficit sector generated by each turbine was approximately 25–30°. Overall array losses were reported as 12.4% (Sørensen *et al.*, 2006).

Atmospheric turbulence and stability can have a significant effect on wake losses (Barthelmie and Jensen, 2010). Turbulent mixing causes the velocity deficit in the wake to broaden and weaken as it convects downwind, and with the low levels of turbulence typical of some offshore sites the individual wakes will persist further downstream, but their effects will be felt over a narrower range of wind directions.

A number of approaches have been used to predict offshore array efficiency. Barthelmie *et al.* (2009) compare six different models. The simplest model uses a simple ‘top-hat’ wake profile, which is assumed to expand linearly based on a wake decay coefficient. Somewhat more sophisticated is the use of wake models derived from the Reynolds-Averaged Navier-Stokes (RANS) equations together with empirical relationships for the wake-generated turbulence and the superposition of multiple wakes; while two of the models use full Computational Fluid Dynamics (CFD) modelling of the flow through the wind farm, treating the rotor disks as momentum sinks. The Barthelmie *et al.* paper notes that although most of the accepted wake models appear to give reasonable results for single wakes or small wind farms, wake losses are under-predicted in large offshore arrays. This is thought to be due to inadequate representation of turbulence induced from the upstream turbines and the effect of atmospheric stability. On land such effects tend to be masked by the higher levels of turbulence caused by the topography. The performance of all models can be improved by modified parameterizations (Barthelmie *et al.*, 2010; Cleve *et al.*, 2009; Schlez and Neubert, 2009) although research is continuing to better understand and predict the flow and wakes in offshore wind farms.

On a larger scale, as more offshore wind farms are built the effect of the ‘wake’ of an entire wind farm on the energy production of another wind farm starts to be of interest, as well as the effect this may have on the total wind energy resource which can be extracted from a given sea area. This is proving difficult to assess, partly due to the lack of measurements. Results summarized in (Frandsen *et al.*, 2009) suggest that a variety of modelling approaches may be appropriate but still need further verification if they are to be used successfully to estimate power output prior to wind farm construction.

11.3 Design loads

11.3.1 International Standards

National design standards specifying wind and wave loadings on offshore structures were developed as North Sea oil began to be exploited, and were subsequently utilised for the design of the first offshore wind farms. However, offshore wind turbines differ from offshore oil platforms in that the wind loading on wind turbine support structures is a much more significant element of the total loading than is the case for oil platforms, so the desirability of design standards tailored to offshore wind turbines was soon recognised.

The first design standard for offshore wind turbine structures to appear was published by Det Norske Veritas in 2004. Offshore Standard DNV-OS-J101, ‘Design of Offshore Wind Turbine Structures’, is a comprehensive document, covering marine conditions, design load cases, wave loading and structural design. The design load cases were the same as those set out in IEC 61400–1, but with appropriate wave heights or sea states specified. However, in 2007, a revised edition was published with the load cases modified to be consistent with those in the draft IEC 61400–3 (see below).

Germanischer Lloyd published *Guidelines for the Certification of Offshore Wind Turbines* in 2005. It covers the rotor and nacelle components as well as the support structure and is structured in the same way as the parallel document for onshore machines, but with individual chapters expanded to include the requirements particular to offshore siting.

The International Electrotechnical Commission standard IEC 61400–3 ‘Wind Turbines – Part 3: Design requirements for offshore wind turbines’ appeared in 2009. Unlike the GL guidelines referred to above, this is not a stand-alone document, but has to be read in conjunction with IEC 61400–1 (see Chapter 5).

The load cases specified in IEC 61400–3 closely correspond to those in IEC 61400–1, with the marine conditions to be taken into account added in each case. Deterministic waves may be considered in conjunction with deterministic gusts in a few load cases, but in most the standard specifies stochastic sea states and turbulent wind, requiring several ten-minute simulations to be run. A minimum of six such simulations is required for each mean hub height wind speed and sea state for each load case, but for some load cases significantly greater numbers are required. The wind and marine conditions specified are considered in the next sections.

Some of the work in developing the IEC 61400–3 guidelines was carried out under the auspices of a European Community supported R & D project ‘Recommendation for Design of Offshore Wind Turbines’ (contract no ENK5-CT-2000–00322), or RECOFF, and was reported in some 70 papers.

11.3.2 Wind conditions

IEC 61400–3 requires the support structure to be designed for site-specific wind conditions, but, in the case of the rotor-nacelle assembly, the alternative approach of designing for one of the wind turbine classes defined in IEC 61400–1 may also be followed.

The following hub height wind parameters are to be determined at the site, preferably by site monitoring:

- the extreme ten-minute mean wind speed with a recurrence period of 50 years, U_{ref} ;
- wind speed and direction distributions;
- turbulence intensity for wind speeds over the operating range, and at U_{ref} ;
- standard deviation of turbulence intensity at each wind speed;
- wind shear;
- air density.

In the absence of site monitoring, the above parameters may be estimated from local standards, or, in the case of turbulence intensity, from the surface roughness parameter, z_o . Sea roughness increases with wind speed, and z_o may be estimated from the Charnock expression:

$$z_o(\bar{U}) = \frac{A_c}{g} \left[\frac{\kappa \bar{U}}{\ln(z_{hub}/z_o(\bar{U}))} \right]^2 \quad (11.1)$$

where A_c is Charnock’s constant, g is the acceleration due to gravity and κ is von Karman’s constant (0.4). The standard recommends values of A_c ranging between 0.011 for open sea and 0.034 for near-coastal locations. The design value of turbulence intensity is taken as the 90% quantile given by the equation:

$$I_u = 1/\ln(z_{hub}/z_o(\bar{U})) + 1.28(1.44I_{15})/\bar{U} \quad (11.2)$$

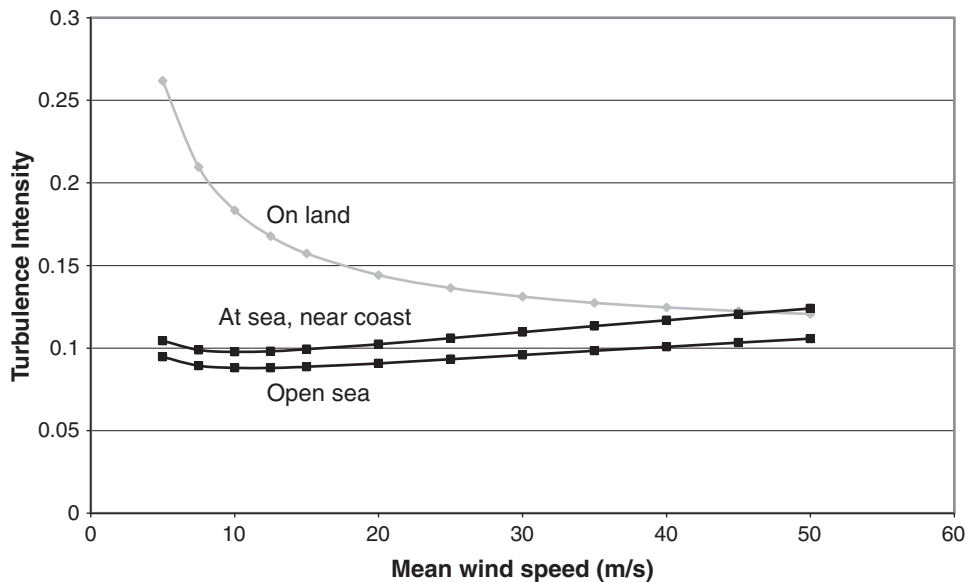


Figure 11.4 Variation of turbulence intensity with wind speed – onshore and offshore

The first term in this expression is an accepted formula for the *mean* turbulence intensity and the second term represents the additional 1.28 standard deviations of turbulence intensity required for the 90% quantile. Accordingly I_{15} can be estimated as $1/\ln(z_{hub}/z_o(15))$.

The resulting variations in turbulence intensity with wind speed at open sea and near-coastal locations are compared with that at an onshore site with medium turbulence (Category B) in Figure 11.4. It is seen that the turbulence intensity at an offshore coastal location is only slightly higher than over the open sea, but in both cases is significantly less than that onshore throughout the operational speed range of typically 5–25 m/s.

11.3.3 Marine conditions

The marine conditions relevant to support structure design encompass waves, currents, water level and, where appropriate, scour, seabed movement and sea ice. In the case of the IEC 61400–3 operational load cases which have their counterparts in IEC 61400–1 (i.e. all but load case 1.6), normal marine conditions are specified, with severe marine conditions applicable in load case 1.6 and extreme conditions applicable in a number of non-operational load cases.

The wave climate is most accurately described in terms of a stochastic sea state model but, in view of the non-linearity of wave loading models in storm conditions, it is necessary to introduce deterministic design waves as well. The various sea states defined in IEC 61400–3, and the alternative deterministic design waves, are described in succeeding sections, together with details of the co-existent currents and water levels to be considered. Before moving to these, however, the wave spectra commonly employed in stochastic sea state models will be introduced.

11.3.4 Wave spectra

The surface profile of the sea can be considered to be the result of the superposition of a very large number of tiny advancing regular waves of different amplitudes and periods upon one another. This ensemble of waves can, in turn, be described by a frequency spectrum – that is a plot of the spectral density of the square of the deviation of the water surface elevation from mean level against frequency. This spectrum is said to define the sea state, which is usually considered to persist for a period of three hours.

The key parameters of the frequency spectrum are the significant wave height, H_s , and the frequency corresponding to the peak spectral density, f_p . The significant wave height is traditionally defined as the average height of the highest one third of waves in a chosen reference period – normally three hours – and corresponds to the height of waves that observers will typically report. The area under the spectrum is equal to the variance of the sea surface elevation, σ_η^2 .

The shape of the frequency spectrum varies according to the fetch and the way wind speeds have varied over the fetch area beforehand. The Pierson-Moskowitz spectrum is a mathematical construct which is considered to provide a good fit to the frequency spectrum of a fully developed sea. In IEC 61400-3, it is specified as:

$$S_{\eta\eta}(f) = \frac{5}{16} \frac{H_s^2}{f} \left(\frac{f_p}{f} \right)^4 \exp \left(-\frac{5}{4} \left(\frac{f_p}{f} \right)^4 \right) \quad (11.3)$$

This formulation is based on the assumption that the significant wave height, H_s , is four times the standard deviation of the sea surface elevation, which is correct for a Rayleigh distribution of wave heights. However, although a Rayleigh distribution of wave heights is strictly generated only by a narrow banded frequency spectrum, it is considered to be a reasonable approximation in the case of real wave spectra.

Fully developed sea states only arise if a wind speed is maintained for long enough over a sufficient fetch. In the majority of situations, either the storm duration or the fetch or both are too small, with the result that the frequency spectra display a more exaggerated peak than the Pierson-Moskowitz (PM) spectrum. The JONSWAP (Joint North Sea Wave Project) spectrum, which was developed from wave measurements near the German island of Sylt, is a variant of the PM spectrum which models this peak enhancement. Essentially the PM spectrum is multiplied by a frequency dependent peak enhancement factor (with a maximum between 1 and 5) and by a normalising factor, which ensures that the area under the JONSWAP spectrum is the same as that under the PM spectrum.

The peak enhancement factor, γ^α , is given in terms of the significant wave height, H_s , measured in metres, the peak spectral period, $T_p = 1/f_p$ seconds, and the frequency, f . $\gamma = \exp\{5.75 - 1.15T_p/H_s^{0.5}\}$ for $T_p/H_s^{0.5}$ in the range 3.6 to 5, $\alpha = \exp\{-0.5(f/f_p - 1)^2/\sigma^2\}$, $\sigma = 0.07$ for $f < f_p$ and $\sigma = 0.09$ for $f > f_p$, and the normalising factor is $1 - 0.287 \ln \gamma$. $\gamma = 5$ when $T_p/H_s^{0.5}$ is less than 3.6 and $\gamma = 1$ when $T_p/H_s^{0.5}$ exceeds 5. It is seen that the JONSWAP spectrum is identical to the PM spectrum when $T_p/H_s^{0.5}$ exceeds 5.

The Jonswap and Pierson-Moskowitz spectra are compared for a significant wave height of 3 m and peak period of 7 seconds in Figure 11.5.

Figure 11.6 shows an example time history of water surface elevation from a simulation utilising the above Jonswap spectrum.

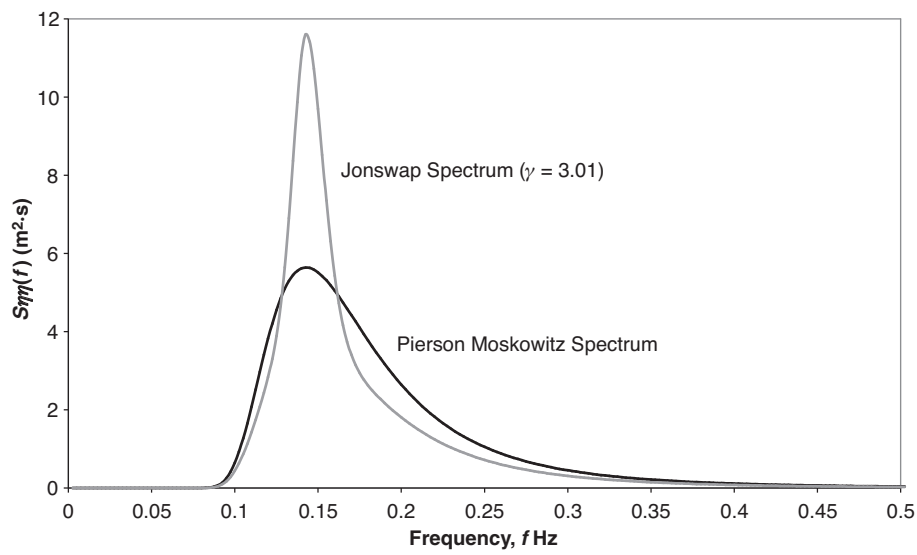


Figure 11.5 Peirson-Moskowitz and Jonswap spectra for $H_s = 3$ m and $T_p = 7$ sec

11.3.5 Ultimate loads: operational load cases and accompanying wave climates

An overview of the IEC 61400–3 ultimate load cases for operational loading is given in Table 11.2, which identifies the equivalent onshore load case in IEC 61400–1 in each case.

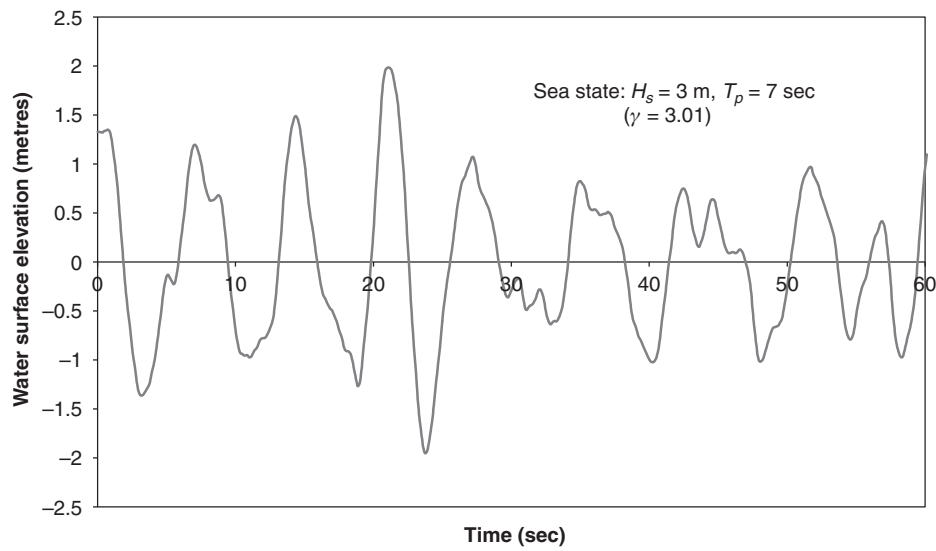


Figure 11.6 Simulated water surface elevation time history based on Jonswap spectrum

Table 11.2 Ultimate load cases during operation

M/c State	Load case no.	Equiv LC on land	Wind conditions	Wind speed range	Wave conditions	Water Level	Grid status; fault type	Load case type
Power Production	1.1	1.1	NTM	Cut-in – Cut-out	Normal sea state i.e. $H_s = E[H_s \bar{U}]$ Extrapolations required	Mean	None	N
	1.3	1.3	ETM	Cut-in – Cut-out	Normal sea state	Mean	None	N
	1.4	1.4	ECD	Rated ± 2 m/s	Normal sea state/wave	Mean	None	N
	1.5	1.5	EWS	Cut-in – Cut-out	Normal sea state/wave	Mean	None	N
	1.6a	None	NTM	Cut-in – Cut-out	Severe sea state – 50 yr return H_s conditional on chosen wind speed	Normal range	None	N
	1.6b	None	NTM	Cut-in – Cut-out	Severe wave height – 50 yr return wave conditional on chosen wind speed	Normal range	None	N
Power production with occurrence of fault	2.1	2.1	NTM	Cut-in – Cut-out	Normal sea state	Mean	Control system or grid loss	N
	2.2	2.2	NTM	Cut-in – Cut-out	Normal sea state	Mean	Protection system or elec. fault	A
	2.3	2.3	EOG	Rated ± 2 m/s	Normal sea state/wave	Mean	Ext or intl. electrical fault	A
Start-up	3.2	3.2	EOG	Cut-in, rated, cut-out	Normal sea state/wave	Mean	None	N
	3.3	3.3	EDC	Cut-in, rated, cut-out	Normal sea state/wave	Mean	None	N
Shut down	4.2	4.2	EOG	Rated, cut-out	Normal sea state/wave	Mean	None	N
Emergency Shut-dn	5.1	5.1	NTM	Rated, cut-out	Normal sea state	Mean	None	N

Normal and abnormal load cases are indicated by the letters N and A respectively. See Table 5.2, Chapter 5, for the applicable load factors.

The wind conditions denoted by the acronyms in the table above are as follows:

NTM	Normal turbulence model
ETM	Extreme turbulence model
ECD	Extreme coherent gust with direction change
EWS	Extreme wind shear
EOG	Extreme operating gust
EDC	Extreme direction change

A description of each wind condition is given in Chapter 5, Section 5.4.2.

The wave conditions are specified as ‘normal sea state’ or ‘normal wave height’ in all load cases, apart from load cases 1.6a and 1.6b, for which the ‘severe sea state’ and the ‘severe wave height’ respectively apply. These four wave conditions are described in turn below.

Normal sea state

IEC 61400–3 defines normal sea states in terms of a stochastic sea state model based on either the Pierson-Moskowitz or Jonswap spectrum as appropriate. For each normal sea state, the significant wave height is to be taken as the expected value at the mean wind speed under consideration, based on analysis of the site-specific metocean database. It is normal to plot the metocean data as a wind-wave scatter diagram, which can be used to derive the expected significant wave height at each wind speed. Typically a two-dimensional wind-wave scatter diagram is presented in the form of a table recording the percentage of time that the mean wind speed and significant wave height lie within different bins in a two-dimensional array. In a three-dimensional scatter diagram, the frequency of occurrence of different peak spectral periods, T_p , or mean zero-crossing period, T_z is recorded also. The mean zero-crossing period can be converted to the peak spectral period by means of the relation

$$T_p = T_z \sqrt{\frac{11 + \gamma}{5 + \gamma}}$$

with gamma as defined in the preceding section.

Data from a two-dimensional scatter diagram for a site designated NL-1 about 15 km off Ijmuiden on the Dutch coast is illustrated in Figure 11.7. This shows the frequency of occurrence of different mean wind speeds for each significant wave height at a 0.5 m interval, with reference to a different horizontal axis in each case.

Account should be taken of the range of peak spectral periods, T_p , encountered and care taken to investigate values of T_p close to the support structure natural frequency where appropriate.

Marine conditions are essentially the same for all ultimate operational load cases except load case 1.6, and consist of the normal sea state in conjunction with ‘normal’ currents and water at mean sea level. ‘Normal’ currents comprise currents induced by the wind, estimated at 1% of the hourly mean wind speed, and breaking wave induced surf currents (if any). The

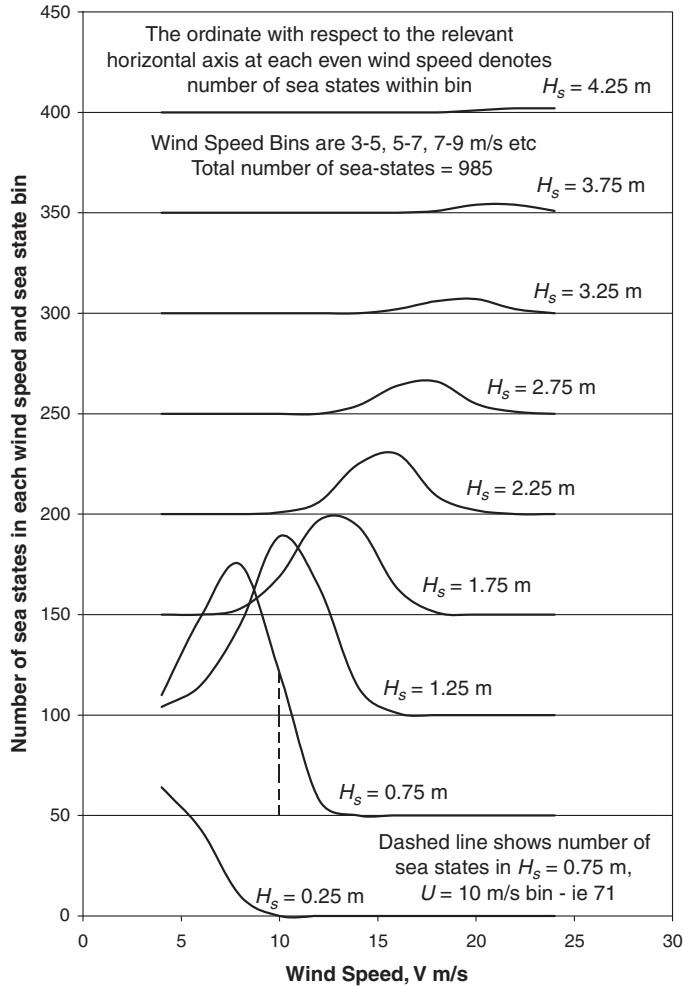


Figure 11.7 Data from wind-wave scatter diagram for site NL-1

latter may be estimated from the formula $U_{bw} = 2s\sqrt{gH_B}$ where s is the sea-floor slope and H_B the breaking wave height.

Normal deterministic design wave

The height of the normal deterministic design wave is taken as equal to the expected significant wave height at the mean wind speed under consideration.

Severe sea state

The IEC 61400-3 severe sea states, which, like the normal sea states, occur during machine operation (load case 1.6a), differ from the normal sea states only in regard to the significant

wave height. For each mean wind speed, $\bar{U}$, the significant wave height, $H_{s,SSS}(\bar{U})$ is to be taken at a value such that its recurrence period in combination with the mean wind speed is 50 years, based on extrapolation of the site-specific metocean database. The subsection entitled 'Environmental contours' at the end of this section outlines how this extrapolation may be carried out. As with the normal sea state, the most adverse anticipated value of the peak spectral period should be considered.

The other marine conditions for load case 1.6, consist of 'normal' currents and water at any level within the 'normal water level range', which may be taken as the range of astro-nomic tides.

Severe deterministic design wave

IEC 61400-3 load case 1.6b specifies deterministic design waves derived from the significant wave height for the severe sea state (above) at each mean wind speed. The height of the deterministic design wave, or severe wave height, $H_{SWH}(\bar{U})$, is the expected height of the extreme wave in the applicable severe sea state. If the wave heights are Rayleigh distributed, then the probability of an individual wave height exceeding H is

$$P(h > H) = \exp \left(-0.5 \left(\frac{H/2}{\sigma_\eta} \right)^2 \right) = \exp \left(-2 \left(\frac{H}{H_s} \right)^2 \right) \quad (11.4)$$

and the wave height likely to be exceeded by the highest of N waves is $H_s \sqrt{0.5 \ln N}$. The expected magnitude of the extreme wave is slightly higher than this at $H_s(\sqrt{0.5 \ln N} + 0.2886/\sqrt{2 \ln N})$.

For a sea state with mean wave period of 10.8 seconds, there will be 1000 waves in the three-hour reference period, so the extreme wave is likely to exceed $1.86H_s$ and its expected magnitude is $1.94H_s$. However, Annex G of IEC 61400-3 permits the height of the deterministic design wave for each severe sea state to be calculated as $H_{SWH}(\bar{U}) = 1.86H_{s,SSS}(\bar{U})$.

The wave period to be adopted is the value within the range $11.1\sqrt{H_{s,SSS}(\bar{U})/g}$ to $14.3\sqrt{H_{s,SSS}(\bar{U})/g}$ that results in the highest wave loading.

Load case 1.6b can be omitted if non-linear wave kinematics can be properly modelled in the analysis of load case 1.6a.

Environmental contours

One of the load cases representing power production, load case 1.6, is concerned with 50-year return combinations of mean wind speed, $\bar{U}$, and significant wave height, H_s , which, taken together, can be considered as forming an environmental contour in $\bar{U}$, H_s space. A method for deriving such a contour using the Inverse First Order Reliability Method (IFORM) is set out in Annex G of IEC 61400-3 and is described in this section.

The probability that a certain combination of mean wind speed and significant wave height will not be exceeded may be written $F(H_s | \bar{U}) \cdot F(\bar{U})$ where $F(\bar{U})$ is the cumulative

distribution function (CDF) of the mean wind speed and $F(H_s|\bar{U})$ is the CDF of the significant wave height conditional on the mean wind speed.

For the 50-year return combination, this probability must equate to $1 - 1/N$, where N is the number of sea states in 50 years – that is, $50(365)24/3 = 146,000$ for a three-hour sea state duration.

The environmental contour may be derived by utilising a transformation from mean wind speed and significant wave height to two uncorrelated normally distributed variables, U_1 and U_2 , so that

$$F(\bar{U}) = \Phi(U_1) \quad \text{and} \quad F(H_s|\bar{U}) = \Phi(U_2) \quad (11.5)$$

$$\text{where } \Phi(U) = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^U \exp[-u^2/2] du \quad (11.6)$$

The joint probability density function (pdf) of the normally distributed variables U_1 and U_2 is

$$p(U_1, U_2) = \frac{1}{\sqrt{2\pi}} \exp\left[-\frac{U_1^2}{2}\right] \cdot \frac{1}{\sqrt{2\pi}} \exp\left[-\frac{U_2^2}{2}\right] = \frac{1}{2\pi} \exp\left[-\frac{U_1^2 + U_2^2}{2}\right] \quad (11.7)$$

so the joint pdf is axi-symmetric about the origin and points in U_1, U_2 space lying at the same radius from the origin have the same probability density. The joint pdf is illustrated in Figure 11.8.

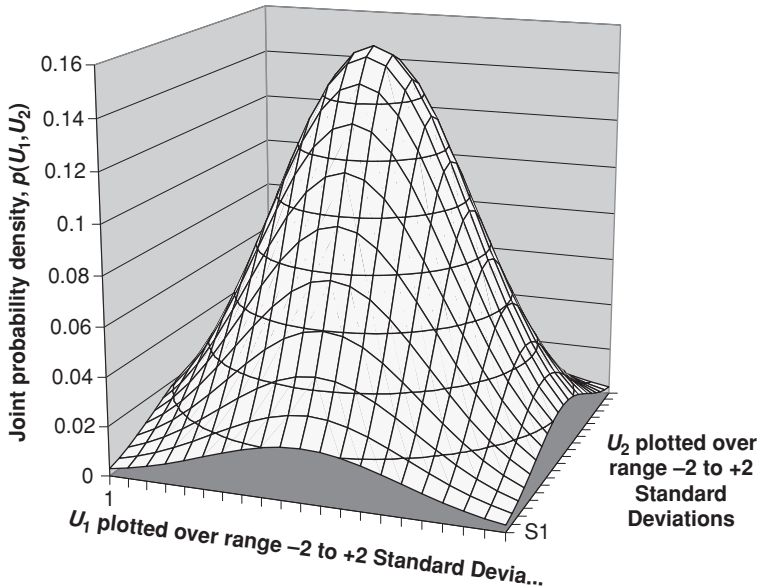


Figure 11.8 Joint probability density of two uncorrelated normally distributed variables

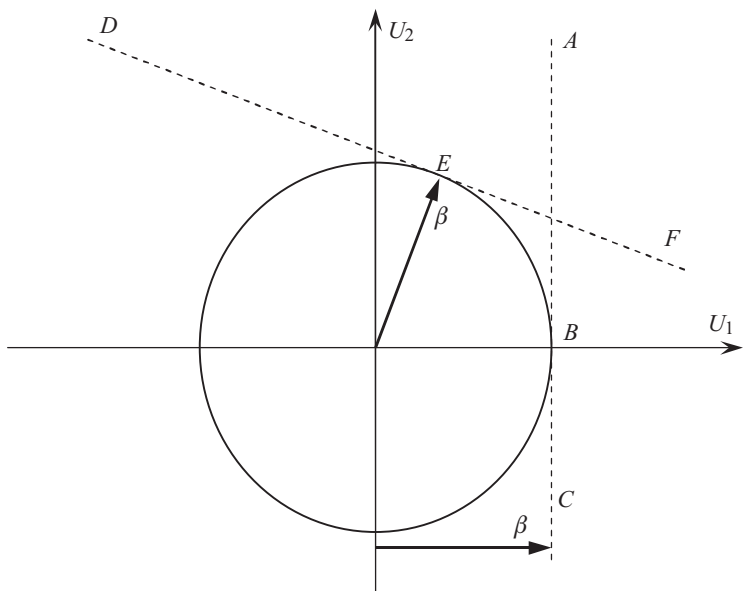


Figure 11.9 Circle of radius β in $U_1 \cdot U_2$ space, representing environmental contour

Clearly the N year return wind speed, irrespective of significant wave height, is that for which $F(\overline{U}) = \Phi(U_1) = 1 - 1/N$, that is, for which $U_1 = \Phi^{-1}(1 - 1/N)$. In U_1, U_2 space, the probability that the wind exceeds the N year return value is represented by the probability that the point U_1, U_2 lies to the right of the line ABC in Figure 11.9, with β set equal to $\Phi^{-1}(1 - 1/N)$.

Given that the volume under the joint probability density function surface (Figure 11.8) lying beyond any tangent to the circle radius β is the same, it follows that the probability that a point U_1, U_2 lies beyond the tangent DEF is the same as the probability that a point U_1, U_2 lies beyond the tangent ABC – that is, $1/N$. Thus, the circle radius β is the internal envelope of a family of tangents such that the probability of U_1, U_2 lying outside each one is $1/N$. In this sense, it represents the N -year environmental contour transformed into two dimensional normal space. Note that the portions of the circle for which U_1 or U_2 is negative represent N -year return *minimum* values in each case.

A procedure for the practical application of the method to metocean data in the form of a mean wind speed/significant wave height scatter diagram is as follows:

1. (a) Using the scatter diagram, calculate the non-exceedance probability, $F(\overline{U})$, associated with the mean wind speed, $\overline{U}$, of a particular wind speed bin.
- (b) Calculate the corresponding value of the normally distributed variable $U_1 = \Phi^{-1}(F(\overline{U}))$.
- (c) Calculate $U_2 = \sqrt{\beta^2 - U_1^2}$. (The negative root corresponds to the *minimum* value of H_s .)

(d) Calculate the 50 year return significant wave height conditional on $\bar{U}$ as

$$H_{s,SSS}(\bar{U}) = \bar{H}_s(\bar{U}) + U_2 \sigma_{H_s}(\bar{U})$$

where $\bar{H}_s(\bar{U})$ and $\sigma_{H_s}(\bar{U})$ are the mean and standard deviation of the significant wave height conditional on $\bar{U}$ extracted from the scatter diagram, respectively.

2. Repeat step 1. for the other wind speed bins.
3. Plot environmental contour linking the points $H_{s,SSS}(\bar{U})$, $\bar{U}$.

Note that in the above procedure, it is assumed that the significant wave heights conditional on $\bar{U}$ are *normally distributed*, which may not necessarily be consistent with the data. It is more conservative to assume that the significant wave heights are log-normally distributed. Whichever model is adopted, goodness of fit tests – visual and/or statistical – should be carried out to confirm its suitability.

An example of 50-year return environmental contours derived from the scatter diagram for the NL-1 site (see Figure 11.7) is given in Figure 11.10. One contour is based on a normal distribution of significant wave heights conditional on $\bar{U}$ and the other is based on a log-normal distribution.

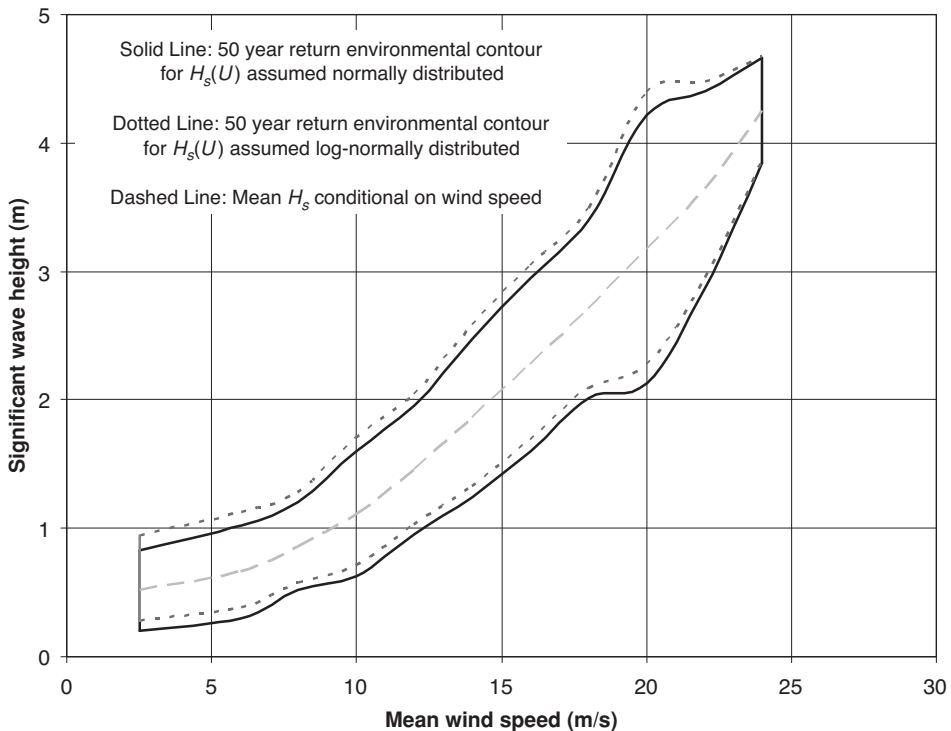


Figure 11.10 50-year significant wave height against mean wind speed environmental contours derived from site NL-1 scatter diagram

11.3.6 **Ultimate loads: non-operational load cases and accompanying wave climates**

An overview of the IEC 61400–3 ultimate load cases for non-operational loading is given in Table 11.3, which identifies the equivalent onshore load case in IEC 61400–1 in each case.

It is seen that for each of the four IEC 61400–1 load cases applicable onshore, IEC 61400–3 specifies two or three sub load cases for use offshore, which are distinguished by suffices *a*, *b* and *c*. These differ as follows:

- Suffix *a* load cases specify the extreme turbulent wind in conjunction with the extreme stochastic sea state.

Table 11.3 Ultimate non-operational load cases

M/c state	Load case no.	Equiv LC on land	Wind conditions	Wind speed	Wave conditions	Water Level	Grid, yaw and fault status	Load case type
Parked (standing still or idling)	6.1a	6.1	50-yr EWM turbulent	$k_1 U_{ref}$	50-yr extreme sea state $H_s = k_2 H_{s50}$	Extreme range	None	N
	6.1b	6.1	50-yr EWM steady	U_{e50}	50-yr reduced wave height $H = \sim 0.7 H_{50}$	Extreme range	None	N
	6.1c	None	50-yr RWM steady	$0.786 \times U_{e50}$	50-yr extreme wave height $H = H_{50}$	Extreme range	None	N
	6.2a	6.2	50-yr EWM turbulent	$k_1 U_{ref}$	50-yr extreme sea state $H_s = k_2 H_{s50}$	Extreme range	Grid loss, 180° yaw	A
	6.2b	6.2	50-yr EWM steady	U_{e50}	50 yr Reduced wave height $H = \sim 0.7 H_{50}$	Extreme range	Grid loss, 180° yaw	A
	6.3a	6.3	1-yr EWM turbulent	$0.8 \times k_1 U_{ref}$	1-yr extreme sea state $H_s = k_2 H_{s1}$	Normal range	Extreme yaw (20°)	N
	6.3b	6.3	1-yr EWM steady	U_{e1}	1-yr reduced wave height $H = \sim 0.7 H_1$	Normal range	Extreme yaw (30°)	N
Parked with fault condition	7.1a	7.1	1-yr EWM turbulent	$0.8 \times k_1 U_{ref}$	1-yr extreme sea state $H_s = k_2 H_{s1}$	Normal range	Fault condition	A
	7.1b	7.1	1-yr EWM steady	U_{e1}	1-yr reduced wave height $H = \sim 0.7 H_1$	Normal range	Fault condition	A
	7.1c	None	1-yr RWM steady	$0.786 \times U_{e1}$	1-yr extreme wave height $H = H_1$	Normal range	Fault condition	A

EWM and RWM stand for extreme wind speed model and reduced wind speed model respectively.

- Suffix *b* load cases specify the extreme steady wind in conjunction with the ‘reduced’ deterministic design wave ($0.7 \times$ height of extreme deterministic design wave).
- Suffix *c* load cases specify the ‘reduced’ steady wind ($1.1/1.4 \times$ extreme steady wind) in conjunction with the extreme deterministic design wave.

The designer is given the option of considering either the stochastic load case *a* or deterministic cases *b* and *c* (where included). The wind and wave/sea state return periods are 50 years for ‘fault-free’ load cases 6.1 and 6.2 and one year for load cases 6.3 and 7.1, which involve extreme yaw and a machine fault respectively. Note that the grid loss load case 6.2 can include a yaw error of up to 180° , unless six hours back-up power supply is provided for the control and yaw system.

Note that in the suffix *a* load cases, it is assumed that the extreme stochastic sea state occurs at the same time as the extreme turbulent wind.

The suffix *b* and *c* load cases simplify the investigation of non-linear wave loading, which is computationally demanding for the multiple superposed linear waves forming the stochastic sea state of the suffix *a* load case. Unfortunately, the suffix *b* and *c* load cases cannot capture the dynamic response of the turbine structure to the turbulent wind, although Annex D of the standard describes how the steady wind speed should be increased above the design value to correct for this. Two deterministic load cases are required, because the possibility of the extreme gust and the extreme wave loadings occurring simultaneously is assumed to be negligible and instances of dominant wind and dominant wave loading both have to be catered for. The chosen reduction factors of 0.7 for wave height in the suffix *b* load cases and $1.1/1.4 (= 0.79)$ for wind speed in the suffix *c* load cases are based on work by Tarp-Johansen (2005). This showed that, if the wind and wave loading are treated as Gaussian quantities and the wave loading is drag-dominated, a reduction factor of $\sqrt{0.35} = 0.59$ for wave height in the suffix *b* dominant wind load cases and a reduction factor of $\sqrt{0.65} = 0.81$ for wind speed in the suffix *c* dominant wave load cases result in the best fit of those load cases to the total loading, regardless of the relative wind and wave load magnitudes.

Consideration of the effect of possible misalignment between wind and wave directions is required for all the non-operational load cases.

Extreme sea state

The stochastic sea state model for the IEC 61400–3 extreme sea state is the same as for the normal sea state in Section 11.2.5, but the significant wave heights to be considered are those with a 50 year or one year return period, depending on the load case. These may be determined by extrapolation based on the tail of the long-term distribution of significant wave heights. As with the normal sea state, the most adverse anticipated value of the peak spectral period is to be considered.

The other marine conditions consist of extreme currents (including currents generated by tides and storm surges) and water level range with the same recurrence period as the significant wave height combined with storm surge.

As the extreme wind speed U_{ref} is defined as the ten-minute mean, and the reference period for the significant wave height is normally three hours, a common averaging period for both parameters needs to be adopted for the simulations of extreme wind and wave loading in the suffix *a* load cases. The standard uses a common averaging period of one hour, defining the hourly mean wind speed as the ten-minute mean multiplied by a factor, k_1 , of 0.95 and

the one hour value of the turbulence standard deviation as the ten-minute value plus 0.2 m/s. Similarly, the significant wave height for a three-hour reference period is multiplied by the factor, k_2 , of 1.09 to convert it to the value appropriate to the shorter, one-hour reference period.

Normally at least six one-hour simulations are required, but shorter simulations are acceptable if it can be demonstrated that this will not reduce the extreme response – for example, by the use of the ‘constrained wave’ described in Section 11.3.10.

Extreme deterministic design wave

Two extreme deterministic design waves are specified – with return periods of 50 years and one year.

If the wave heights in the extreme sea states are Rayleigh distributed, then the wave height expected to be exceeded by the highest of N waves is $H_s \sqrt{0.5 \ln N}$. Taking $N = 1000$ as before results in an extreme wave height with 50-year and one-year return periods of $H_{50} = 1.86H_{s50}$ and $H_1 = 1.86H_{s1}$ respectively.

Modification of wave climate in shallow water

Waves begin to break when the wave height reaches 78% of the mean water depth, so the upper tail of the Rayleigh distribution of wave heights becomes curtailed as deep sea waves advance into shallower water, leading to conservatism in the use of the $1.86H_s$ formula for the wave height exceeded by the largest of 1000 waves. Battjes and Groenewald (2000) have developed a model for calculating the modified distribution of wave heights for waves propagating over a uniformly sloping seabed – see Annex C of IEC-61400–3. The model depends on the availability of measured site-specific wave data and is applicable provided the wave direction is within 30° of the direction of steepest seabed slope.

11.3.7 Fatigue loads

An overview of the IEC 61400–3 fatigue load cases – operational and non-operational – is given in Table 11.4, which identifies the equivalent onshore load case in IEC 61400–1 in each case.

The wind conditions and water levels denoted by the acronyms in the table above are as follows:

NTM Normal turbulence model

NWP Normal wind profile (following a power law with exponent of 0.14), without turbulence

NSS Normal sea state

NWLR Normal water level range

MSL Mean Sea Level

Sea currents are ignored.

A range of significant wave heights and wave periods should be considered at each wind speed, with the duration of each combination of wind speed and sea state extracted from a site specific three-dimensional scatter diagram. In particular T_p values corresponding to the natural frequency of the support structure should be investigated. However, it is preferable to avoid a very large number of simulations, so it is advantageous to lump load cases together

Table 11.4 Fatigue load cases

M/c state	Load case no.	Equiv LC on land	Wind conditions	Wind speed	Wave conditions	Water level	Grid, yaw and fault status
Power production	1.2	1.2	NTM	Cut-in – Cut-out	NSS Joint prob. distribution of $H_s, T_p, \bar{U}$	NWLR or $\geq \text{MSL}$	None
Power production with fault	2.4	2.4	NTM	Cut-in – Cut-out	Normal sea state i.e. $H_s = E[H_s \bar{U}]$	NWLR or $\geq \text{MSL}$	Fault condition
Start-up	3.1	3.1	NWP	Cut-in – Cut-out	Normal Sea State i.e. $H_s = E[H_s \bar{U}]$	NWLR or $\geq \text{MSL}$	None
Normal shut down	4.1	4.1	NWP	Cut-in – Cut-out	Normal sea state i.e. $H_s = E[H_s \bar{U}]$	NWLR or $\geq \text{MSL}$	None
Parked	6.4	6.4	NTM	$< 0.7U_{ref}$	NSS joint prob. distribution of $H_s, T_p, \bar{U}$	NWLR or $\geq \text{MSL}$	None
Parked with fault condition	7.2	None	NTM	$< 0.56U_{ref}$	NSS joint prob. distribution of $H_s, T_p, \bar{U}$	NWLR or $\geq \text{MSL}$	Fault condition

if possible. The errors due to coarsening the wave height and wave period bins can be reduced if the ‘damage equivalent mean wave height’ and ‘damage equivalent mean period’ are calculated for each bin. The former is determined as

$$H_s = \left(\frac{1}{N} \sum_{i=1}^N H_{s_i}^m \right)^{\frac{1}{m}}$$

where m is the inverse slope of the S – N curve, and the latter is taken as the reciprocal of the mean wave frequency. In a case study, Evans (2005) showed that, for a wind speed bin width of 2 m/s, the error in damage equivalent load resulting from a move from 0.5 m wave height and 0.5 s wave period bin widths to a single wave height/wave period bin with this approach was only a few per cent.

IEC 61400–3 specifies two fatigue load cases for machines that are parked or idling – one without and the other with a fault condition. The latter is absent from IEC 61400–1, but is considered necessary offshore because of the potentially much longer machine down-times and the large reduction in aerodynamic damping when the turbine is shut down.

The maximum wind speed to be considered for fatigue loading on a parked turbine in the absence of a fault (DLC 6.4) is $0.7U_{ref}$ – a value likely to be exceeded for 0.6 hour per year.

However, for fatigue loading on a parked turbine with a fault condition (DLC 7.2), the upper wind speed to be considered is only $0.56U_{ref}$, which is likely to be exceeded for 19 hours per year. No guidance is given on the likely total turbine down time corresponding to DLC 7.2 and clearly this needs to be predicted with great care.

11.3.8 Wave theories

Several wave theories exist to describe the fluid motion within regular, periodic, two-dimensional waves, but only three are routinely used for the calculation of loads on offshore structures – Airy, Stokes and Dean’s Stream Function Theories.

Airy wave theory is linear and, therefore, allows the water particle kinematics of a real sea state, consisting of a multitude of small waves of varying frequencies and wave lengths, to be modelled by superposition. However, as wave height increases in relation to water depth or wave-length, Airy theory becomes less accurate and non-linear theories such as Stokes and Dean’s Stream Function have to be employed instead. However, because of their non-linearity, these theories can only be applied to single, regular wave trains. Figure 11.11 shows

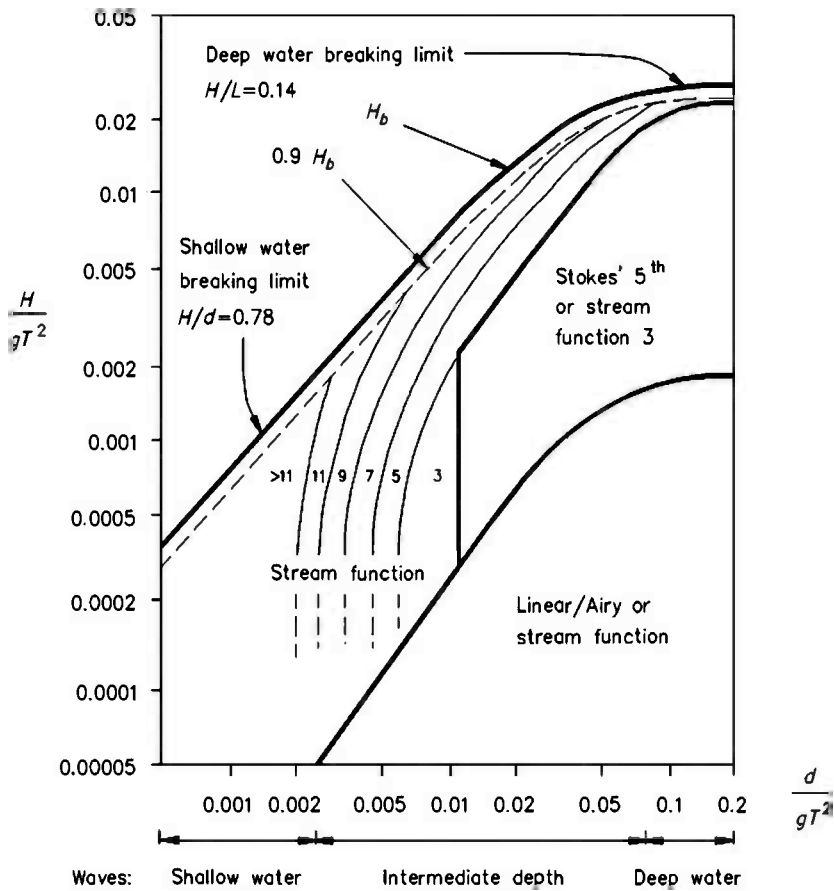


Figure 11.11 Regular wave theory selection diagram: log scales. Barltrop *et al.*, 1990. Reproduced under the Open Government License v1.0

which theories are applicable for different relationships between wave height, water depth and wave period.

It is apparent from Figure 11.11 that Dean's Stream Function theory can be used for all but near breaking waves. In view of this, and the relative complexity of Stokes Wave Theory, the latter is not discussed further and this section concentrates on the other two theories.

Airy wave theory

The fluid motion within regular, periodic two-dimensional waves must satisfy the Laplace equation, $\nabla^2 \phi(x, z, t) = 0$, where ϕ is the velocity potential, defined so that horizontal and vertical particle velocities are given by $u = -\partial \phi / \partial x$ and $w = -\partial \phi / \partial z$ respectively. x is the horizontal co-ordinate parallel to the direction of wave motion and z is the vertical co-ordinate measured upwards with respect to mean water level, as shown in Figure 11.12.

The fluid motion must also satisfy five boundary conditions as follows:

1. Zero flow vertical particle velocity, w , at the seabed, which is assumed horizontal.
2. Zero flow perpendicular to the free surface.
3. The unsteady form of the Bernoulli equation at the free surface.
4. The periodic boundary condition in time – $\phi(x, z, t) = \phi(x, z, t + T)$, where T is the wave period.
5. The periodic boundary condition in space – $\phi(x, z, t) = \phi(x + L, z, t)$, where L is the wave length.

Condition (2) is expressed as:

$$w = \frac{\partial \eta}{\partial t} + u \frac{\partial \eta}{\partial x} \quad (11.8)$$

where η is the water surface elevation above the mean and u is the horizontal water particle velocity. The atmospheric pressure acting on the free surface of the water is treated as constant, so the unsteady form of the Bernoulli equation in condition (3) becomes:

$$-\frac{\partial \phi}{\partial t} + \frac{u^2 + w^2}{2} + gz = C(t) \quad (11.9)$$

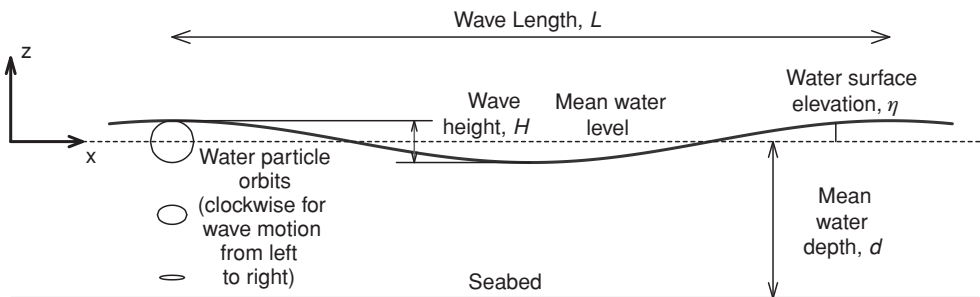


Figure 11.12 Parameter definitions and co-ordinates for regular, periodic, two-dimensional waves

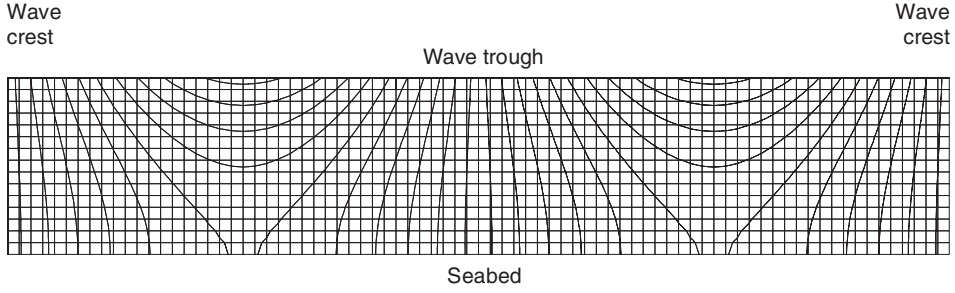


Figure 11.13 Velocity potential contours for Airy wave theory

According to Airy wave theory, the velocity potential at any point in the water is:

$$\phi = -\frac{Hg}{2\omega} \frac{\cosh k(z+d)}{\cosh kd} \sin(kx - \omega t) \quad (11.10)$$

where H is the wave height, d is the mean water depth, $k = 2\pi/L$ and $\omega = 2\pi/T$. Figure 11.13 shows the corresponding velocity potential contours over a full wave length for the case of a 15 m water depth and 80 m wave length. Note that the contours are shown as extending only up to still water level, as Airy wave theory only approaches 100% accuracy as the wave height becomes vanishingly small. Assuming the waves are moving from left to right, the maximum velocity potential is at the water surface at the right hand quarter point.

The equation for velocity potential satisfies conditions (1), (4) and (5) above exactly, but only satisfies ‘linearised’ versions of Equations 11.8 and 11.9.

The ‘linearised’ version of Equation 11.9 is obtained by expanding the left hand side of the equation with respect to the value at $z = 0$ using a truncated Taylor’s series to give.

$$\begin{aligned} \left(-\frac{\partial \phi}{\partial t} + \frac{u^2 + w^2}{2} + gz \right)_{z=\eta} &= \left(-\frac{\partial \phi}{\partial t} + \frac{u^2 + w^2}{2} + gz \right)_{z=0} \\ &+ \eta \left(-\frac{\partial^2 \phi}{\partial t \partial z} + \frac{\partial}{\partial z} \left(\frac{u^2 + w^2}{2} \right) + g \right)_{z=0} + \dots = C(t) \end{aligned} \quad (11.11)$$

Then, assuming all first order terms are small, all second order terms are very small, and can be neglected, giving

$$\left(-\frac{\partial \phi}{\partial t} \right)_{z=0} + \eta g = C(t) \quad \text{or} \quad \eta = \frac{1}{g} \left(\frac{\partial \phi}{\partial t} \right)_{z=0} + \frac{1}{g} C(t) \quad (11.12)$$

Substituting Equation 11.10 in Equation 11.12 and noting that the spatial and temporal means of the water surface elevation are defined as zero yields $C(t) = 0$ and

$$\eta = \frac{H}{2} \cos(kx - \omega t) \quad (11.13)$$

Application of a similar procedure to Equation 11.8 yields $w = \partial\eta/\partial t$ at $z = 0$, and substitution of

$$w = -\left(\frac{\partial\phi}{\partial z}\right)_{z=0} = \frac{H g k}{2\omega} \frac{\sinh kd}{\cosh kd} \sin(kx - \omega t) \quad (11.14)$$

and $(\partial\eta/\partial t)_{z=0} = (H/2)\omega \sin(kx - \omega t)$ results in

$$L = \frac{gT^2}{2\pi} \tanh kd \quad (11.15)$$

The horizontal and vertical particle velocities at any depth, u and w , are given by

$$u = -\left(\frac{\partial\phi}{\partial x}\right) = \frac{H g k}{2\omega} \frac{\cosh k(z+d)}{\cosh kd} \cos(kx - \omega t) = \frac{H\omega}{2} \frac{\cosh k(z+d)}{\sinh kd} \cos(kx - \omega t) \quad (11.16)$$

$$w = -\left(\frac{\partial\phi}{\partial z}\right) = \frac{H g k}{2\omega} \frac{\sinh k(z+d)}{\cosh kd} \sin(kx - \omega t) = \frac{H\omega}{2} \frac{\sinh k(z+d)}{\sinh kd} \sin(kx - \omega t) \quad (11.17)$$

The particle displacements can be obtained by integrating the above expressions. The water particles follow elliptical orbits with the major axis horizontal, as shown in Figure 11.12.

Figure 11.14 shows the accuracy of Airy wave theory in predicting the horizontal particle velocity at the wave crest by comparison with the more accurate Dean/Stokes 5th theories for different H/gT^2 and d/gT^2 ratios. This Figure is taken from *Dynamics of Fixed Marine Structures* by Barltrop and Adams (1991), which contains similar Figures applicable to the prediction accuracy for other parameters.

Dean's stream function theory

Dean's stream function theory (Dean and Dalrymple, 1984) is effectively an extension of Airy wave theory achieved by the addition of a series of harmonic terms to the expression for velocity potential, which enable the theory to satisfy the boundary conditions at the free surface (Equations 11.8 and 11.9) without the need for 'linearisation'. However, rather than use the velocity potential, the theory is developed with reference to the parameter known as the *stream function*, which is now introduced.

The paths followed by water particles within the waves are termed streamlines, along each of which the *stream function* parameter has a constant value. The stream function is defined such that its gradient perpendicular to the streamlines is proportional to the local particle velocity, so that

$$u = -\frac{\partial\psi}{\partial z} \quad \text{and} \quad w = \frac{\partial\psi}{\partial x} \quad (11.18)$$

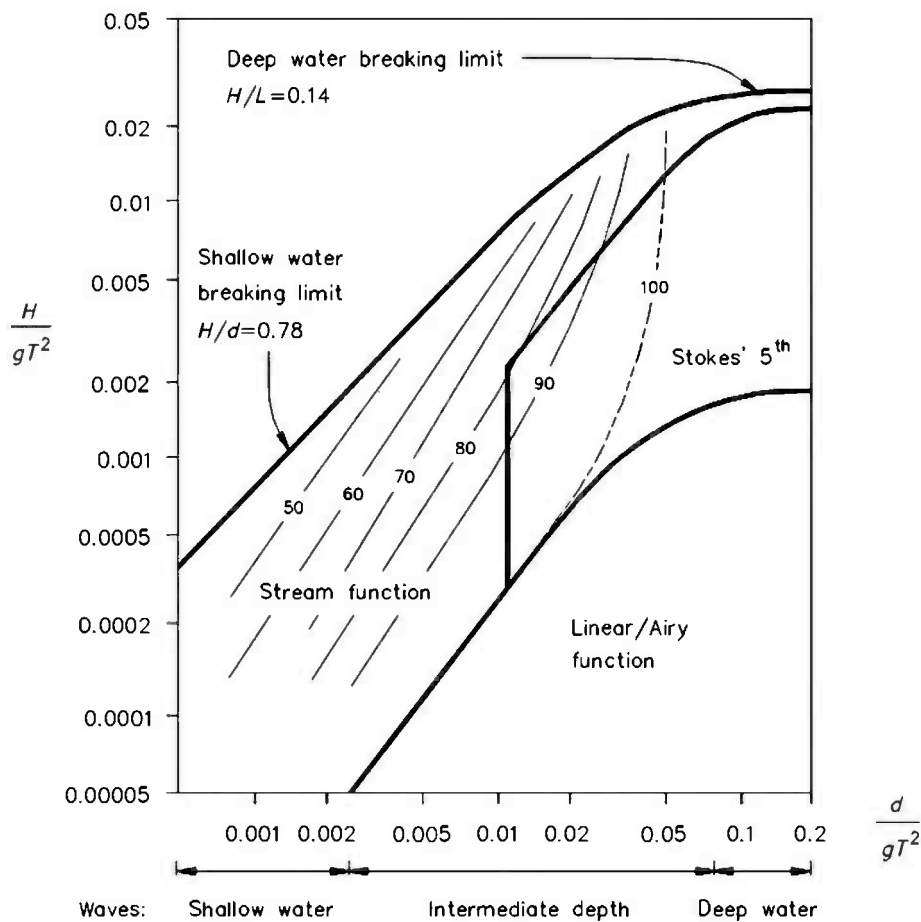


Figure 11.14 Horizontal particle velocity at the wave crest: Airy theory as % of theory shown in Figure 11.11. Barltrop *et al.*, 1990. Reproduced under the Open Government License v1.0

For Airy wave theory, the stream function is

$$\psi = -\frac{Hg}{2\omega} \frac{\sinh k(z+d)}{\cosh kd} \cos(kx - \omega t) \tag{11.19}$$

The corresponding streamlines are shown over a full wave length in Figure 11.15, for the case of a 15 m water depth and 80 m wave length.

The streamlines in Equation 11.19 and Figure 11.15 are defined with respect to a fixed frame of reference. However, it is perfectly possible to define them with respect to a *moving* frame of reference which travels at the same speed as the wave crests and troughs – that is,

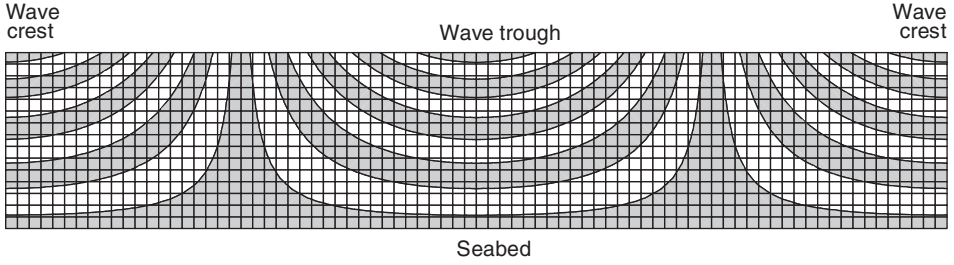


Figure 11.15 Streamlines for Airy wave theory with the frame of reference fixed. Note that, as with the figure illustrating the velocity potential contours, the streamlines are shown as only extending up to still water level, because Airy wave theory only approaches 100% accuracy as the wave height becomes vanishingly small

with celerity $C = L/T$. Then the Airy wave theory stream function becomes

$$\bar{\psi} = Cz - \frac{Hg}{2\omega} \frac{\sinh k(z+d)}{\cosh kd} \cos(kx) \quad (11.20)$$

which has the advantage of being invariant with time. The streamlines corresponding to the moving frame of reference are shown over a full wave length in Figure 11.16, for the same case as above. However, for clarity a relatively large wave height of 4 m is chosen, for which the Airy solution is inaccurate.

Dean's stream function is defined with respect to the same moving frame of reference, and consists of a harmonic series of terms as follows:

$$\bar{\psi} = Cz + \sum_{n=1}^N X_n \sinh nk(z+d) \cos(nkx) \quad (11.21)$$

For $N = 1$, Equation 11.21 equates to Equation 11.20, so

$$X_1 = -\frac{Hg}{2\omega \cosh kd} \quad (11.22)$$

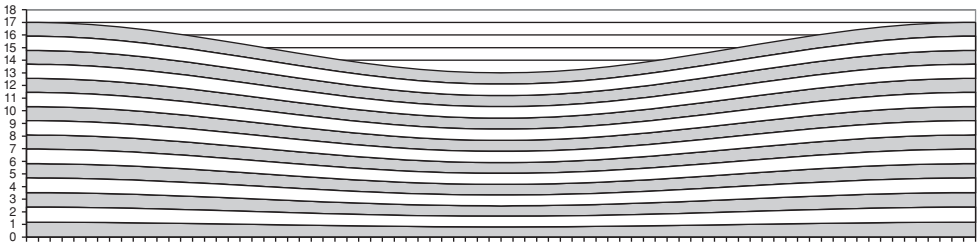


Figure 11.16 Streamlines for Airy wave theory with frame of reference moving with the wave train

The corresponding water particle velocities are:

$$u = -\frac{\partial \bar{\psi}}{\partial z} = -C - \sum_{n=1}^N nk X_n \cosh nk(z+d) \cos(nkx) \quad (11.23)$$

$$w = \frac{\partial \bar{\psi}}{\partial x} = -\sum_{n=1}^N nk X_n \sinh nk(z+d) \sin(nkx) \quad (11.24)$$

The number of terms, N , required increases as the height of the waves approaches the breaking limit. A solution using N terms is described as N th order. Substituting $z = \eta$ at the free surface in Equation 11.21 gives:

$$\eta = \frac{\bar{\psi}_0}{C} - \frac{1}{C} \sum_{n=1}^N X_n \sinh nk(\eta+d) \cos(nkx) \quad (11.25)$$

Given that the spatial average of the water surface elevation, η , is zero by definition, we have

$$\bar{\psi}_0 = \frac{1}{L} \int_0^L \sum_{n=1}^N X_n \sinh nk(\eta+d) \cos(nkx) dx \quad (11.26)$$

(The expression on the right hand side of this equation is non-zero by virtue of the variable weighting of $\cos(nkx)$ by the hyperbolic factor, $\sinh nk(\eta+d)$.)

The boundary conditions to be satisfied are as follows:

1. Zero flow vertical particle velocity, w , at the seabed, which is assumed horizontal.
2. Zero flow perpendicular to the free surface.
3. The steady form of the Bernoulli equation at the free surface.
4. The periodic boundary condition in space $\bar{\Psi}(x, z) = \bar{\Psi}(x + L, z)$, where L is the wave length.

Equations 11.24 and 11.21 satisfy condition (1) and (4) respectively, while condition (2) is satisfied because the free surface is a streamline.

The steady form of the Bernoulli equation at the free surface (condition (3)) is:

$$\left(\frac{u^2 + w^2}{2} \right) + g\eta = K \quad (11.27)$$

If the wave length, the mean water depth and the wave height, H , are the known quantities, the solution to the problem resolves itself into finding of a set of values of the X_n coefficients combined with a value of celerity, C , such that Equation 11.27 is satisfied as closely as possible at all points along the wave profile subject to the condition $\eta_{\max} - \eta_{\min} = H$.

The following steps enable a low order solution to be obtained on a spreadsheet:

1. Set $X_2, X_3 \dots X_N$ all equal to zero.
2. Set $C = \sqrt{(g/k) \tanh kd}$, the value according to Airy wave theory.
3. Set X_1 somewhat smaller than given by Equation (11.22) – say $X_1 = -0.8Hg/(2\omega \cosh kd)$.
4. Calculate the water level profile at M equally spaced points over a half wave length using repeated iterations of Equations 11.25 and 11.26.
5. Using Equation 11.27, calculate values of $K_1, K_2, \dots K_m, \dots K_M$ at these M points.
6. Calculate the summation $\sum_{m=1}^M (K_m - \bar{K})^2$, where $\bar{K} = \sum_{m=1}^M K_m/M$.
7. Minimise the above summation with respect to X_1 and C , subject to the condition $\eta_{\max} - \eta_{\min} = H$, to obtain new values of X_1 and C .
8. Repeat steps 4–7 with X_2 allowed to vary as well.
9. Repeat steps 4–7 with X_2 and X_3 allowed to vary as well.
10. Etc., etc.

Streamlines derived from a fifth order solution using this method with an M of 40 are shown over a half wave length in Figure 11.17, for the case of a 15 m water depth, 100 m wave length and 10 m wave height. It is evident that the wave profile is far from sinusoidal, with a narrow crest and a broad trough, so that the crest is much higher above mean sea level than the trough is below it.

Horizontal particle velocities may be calculated from the stream function using Equation 11.23. The profiles of horizontal particle velocities relative to a fixed frame of reference beneath the crest and beneath the trough are shown in Figure 11.18 for the above example.

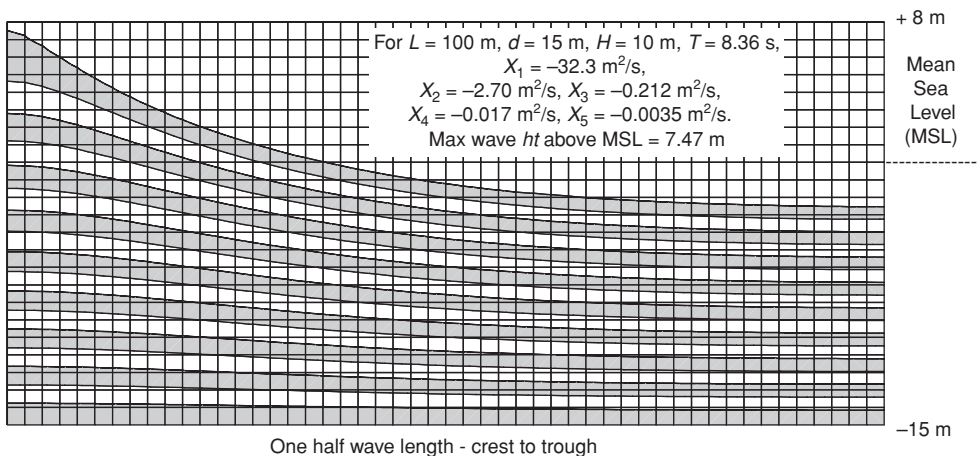


Figure 11.17 Streamlines for Dean stream function wave theory with moving frame of reference

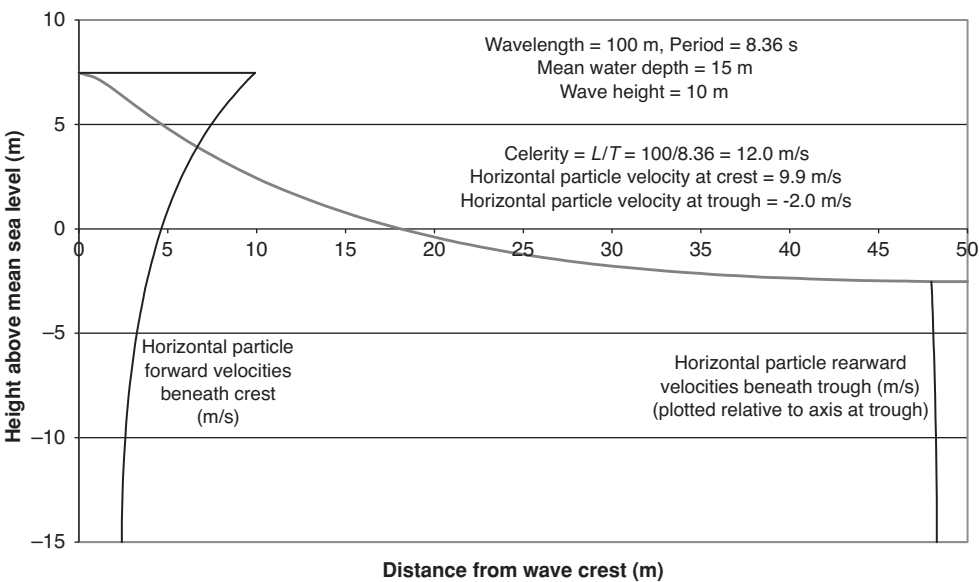


Figure 11.18 Horizontal particle velocity profiles below wave crest and trough. Note the high horizontal particle velocity at the crest, which approaches the wave celerity

A useful on-line resource providing the Dean stream function theory coefficients and the wave profile for user input wave parameters is to be found at <http://www.coastal.udel.edu/faculty/rad/streamless.html> at the time of writing.

11.3.9 Wave loading on support structure

The wave loading on immersed structures depends on their size, shape and surface roughness. For elements that are small compared with the wavelength, the disturbance to wave behaviour caused by the structure can be ignored and the loading based on the undisturbed wave kinematics, using Morison’s equation. However, for larger elements, it is necessary to take account of the way the wave pattern is modified. This can be done by means of a diffraction analysis, in which the varying pressures on discrete facets of the structure are calculated by numerical methods.

Morison’s equation

For structural elements with a width less than about one fifth of the wave length, it is normal to describe the loading in terms of separate, additive drag and inertia components, as given by Morison’s equation:

$$F = \frac{1}{2} C_D \rho D |u| u + C_M \rho A \frac{\partial u}{\partial t} \tag{11.28}$$

in which F is the force per unit length of the member, D the member width, A its cross-sectional area, C_D the drag coefficient, and C_M the inertia coefficient. The water particle velocity, u , and the acceleration are the undisturbed values that would apply at the member centreline. Note that the drag loading given by the first term is non-linear, as it depends on the square of the water particle velocity. It reaches its peak when the wave crest passes the member, whereas the inertia loading reaches its peak beforehand ($T/4$ beforehand for Airy waves).

Although Morison's equation applies to prismatic members in general, its application to cylindrical members, whether single monopiles or elements of a jacket structure, is of most interest here.

Values of drag and inertia coefficients

The empirical drag and inertia coefficients are dependent on:

- The Reynolds number, uD/ν , where ν is the kinematic viscosity of seawater ($1.35 \times 10^{-6} \text{ m}^2/\text{s}$ at 10°C).
- The relative roughness of the cylinder, k/D , where k is the surface roughness in terms of the peak to valley height.
- The Keulegan-Carpenter number, $K_C = u_{\max}T/D$, where $u_{\max}$ is the maximum water particle velocity (at the water surface, $u_{\max} = H\omega/2 \tanh kd$ (Equation 11.16), so $K_C = \pi H/D \tanh kd$).
- The ratio of current to the maximum wave-induced water particle velocity.

Influence of Reynolds number In the case of wind turbine support structures, the Reynolds number will correspond to the post-critical regime in all but the calmest seas, with the result that the steady flow drag coefficient, C_{DS} , is independent of Reynolds number.

Influence of relative roughness The steady flow drag coefficient increases from about 0.65 for smooth cylinders to about 1.05 for rough cylinders, as illustrated in Figure 11.19. This shows the recommendations of ISO 19902:2007 'Petroleum and natural gas industries – fixed steel offshore structures' and DNV-RP-205 'Environmental conditions and environmental loads' (April 2007), which are fairly similar with respect to C_{DS} .

The effect of marine growth on both surface roughness and member diameter must be considered below high tide level. Terminal thicknesses of marine growth of 200 mm and 60–100 mm are quoted in HSE Offshore Technology Report 2001/010 'Environmental considerations' and DNV-RP-C105 respectively, but marine growth is very area specific. If it is pessimistically assumed that the surface peak to valley height approximates to the terminal thickness, then the relative roughness of a typical 4 m diameter wind turbine monopile may be expected to be in the range 0.015–0.05, resulting in a steady flow drag coefficient of 1.05–1.10.

Above high tide level, marine growth will be negligible and the values of surface roughness, k , given in the following Table 11.5 can be used.

Influence of Keulegan-Carpenter number The oscillatory flow associated with progressive waves means that the cylinder wake transfers from one side of the cylinder to the other as the flow reverses. After flow reversal, the volume of water that was previously slow moving in the wake acquires increased velocity in the reverse direction compared to that which would

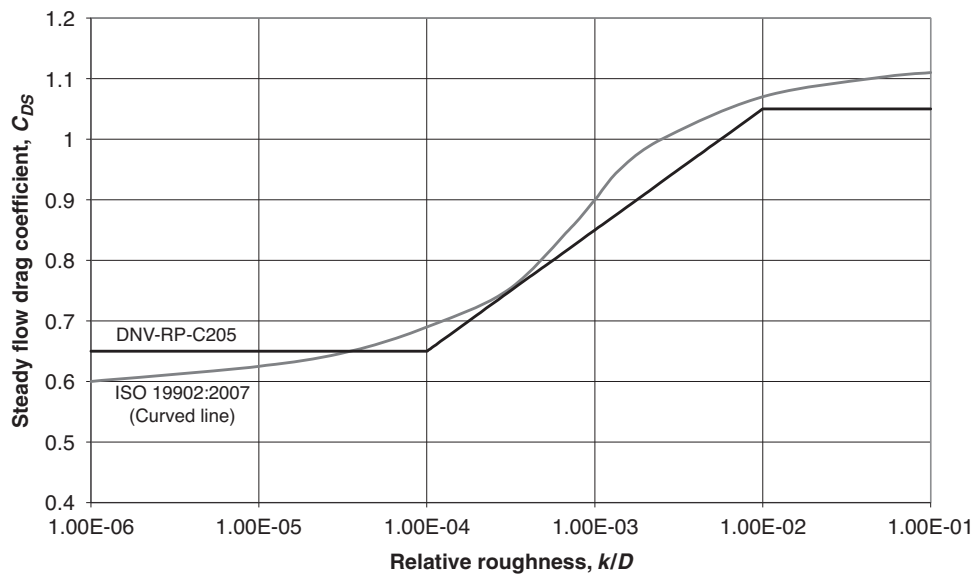


Figure 11.19 Dependence of steady flow drag coefficient on relative roughness

have been obtained in the absence of the cylinder. In calculating the drag coefficient for wave loading, C_D , this effect is allowed for by multiplying the steady flow drag coefficient, C_{DS} , by a wake amplification factor, ψ , dependant on the Keulegan-Carpenter number. Figure 11.20 shows the relationship between ψ and K_C given in DNV-RP-C105 for rough and smooth cylinders. Those given in ISO 19902:2007 are generally very similar.

At very low Keulegan-Carpenter numbers, the inertia coefficient may be taken as 2.0, the theoretical value from potential theory in the absence of flow separation. For $K_C > 3$ it begins to drop with the onset of flow separation, reaching asymptotic values of 1.6 and 1.2 for smooth and rough cylinders respectively. Figure 11.21 shows the relationship between C_M and K_C given in DNV-RP-C105 for rough and smooth cylinders. Those given in ISO 19902:2007 are essentially the same.

Influence of current The effect of a steady current is to move the drag coefficient C_D closer to the steady flow value, C_{DS} , regardless of the direction of the current relative to the direction of wave propagation. In the case of an in-line current, C_D can be taken as equal to C_{DS} when the current is greater than 0.4 times the maximum orbital velocity due to wave motion.

Table 11.5 Surface roughnesses (from DNV-RP-C205)

Material	Surface roughness, k (millimetres)
Steel, new, uncoated	0.05
Steel, painted	0.005
Steel, highly corroded	3
Concrete	3

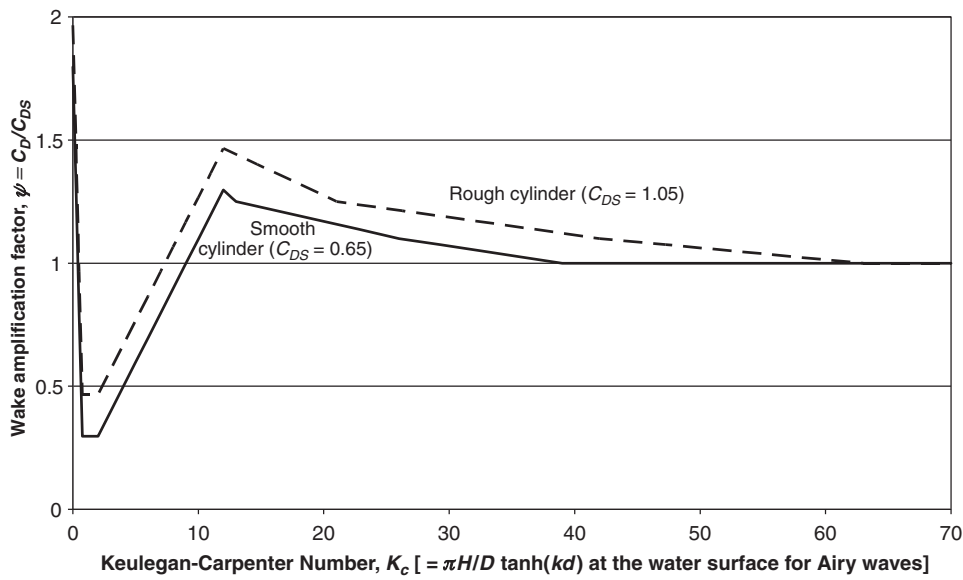


Figure 11.20 Variation of wake amplification factor, $\psi = C_D/C_{DS}$ with Keulegan-Carpenter number from DNV-RP-C205

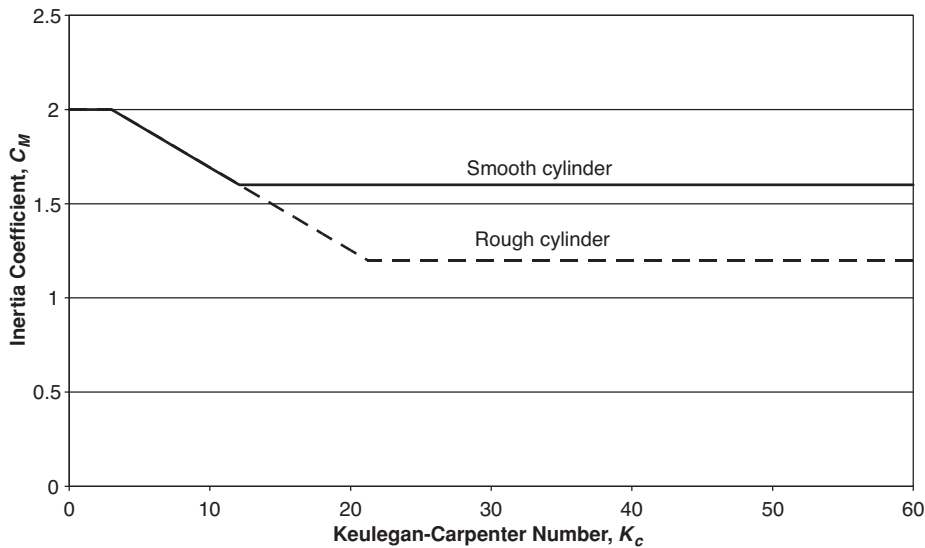


Figure 11.21 Variation of inertia coefficient, C_M , with Keulegan-Carpenter number from DNV-RP-C205

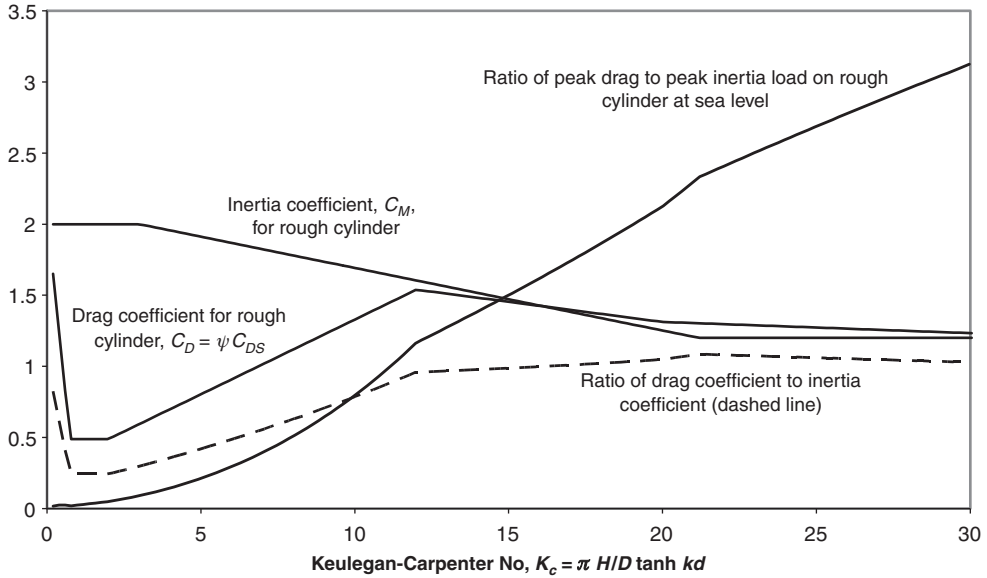


Figure 11.22 Variation of C_D , C_M , C_D/C_M and the ratio of maximum drag force to maximum inertia force at sea level with Keulegan-Carpenter number for a rough cylinder

Ratio of drag force to inertia force

For Airy waves, it can be shown that the ratio of the maximum drag force per unit length to the maximum inertia force per unit length at sea level is:

$$\frac{F_{D,Max}}{F_{M,Max}} = \frac{C_D}{C_M} \frac{H}{D} \frac{1}{\tanh(kd)} = \frac{C_D}{C_M} \frac{K_C}{\pi^2} \quad (11.29)$$

Figure 11.22 shows how C_D and C_M vary with Keulegan-Carpenter number for a rough cylinder ($C_{DS} = 1.05$), based on the recommendations of DNV-RP-C105. The resulting variation in the ratio of maximum drag force to maximum inertia force has been derived from Equation 11.29, and is shown in the same figure.

Drag dominant or inertia dominant regimes can be defined according to which load component is the greater. For a rough cylinder, the transition from the inertia dominated regime to the drag dominated regime occurs when the Keulegan Carpenter number exceeds $\pi^2 C_M / C_D$ – that is, approximately $\pi^2 (1.64/1.45) = 11.2$, so for deep water, with $\tanh kd$ approaching unity, drag force dominates when the wave height to cylinder diameter ratio exceeds $11.2/\pi$ – that is, about 3.5. In shallow water, where $\tanh kd$ is less than unity, this threshold is reduced in proportion. It can be seen that, for normal sea states, the loading on a monopile of around 4.0 m diameter is likely to be dominated by inertia loading.

Airy wave loading on cylinder

The cylinder loading from an Airy wave is easily found by inserting the horizontal particle velocity (Equation 11.16) and acceleration into Equation 11.28. However, as Airy wave theory

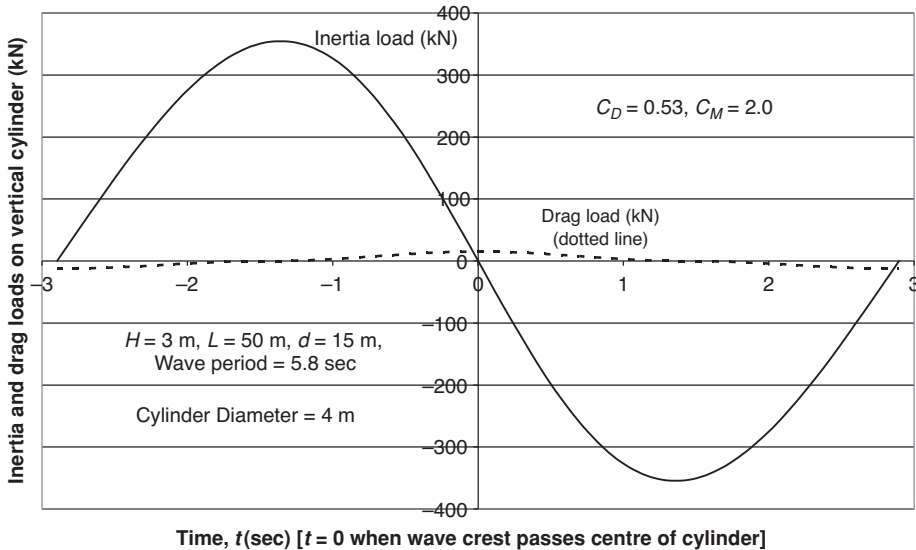


Figure 11.23 Variation of wave loading on a 4 m diameter vertical cylinder over a complete wave period for a 3 m Airy wave and 15 m mean water depth

is based on infinitesimal waves, it does not provide velocities and accelerations above still water level. Accordingly it is normal to make the assumption that the kinematics at the instantaneous free surface are the same as those given by Airy theory at still water level, and that at intermediate depths the kinematics are those that would be given by Airy theory after the water column had been compressed or stretched uniformly to a height corresponding to mean water depth. This procedure was proposed by Wheeler (1970) and is known as *Wheeler stretching*.

Figure 11.23 illustrates how the drag and inertia loads on a 4 m diameter vertical cylinder integrated over the full local water depth vary with time for a 3 m Airy wave. As expected, the drag component is much less than the inertia component. It can be seen that Wheeler stretching displaces the inertia load peak closer to the crest.

Clearly, in this case, the drag load does not contribute to the maximum total load at all. In the general case, the drag load only begins to affect the maximum total load when K_C exceeds $0.5\pi^2 C_M / C_D$.

Cylinder loading for non-linear waves

In the case of larger, non-linear waves, the water particle velocities and accelerations input into Morison's equation have to be based on a non-linear wave theory. Figure 11.24 shows the variation of drag and inertia loading on a 4 m diameter vertical cylinder in the case of 10 m regular waves, based on fifth order Dean stream function theory. The drag and inertia coefficients used are based on the recommendations of DNV-RP-C105. The loads are integrated over the full local water depth as before.

It is evident that the drag loading reaches a pronounced peak as the crest passes the cylinder, due to the increased depth and high water particle velocity in the wave

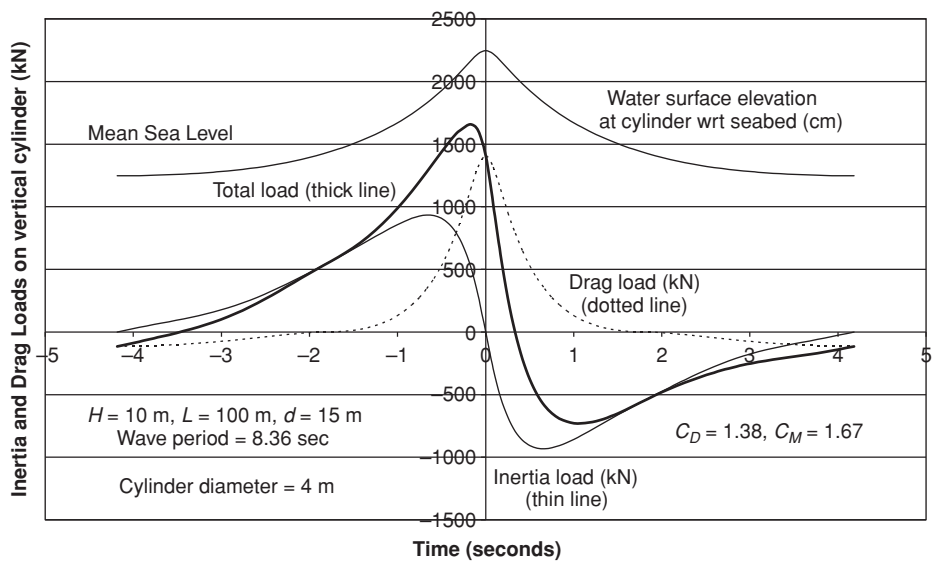


Figure 11.24 Variation of wave loading on a 4 m diameter vertical cylinder over a complete wave period for a 10 m regular wave and 15 m mean water depth

crest – 9.9 m/s compared with the wave celerity of 12 m/s. Also, the phase lead of the inertia loading maximum is much reduced compared with the $T/4$ applicable in the case of an Airy wave. It is striking that the ratio of the peak drag load to peak inertia load has increased by a factor of 35 compared with the Airy wave example above, although the $(C_D/C_M) \cdot (H/D)$ ratio has only increased by a factor of 10.4.

Diffraction

Structural elements that are of significant width compared with the wave length modify the wave behaviour, so the transparency assumption implicit in Morison’s equation no longer applies. Instead, a diffraction analysis has to be used, in which the modified wave flow around the structure is obtained in terms of a velocity potential distribution that satisfies the extra requirement that there should be no flow perpendicular to the surface of the structure. Note that the diffraction analysis determines the inertia loading on the structure, but not the drag loading. This is normally of little concern as the drag loads on large structures are usually negligible beside the inertia loads. A more significant limitation of diffraction theory is that it is restricted to linear waves, as the velocity potential only satisfies linearised versions of the boundary conditions at the free water surface.

A closed-form analytical solution for the velocity potential in terms of Bessel functions has been developed for the case of a large vertical cylinder projecting above the water surface by MacCamy and Fuchs (1954). The method adopted is to treat the velocity potential as composed of two parts – the first the velocity potential that would exist in the absence of the cylinder and the second one that cancels out the flows across the cylinder surface postulated by the first. The result was used in conjunction with the unsteady Bernoulli equation to calculate the pressure distribution on the cylinder and hence the load per unit length. Figure 11.25 shows

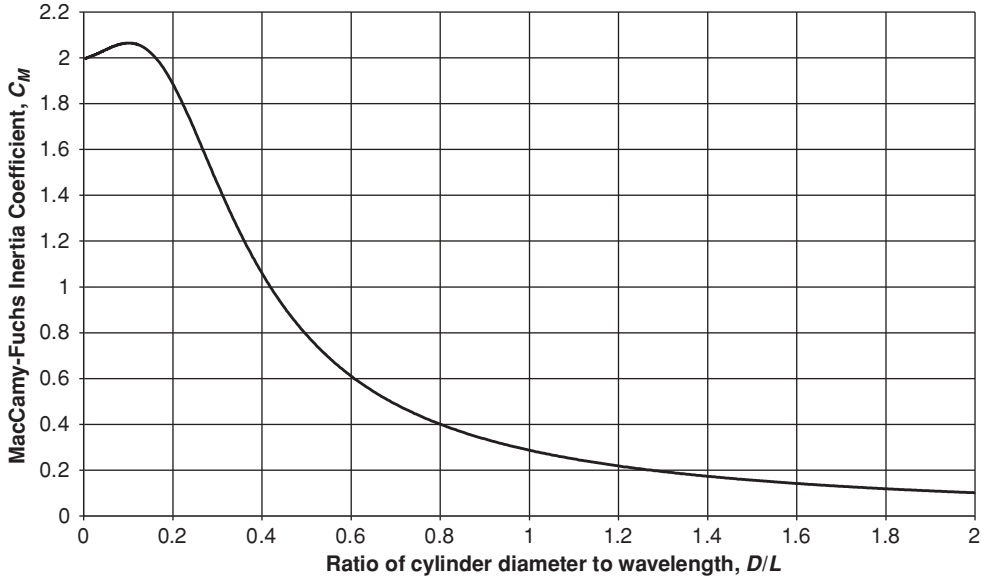


Figure 11.25 Effect of large cylinder diameter on inertia coefficient, based on MacCamy and Fuchs

how the resulting inertia coefficient varies with the diameter to wave length ratio, D/L . It is seen that after an initial small increase, the inertia coefficient drops rapidly as D/L increases.

The derivation of the MacCamy and Fuchs expressions for the velocity potential and cylinder loading is described in Dean and Dalrymple (1984).

In the general case, no closed-form solution will be available and resort has to be made to numerical methods. The method again involves determination of the velocity potential that would exist in the absence of the structure and the derivation of a second velocity potential superposed upon the first that results in zero flow across the structure surface, while satisfying the Laplace equation and the other boundary conditions. The surface of the structure is divided up into N area elements or facets, each with its own oscillating flow source taken as uniformly distributed over the facet, and the second velocity potential is taken as the summation of the velocity potentials resulting from these N oscillating sources. Then application of the zero flow requirement at each facet results in N simultaneous equations for the determination of each source magnitude.

Denoting the source intensity at facet i as f_i per unit area and the velocity potential due to a unit oscillating source radiating from the centre of facet i as $\Phi_i(x, y, z, t)/4\pi$, the total potential gradient perpendicular to facet j due to all N oscillating sources becomes:

$$\begin{aligned} \left(\frac{\partial \phi_S}{\partial n} \right)_j &= \frac{1}{4\pi} \left[\left(\frac{\partial \Phi_1}{\partial n} \right)_j f_1 S_1 + \cdots + \left(\frac{\partial \Phi_i}{\partial n} \right)_j f_i S_i + \cdots + \left(\frac{\partial \Phi_N}{\partial n} \right)_j f_N S_N \right] \\ &= \frac{1}{4\pi} \sum_i \left(\frac{\partial \Phi_i}{\partial n} \right)_j f_i S_i \end{aligned}$$

where S_i is the surface area of facet i .

For zero flow across facet j , $(\partial\phi_I/\partial n)_j = -(\partial\phi_S/\partial n)_j$, where $(\partial\phi_I/\partial n)_j$ is the potential gradient perpendicular to the facet that would occur in the absence of the structure. This leads to a set of equations

$$\left(\frac{\partial\phi_I}{\partial n}\right)_j = -\frac{1}{4\pi} \sum_i \left(\frac{\partial\phi_i}{\partial n}\right)_j f_i S_i \quad (11.30)$$

which can be solved for the source intensities, f_i .

Wehausen and Laitone (1960) give an expression for $\Phi_i(x, y, z, t)$ on page 478 of *Surface Waves*. In addition to the expected terms $(1/r)\cos\omega t$ associated with the point source and $(1/r_2)\cos\omega t$ associated with an imaginary ‘mirror’ source beneath the seabed (where r and r_2 are the radii from the point source and ‘mirror’ source respectively to the point x, y, z), there are two other terms involving Bessel functions of the first kind, one in phase and the other 90° out-of-phase with the source. These terms are required to satisfy the free surface boundary conditions and the expected decay towards infinity.

As noted above, diffraction theory can only provide an exact solution for linear waves. However, an approximate solution for non-linear waves can be obtained by increasing the load on each facet of the structure surface obtained from the diffraction analysis by the ratio of the non-linear wave particle acceleration to the linear wave particle acceleration at the facet, treating the structure as transparent.

Breaking waves

Deep water waves break when they reach shallower water such that the wave height to water depth ratio exceeds 0.78, and can result in short-term loads an order of magnitude greater than those from non-breaking waves.

Breaking waves are normally divided into three different types – spilling, plunging or surging. Muir Wood and Fleming (1981) distinguish them as follows: ‘In principle a plunging breaker occurs when the crest velocity exceeds that of the body of the wave, a spilling breaker occurs when the crest velocity remains approximately equal to that of the wave and a surging breaker occurs when the base of the wave surges up the beach before the crest can plunge forward’. Clearly plunging breakers are of greatest concern in relation to wind turbine structures, because the advancing wall of water causes impulsive loads on vertical surfaces.

The hydrodynamics of wave impact on a structure is a complex three-dimensional problem, depending on the form of the structure, the shape of the wave front and the presence of entrained air, so simplifying assumptions are necessary. Figure 11.26 illustrates a plunging breaker striking a vertical cylinder. Although the surface of the front of the wave is typically concave, it is normal to treat it as vertical, with a height λ times the wave crest elevation, η_b , and consider the water flow around the cylinder as a two-dimensional problem. λ is known as the curling factor.

Various theories for the impulsive loading on the cylinder as the breaking wave envelops it have been proposed, and are summarised in Wienke and Oumeraci (2005). The time histories of the impulsive force per unit length for some of the theories are presented in Figure 11.27.

It can be seen that the magnitude of the initial impulsive force coefficient, $f/(\rho V^2 R)$, is quite uncertain, ranging from π to 2π . The higher value results from allowance being made for the pile-up of water on either side of the cylinder (Figure 11.28).

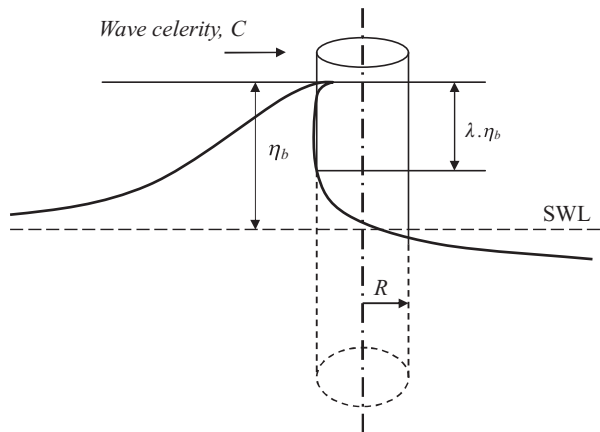


Figure 11.26 Wave breaking at vertical cylinder

The force on the cylinder equates to the integral of the pressure distribution over the immersed surface area, obtained from the unsteady Bernoulli equation

$$p(x, t) = \rho \frac{\partial \phi}{\partial t} - \frac{1}{2} \rho \left[\left(\frac{\partial \phi}{\partial x} \right)^2 + \left(\frac{\partial \phi}{\partial y} \right)^2 \right] + C(t) \quad (11.31)$$

The gravity term is omitted, as the height is constant.

Velocity potential utilising flat plate idealisation The problem of determining the velocity potential, ϕ , is simplified if the instantaneous flow field is assumed to be that which would

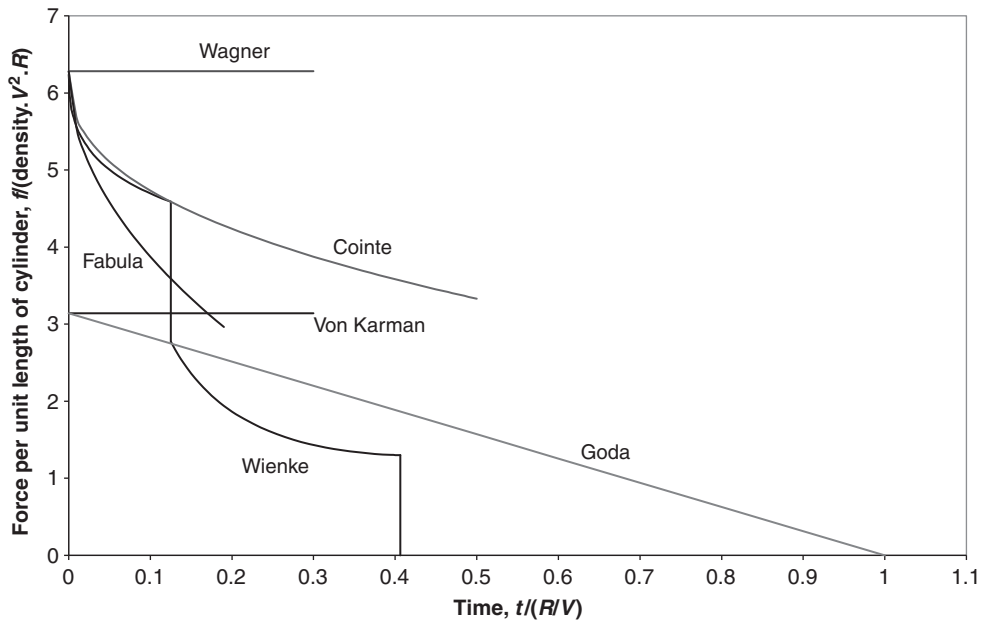


Figure 11.27 Time histories of impulsive force on cylinder according to different theories

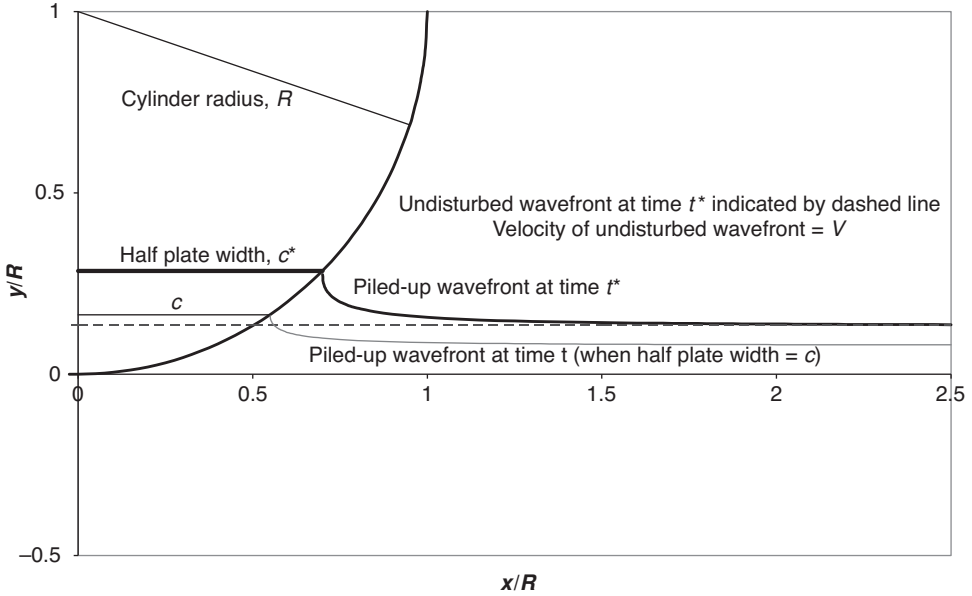


Figure 11.28 Development of water pile-up as wavefront advances around cylinder

apply to flow around a *flat* plate of width equal to the immersed width of the cylinder at that instant (Figure 11.28). The complex potential for the flow around a flat plate perpendicular to the mean flow direction is given by Milne-Thomson (1967) as

$$w = -iV\sqrt{z^2 - c^2} \quad (11.32)$$

where V is the flow velocity remote from the plate in the negative y direction, $z = x + iy$, and the width of the plate is $2c$. The velocity potential is given by the real part of w and equates to $\phi = V\sqrt{c^2 - x^2}$ at the upstream surface of the plate.

At a point far upstream of the plate, the fluid pressure, $\partial\phi/\partial t$ and $\partial\phi/\partial x$ are all zero, so $C(t) = 0.5\rho V^2$ and Equation 11.31 becomes

$$\begin{aligned} p(x, t) &= \rho \frac{\partial}{\partial t} \left(V\sqrt{c^2 - x^2} \right) - \frac{1}{2}\rho \left(\frac{\partial\phi}{\partial x} \right)^2 + \frac{1}{2}\rho V^2 \\ &= \frac{\rho Vc}{\sqrt{c^2 - x^2}} \frac{dc}{dt} - \frac{1}{2}\rho V^2 \frac{x^2}{c^2 - x^2} + \frac{1}{2}\rho V^2 \end{aligned} \quad (11.33)$$

Wavefront pile-up Wagner (1932) set out a method for calculating the accelerated advance of the wavefront around each side of a boat hull impacting a still water surface which can be used to obtain dc/dt for a wavefront impacting on a cylinder (see Figure 11.28). From

Equation 11.32, it can be shown that the velocity of flow past the notional plate at a point in the plane of the plate is

$$\left(\frac{\partial \phi}{\partial y}\right)_{y=0, x>c} = \frac{V}{\sqrt{1-(c/x)^2}} \quad (11.34)$$

The integral of this expression with respect to time gives the y co-ordinate of the point (c^*, y) where the wavefront meets the cylinder surface at time t^*

$$y(t^*) = \int_0^{t^*} \frac{V dt}{\sqrt{1-(c/c^*)^2}} \quad (11.35)$$

We now define the quantity $\mu = V/(dc/dt)$, so that $V dt = \mu dc$ and

$$y(t^*) = \int_0^{c^*} \frac{\mu \cdot dc}{\sqrt{1-(c/c^*)^2}} \quad (11.36)$$

If we suppose that $\mu = \alpha_1 c + \alpha_3 c^3 + \alpha_5 c^5 + \dots$, the integral can be performed and the values α_n determined from the requirement that $y(t^*)$, c^* follow the profile of the cylinder. For example, if μ is taken as $\alpha_1 c$.

$$y(t^*) = \int_0^{c^*} \frac{\alpha_1 c \cdot dc}{\sqrt{1-(c/c^*)^2}} = \alpha_1 \left[-c^* \sqrt{1-(c/c^*)^2} \right]_0^{c^*} = \alpha_1 c^{*2} \quad (11.37)$$

Solution for parabolic profile Equating $\alpha_1 c^{*2}$ to the parabolic approximation to a circle, $y(t^*) = 0.5c^{*2}/R$, yields $\alpha_1 = 0.5/R$. Hence $\mu = 0.5c/R$ and $dc/dt = V/\mu = V \cdot 2R/c$, which can be substituted in Equation 11.33 to give

$$p(x, t) = \frac{2\rho V^2 R}{\sqrt{c^2 - x^2}} + \frac{1}{2}\rho V^2 - \frac{1}{2}\rho V^2 \frac{x^2}{c^2 - x^2} = \frac{2\rho V^2 R}{\sqrt{c^2 - x^2}} + \frac{1}{2}\rho V^2 \left(2 - \frac{c^2}{c^2 - x^2} \right) \quad (11.38)$$

This expression is integrated over the width of the plate to give the force per unit length on the cylinder

$$\begin{aligned} f &= 2\rho V^2 R \cdot 2 \int_0^c \frac{dx}{\sqrt{c^2 - x^2}} + \frac{1}{2}\rho V^2 \cdot 2 \int_0^c \left(2 - \frac{c^2}{c^2 - x^2} \right) dx \\ &= 4\rho V^2 R \arcsin\left(\frac{x}{c}\right) + \rho V^2 \int_0^c \left(2 - \frac{c^2}{c^2 - x^2} \right) dx = \rho V^2 R \left\{ 2\pi + \frac{2c}{R} - \frac{c}{R} \left[\tanh^{-1}\left(\frac{x}{c}\right) \right]_0^c \right\} \end{aligned} \quad (11.39)$$

The third term in Equation 11.39 is theoretically infinitely negative because the pressure tends to minus infinity at the edge of the notional flat plate. However, if negative pressures are ignored, a conservative result may be obtained. The initial force per unit length, for $c = 0$, is $2\pi\rho V^2 R$, which is an order of magnitude greater than the drag force for this velocity.

Two stage solution Wienke (2001) approximates the circular profile by the above parabola for $c^*/R < 1/\sqrt{2}$ and by a quartic curve beyond. This results in the stepped force time-history shown in Figure 11.27.

Multi-term solution for near circular profile The circular profile of the cylinder, $y(t^*)/R = 1 - \sqrt{1 - (c^*/R)^2}$, can be approximated by the binomial series

$$y(t^*) = \frac{1}{2} \left(\frac{c^*}{R} \right)^2 + \frac{1}{8} \left(\frac{c^*}{R} \right)^4 + \frac{1}{16} \left(\frac{c^*}{R} \right)^6 + \frac{5}{128} \left(\frac{c^*}{R} \right)^8 + \dots \quad (11.40)$$

Substitution in Equation 11.36 with $\mu = \alpha_1 c + \alpha_3 c^3 + \alpha_5 c^5 + \dots$ results in

$$\begin{aligned} \mu = & \frac{1}{2} \left(\frac{c}{R} \right) + \frac{3}{16} \left(\frac{c}{R} \right)^3 + \frac{15}{128} \left(\frac{c}{R} \right)^5 + \dots \\ & + \frac{(2n+1)(2n-1)(2n-3) \dots (2n-1)(2n-3)(2n-5) \dots}{2n(2n-2)(2n-4) \dots} \frac{(c/R)^{2n+1}}{(2n+2)(2n)(2n-2) \dots} \end{aligned}$$

Although only a few terms of the series (Equation 11.40) are required to represent the profile for $c^*/R < 0.8$, a large number of terms are required as c^*/R approaches unity. Equation 11.33 for the pressure distribution becomes

$$p(x, t) = \frac{\rho V^2 c / \mu}{\sqrt{c^2 - x^2}} + \frac{1}{2} \rho V^2 \left(2 - \frac{c^2}{c^2 - x^2} \right) \quad (11.41)$$

The force per unit length is obtained by integrating Equation 11.41 over the immersed width, ignoring negative pressures near the edge of the notional plate as before. Denoting the value of x at which the pressure changes sign as c' , the force per unit length on the cylinder is then

$$f = \rho V^2 R \left\{ \frac{2c}{\mu R} \arcsin \left(\frac{c'}{c} \right) + \frac{2c'}{R} - \frac{c}{R} \tanh^{-1} \left(\frac{c'}{c} \right) \right\} \quad (11.42)$$

The relationship between the time elapsed since the wave first hits the cylinder and the immersed width, $2c$, is obtained by integrating the equation $V dt = \mu dc$. Figure 11.29 shows the resulting time history of the force per unit length obtained when 100 terms are used in the series expression for μ .

Note that the plot becomes inaccurate as c/R approaches unity, because of the inevitable inaccuracy of the series approximation to a circle there. However, this is less of a concern than the fact that the flat plate idealisation becomes unrealistic for large c/R anyway.

Experimental measurements An extensive series of measurements on vertical and inclined cylinders subjected to breaking waves in the wave tank at the Forschungszentrum Küste, Hannover have been reported by Wienke and Oumeraci (2005). Pressure measurements around the cylinder circumference confirmed the validity of the Wagner wavefront pile-up model and measurements of the total loading indicated a maximum value of the curling factors of 0.46 for vertical cylinders. This is in line with the range of 0.4–0.5 given by Goda (1966).

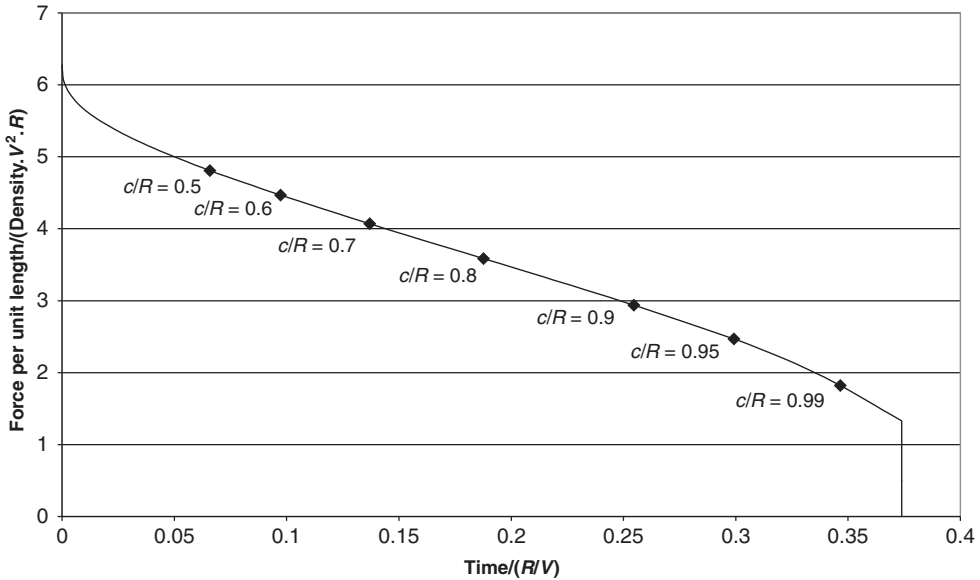


Figure 11.29 Time history of force per unit length on cylinder due to breaking wave using flat plate idealisation

11.3.10 Constrained waves

The analysis of structural loading caused by large waves in a random sea is a complex undertaking, because the linear wave theory, which applies to the myriad individual tiny sinusoidal components of the waveform, when each is on its own, breaks down when they are combined together, because of the large resulting wave amplitude. Annex D of IEC 61400–3 suggests several approaches to simplifying the problem, one of which – the constrained wave approach – involves the embedment of a single non-linear regular wave (commencing and ending at a trough) into a series of irregular, linear waves forming a longer simulation.

Clearly, a smooth transition between the simulated waveform and the embedded wave is required in order to preserve a basic similarity between the wave kinematics of the two. Ideally one would find a wave in the simulated waveform of similar magnitude to the specified regular wave and replace the former with the latter. However, many simulations are necessary to find a large enough wave, and the troughs on either side of the crest are unlikely to be of the same height. Furthermore, the crest height above still water level is normally about half the wave height for any simulated wave, instead of about three quarters for the embedded non-linear wave.

In order to circumvent these difficulties, Rainey and Camp (2007) have proposed constraining the elevation and slope of the simulated waveform at three points coinciding with the crest and troughs of the embedded wave. The constrained waveform can be obtained by ‘correcting’ an initial simulated waveform using New Wave theory (Tromans *et al.*, 1991) so that it satisfies the imposed conditions.

Bierbooms (2006) explains the method as applied to wind simulation, and derives an expression for the time history of a constrained gust. The analogous expression for the time

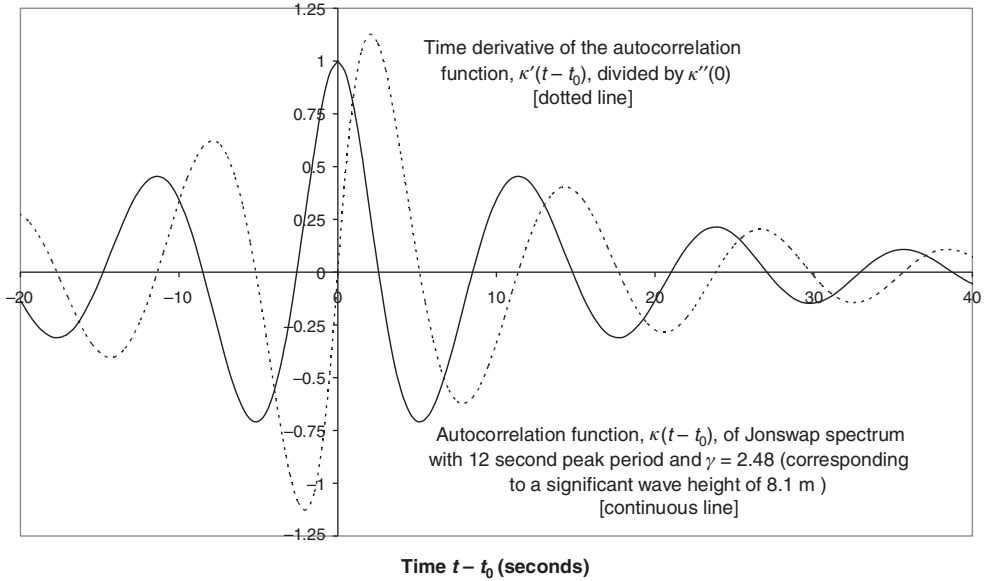


Figure 11.30 Jonswap spectrum autocorrelation function and its time derivative

history of water surface elevation constrained to have an extreme value of A at time $t = t_0$ is as follows:

$$\eta_c(t) = \eta(t) + \kappa(t - t_0)(A - \eta(t_0)) - \frac{\dot{\kappa}(t - t_0)}{\ddot{\kappa}(0)}\dot{\eta}(t_0) \quad (11.43)$$

Here $\eta(t)$ is the water surface elevation obtained from the initial simulation, $\kappa(t - t_0)$ is the water surface elevation autocorrelation function, $\dot{\kappa}(t - t_0)$ is its time derivative and $\ddot{\kappa}(0)$ is its second time derivative at time $t = t_0$. The autocorrelation function and its time derivative (normalised by $\ddot{\kappa}(0)$) are plotted out in Figure 11.30, which illustrates how the two correction terms fluctuate at the wave frequency and attenuate as $|t - t_0|$ increases.

The method can be extended to the imposition of constraints on water surface elevation and slope at three successive points in time. Figure 11.31 shows an example water surface elevation simulated time history and desired constrained elevations of C at the crest (time t_2) and D at the preceding and succeeding troughs (times t_1 and t_3).

The equation for the simulated time history of water surface elevation with these six constraints is

$$\begin{aligned} \eta_c(t) = & \eta(t) + a\kappa(t - t_1)(D - \eta(t_1)) + b\frac{\dot{\kappa}(t - t_1)}{\ddot{\kappa}(0)}\dot{\eta}(t_1) + c\kappa(t - t_2)(C - \eta(t_2)) \\ & + d\frac{\dot{\kappa}(t - t_2)}{\ddot{\kappa}(0)}\dot{\eta}(t_2) + e\kappa(t - t_3)(D - \eta(t_3)) + f\frac{\dot{\kappa}(t - t_3)}{\ddot{\kappa}(0)}\dot{\eta}(t_3) \end{aligned} \quad (11.44)$$

where a, b, c, d, e and f are six unknown coefficients. These can be determined by solving the six simultaneous equations obtained by substituting $t = t_1, t = t_2$ & $t = t_3$ in turn in

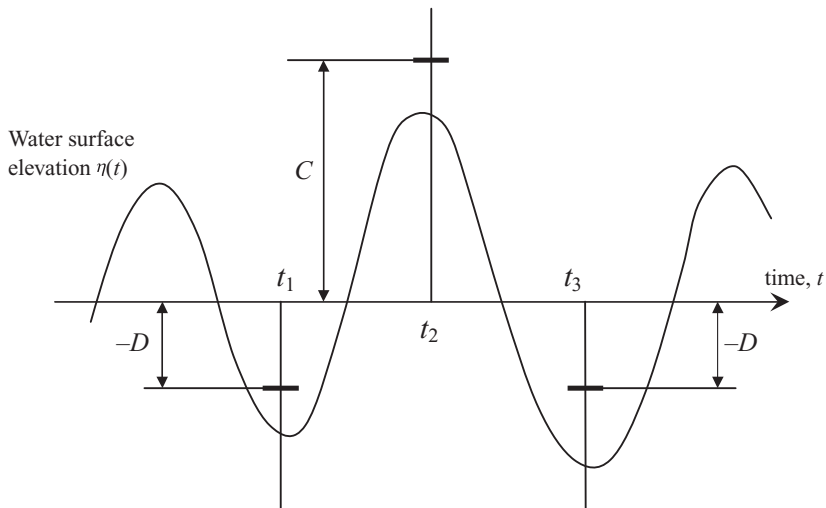


Figure 11.31 Simulated water surface time history and desired constraints at times t_1 , t_2 and t_3

Equation 11.44 and in its time derivative, and equating $\eta_c(t_1)$ to D , $\eta_c(t_2)$ to C , $\eta_c(t_3)$ to D and $\dot{\eta}_c(t_1)$, $\dot{\eta}_c(t_2)$ and $\dot{\eta}_c(t_3)$ to zero.

Figure 11.32 provides an illustration of a simulated water surface elevation time history constrained to permit the insertion of a 15 m regular wave. A Jonswap spectrum was used for the simulation with a significant wave height of 8.1 m – that is, that expected to yield an

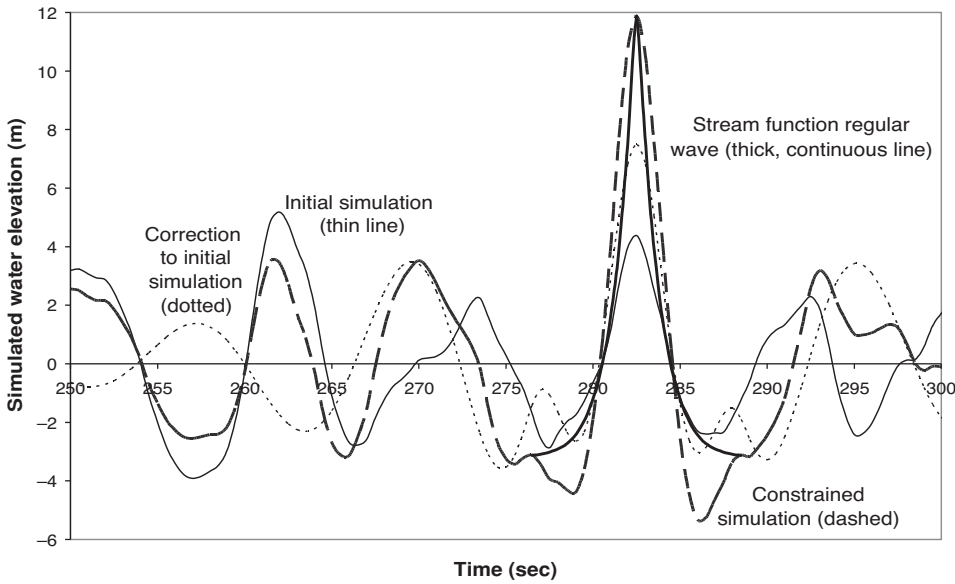


Figure 11.32 Example of a simulated water surface elevation time history constrained to enable the insertion of a 15 m regular wave

extreme wave exceeding 15 m in a three-hour reference period. The wave period corresponding to the spectral peak frequency was 12 seconds.

The profile of the embedded wave, which replaces that of the constrained simulation over the 12 seconds commencing at 276.5 seconds, is derived from fifth order stream function theory. Rainey and Camp (2007) propose the use of cosine blending functions in the vicinity of each trough to smooth the inevitable discontinuities in the wave kinematics (and hence the wave loading) at the boundary between assumed linear and non-linear behaviour. The water particle velocities and accelerations in the transition regions are then calculated as weighted averages of those derived from the linear and non-linear solutions, with the weighting determined by the blending functions.

Although this does not provide a rigorous solution to the wave equation, it does provide a pragmatic engineering approach in which the background irregular sea state sets the initial dynamic conditions of the structure before the passage of the non-linear extreme wave.

11.3.11 Analysis of support structure loads

Loadings on oil and gas offshore structures have customarily been calculated considering deterministic, regular, non-linear waves for extreme loads and exposure to different stochastic sea states for fatigue loads. In the latter case, spectral analysis permits the dynamic behaviour of the structure to be accounted for using a frequency-dependent transfer function relating response to excitation, but depends on the assumption of linear waves and relatively small drag loading, permitting 'linearization' of the drag term in Morison's equation.

The loading on wind turbine support structures differs from that on the typical offshore structure for oil or gas extraction in several ways:

1. Wind turbines are sited in shallower waters, so that non-linearity of wave loading is more important.
2. Wind turbines project much higher above still water level, which, in conjunction with shallow water siting, results in wind loading being much more dominant.
3. The wind loading on the support structure of an operating wind turbine is completely different from that on a oil platform or a parked wind turbine, because of the large thrust loading even in low winds, the pronounced randomly varying periodic loading resulting from gust slicing and the active control of rotor speed and pitch angle.
4. Wind turbines are typically supported on a single column, which is less stiff than the lattice structures normally used for oil or gas platforms.

The randomly varying periodic loads referred to in (3), combined with non-linear aerodynamics above rated wind speed, are the main factors driving the use of multiple simulations in the time domain for the analysis of wind turbine loads. In the case of offshore wind turbines, wave loads have to be input into the simulations as well.

The simulation of wave loads starts with the wave spectrum (Section 11.3.4) defining the sea state under investigation and involves the superposition of many tiny waves of different frequency and random phase, with each wave amplitude determined by the square root of the spectral density at the frequency concerned. This enables time histories of the water surface elevation and the water particle kinematics to be generated, under the key assumption

that *linear* wave theory applies. Finally, wave loads are calculated from the water particle kinematics using Morison's equation.

Although wave loads do not act on the rotor and nacelle directly, tower head accelerations due to support structure wave loading result in inertia loading on the turbine. However, these loads are normally relatively small, to the extent that onshore wind turbine designs can often be used offshore.

The treatment of wave loads in relation to extreme and fatigue load cases are considered separately below.

Extreme loads

Generally, extreme load cases are specified in terms of a turbulent wind and a stochastic sea state, which requires the assumption of linear wave theory in the calculation of the water particle kinematics and the derivation of the loads. Many of the load cases detailed in sections 11.3.5 and 11.3.6 are principally concerned with extreme *wind* loading, so the error in the linear wave assumption is not a concern. However, in the case of load cases 1.6a, 6.1a, 6.2a, 6.3a and 7.1a, which involve extreme waves, the standard requires non-linear wave kinematics to be properly represented (e.g. by the inclusion of a constrained wave, or by other methods set out in Annex D). Failing this, alternative load cases with regular, non-linear waves are to be investigated instead.

Fatigue loads

As set out in Section 5.9.4, Chapter 5, simulations in the time domain are preferred to spectral analysis for the evaluation of fatigue damage due to wind loading, because they account for both non-linear aerodynamics and the combination of deterministic and stochastic load components. In the case of offshore wind turbine structures, simulations in the time domain have the further advantage of directly modelling the aerodynamic damping of wave-induced tower motion and enabling the *combined* stress fluctuation resulting from the disparate wind and wave loadings to be obtained directly for rain-flow counting of the stress cycles.

The calculation of wave loads is again based on linear wave theory, but the errors introduced by this assumption are relatively minor, because of the smaller waves.

The analysis of fatigue loading in the time domain inevitably involves a large number of simulations, particularly when several turbine sites are being investigated. Although this method is advisable for final design verification, it is demanding of computing time, so fatigue analysis in the frequency domain is preferred for the development of an initial support structure design. This approach is discussed further in relation to monopile support structures in Section 11.5.2.

11.4 Machine size optimisation

In principle, the optimum turbine diameter for an offshore site can be estimated using a cost model and appropriate scaling rules in the same way as discussed for land-based machines in Chapter 6. One of the main challenges is to develop a realistic model for the support structure construction and installation cost, which is typically a significantly greater proportion of the total capital cost offshore. The model needs to account for differing water depths and foundation types as well as for different turbine diameters and should ideally be based on

cost data from wind farms that have already been constructed, although in practice such information is rarely easy to obtain.

Clearly the establishment of any permanent structure in the sea is a very costly undertaking, as is illustrated by the fact that a 100 m met mast supported on a monopile driven into the seabed can cost anything from €1 to €5 million (*Wind Energy – The Facts*, 2009). A rough wind turbine cost model can be developed by assuming that the installation costs of offshore wind turbines of different diameters contain a fixed element, dependent on water depth, which is of a similar order of magnitude to the cost of met mast deployment. This has been done by modifying the tentative NREL offshore wind turbine cost model (NREL, 2006), which is an extension of the land-based turbine cost model discussed in Section 6.2.3, Chapter 6, so that the turbine installation cost at a shallow water site comprises a fixed element of \$1 million plus a varying element equal to 10% of turbine capital cost. Following the NREL model, turbine capital cost is increased by 13.5% relative to land-based turbines to allow for marination.

The other major cost elements specific to offshore are the support structure for the tower and the undersea cable for grid connection. The support structure is assumed to be a steel monopile, weighing (and costing) twice as much as the tower for a 70 m diameter turbine. The length of the monopile is determined principally by water depth and wave height and is, thus, largely independent of turbine diameter, so monopile weight and cost are taken as scaling as turbine diameter squared rather than as diameter cubed. Following NREL, the cost of the grid connection for a given distance to shore is assumed to be proportional to turbine rated power and taken as \$300/kW. The NREL allowances of \$55/kW for scour protection and \$60k/turbine for personnel access equipment are also included.

Figure 11.33 illustrates the resulting variation of cost of energy with turbine diameter.

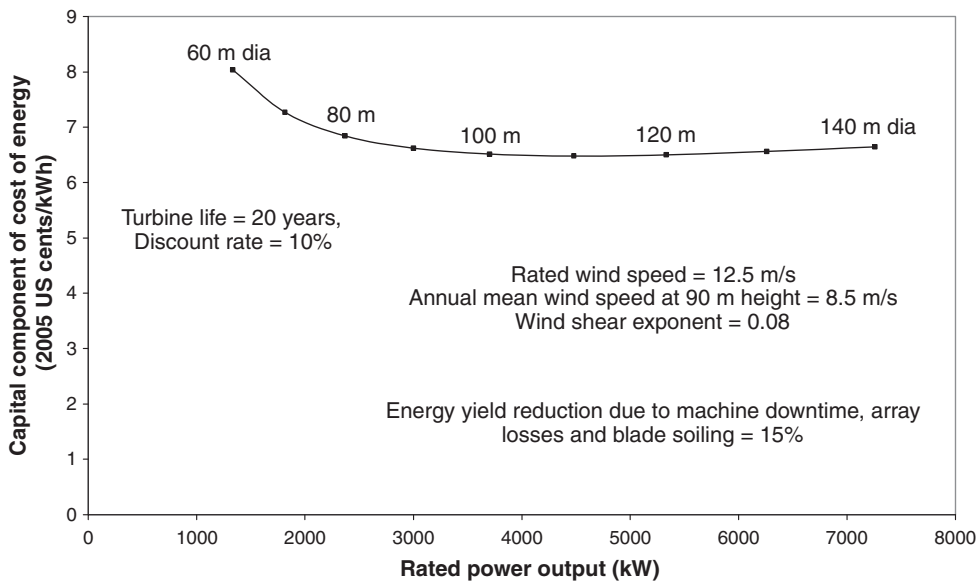


Figure 11.33 Variation of cost of energy with turbine diameter based on modified NREL cost model with a fixed component of turbine installation cost of \$1 million

The figure suggests that the optimum machine rating would be 4.5 MW, corresponding to a machine diameter of 110 m, for the chosen rated wind speed of 12.5 m/s. However, as set out above, the figure is based on several unsubstantiated assumptions and should, therefore, be treated as illustrative only. It is nevertheless interesting to note that the minimum capital component of the cost of energy of 6.5 US cents/kWh in 2005 shown by Figure 11.33 is almost double the corresponding value for the onshore machine (Figure 6.3 in Chapter 6), despite the increased annual mean wind speed.

At the time of writing the EU funded 'Upwind' project is researching the design challenges of very large wind turbines (8–10 MW). Preliminary findings are that the cost of energy would increase slightly with the move from 5 MW to 10 MW turbines offshore (Hendrix, 2008).

11.5 Reliability of offshore wind turbines

Wind turbines are expected to operate at high levels of reliability, whether sited on land or at sea, because any equipment failure is likely to lead to significant downtime while a repair crew is mobilised. Figure 11.34 shows the failure rates for different wind turbine sub-assemblies derived from two large European surveys, together with the downtimes per failure. (WMEP = Wind scientific Monitoring and Evaluation Programme, based at the Fraunhofer Institute of Wind Energy Systems, Kassel and LWK = Landwirtschafts-kammer, Schleswig-Holstein.)

In the case of offshore wind turbines, access is both time consuming and expensive in itself and also extremely weather sensitive, leading to much longer downtimes, so the premium on high reliability is an order of magnitude greater.

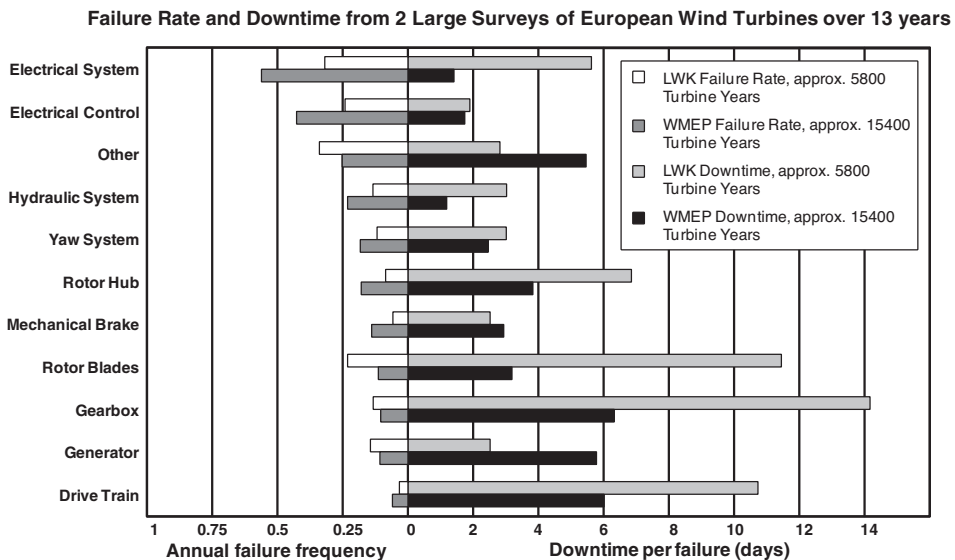


Figure 11.34 Wind turbine sub-assembly failure rates and downtime per failure based on 20,000 turbine years of data from two European surveys. Reproduced with the permission of the University of Durham and the Fraunhofer Institute of Wind Energy Systems

This section briefly considers measures to improve turbine reliability under the following headings – machine architecture, redundancy, protection against corrosion and condition monitoring.

Machine architecture

Although a wind turbine has an irreducible number of core components, the pursuit of maximum energy yield, grid compatibility and drive train compliance, particularly in the context of large machines, has led to greater complexity and the overwhelming dominance of variable speed, pitch regulated machines offshore. It so happens that the two additional systems required for this type of turbine – namely the frequency convertor and the pitch control mechanism – were found to be the systems responsible for the largest percentages of downtime (20% and 13% respectively) in the Reliawind reliability field study (Wilkinson *et al.*, 2010).

While it has to be accepted that large, fixed speed wind turbines are impractical because power swings are insufficiently damped, the benefit of the pitch control system can be questioned offshore, it being necessary to weigh up whether the extra energy generated justifies the net additional cost, after allowing for the energy lost in extra downtime. Clearly, in the absence of pitch control, an alternative braking system has to be provided – in the form of tip-brakes, for example – and its cost accounted for.

The use of a direct drive generator enables one major component, the gearbox, and its attendant failures to be eliminated. Although gearbox failure rates are relatively low, the resultant downtime was found to be the highest of all sub-assemblies in the WMEP and LWK studies (Figure 11.34). However, research by Spinato *et al.* (2009) indicates that the aggregate failure rate of generators and converters in direct drive wind turbines is greater than the aggregate failure rate of gearboxes, generators and converters in geared wind turbines, blunting the benefit of gearbox elimination. Nevertheless, the transition to direct drive may still result in improved availability as the downtimes of failed generators and converters are less than those of gearboxes.

Redundancy

While the provision of redundancy is mandatory for safety critical systems, the duplication of critical components to secure greater operational reliability also merits consideration, particularly offshore. Examples already implemented by some manufacturers include the deployment of two separate systems to measure blade pitch angle and the provision of spare frequency converter modules. A similar philosophy can be extended to the design of mechanical systems such as the pitch mechanism and yaw drive, whereby a reserve of strength is built in to increase the likely service life.

Component quality

Reliability benefits will justify the use of higher quality components offshore in some cases. For example, double row pitch bearings will be more robust than those with only a single race.

Spinato *et al.* (2009) observe that the failure rates of wind turbine generators and converters are higher than found in other industries, at least during the initial phases of operation, and recommend that they be subjected to more thorough testing. On the other hand they found

the reliability of wind turbine gearboxes to be on a par with that of comparable gearboxes in other industries.

Protection against corrosion

The concentration of salt particles in the marine atmosphere, which is much augmented by the wind (Gong *et al.*, 1997), means that extra measures must be taken to protect offshore wind turbine components. Besides high-spec surface coatings, these include the use of a sealed generator with an air to air heat exchanger and similar arrangements for other items of electrical equipment, such as the transformer. Air discharged from these heat exchangers helps to maintain the air in the nacelle above the dew point and prevent condensation. In addition, the nacelle and/or hub can be pressurised, to prevent the entry of moist, salty air. In the absence of a partition between the nacelle and hub, this requires the provision of a labyrinth seal between the spinner and nacelle to minimise air leakage.

Condition monitoring

Clearly reliability can be much improved if incipient defects can be detected by condition monitoring before they result in a systems failure. Traditionally the health of wind turbine systems has been monitored by measures such as:

- Automatic flagging of alarms when parameters such as oil pressures, bearing temperatures and vibration exceed levels short of the trip level.
- Regular inspections – for example, of gear teeth wear and high speed shaft alignment.
- Observation of machine running.
- Regular oil sampling.

These are all examples of condition monitoring, which has been defined as ‘the process of monitoring a parameter of condition of machinery, such that a significant change is indicative of a developing failure’.

The expense and difficulty of regular visits to offshore turbines has encouraged the expansion and development of remote or on-line conditioning monitoring techniques, particularly in relation to gear trains and bearings. They require detection of the onset of particular failure mechanisms and the establishment of measurable criteria for determining whether intervention is necessary. The Connrow Final Report entitled ‘Condition monitoring for offshore wind farms’ (Wiggelinkhuizen *et al.* 2007) and the Offshore M & R Final Report ‘Advanced maintenance and repair for offshore wind farms using fault prediction and condition monitoring techniques’ (ISET, 2006) provide helpful surveys of available techniques and their usefulness, together with reports on field trials, while Crabtree (2010) lists condition monitoring systems currently commercially available.

Some of the different condition monitoring techniques are considered below.

Drive train vibration monitoring

Accelerometers are mounted on the gearbox and generator casings, permitting the vibration signature at each location to be continuously monitored. For fixed speed machines, each signal

can be processed using a Fast Fourier Transform (FFT) to compute the vibration spectrum, which will contain spectral peaks corresponding to gear teeth meshing frequencies and shaft rotational speeds of the different gear stages. The behaviour of such vibration signatures over time can then be monitored in order to observe changes due to bearing wear or tooth damage, although this will be more difficult for some components, such as planet bearings, where the indirect vibration transmission path to the accelerometer on the casing makes the spectral peak difficult to detect. Also, in the case of the low speed shaft bearing, the shaft rotational speed is too low for wear detection by accelerometer.

Vibration signatures will vary with power level, wind speed and yaw error, so they should be sorted into different bins accordingly. Irregularities will be more apparent if data is centralised, so that comparisons between different turbines of the same type can be made.

In the case of variable speed machines, the frequencies of the various spectral peaks will be continuously changing, so order tracking techniques must be used to produce spectra in which the frequencies are normalised by the turbine rotational speed. One such method is Synchronous Order Tracking in which a shaft encoder is used to control the signal sampling rate at a fixed number per revolution before FFT analysis. Alternatively a once-per-rev tachometer signal can be fed into the analyser so that its sample rate is varied in proportion to RPM. A simpler option is to restrict vibration monitoring to periods of operation above rated wind speed, when the turbine rotational speed is almost constant.

It is clear from the above that a vibration monitoring system will produce a vast amount of data, which requires dedicated signal processing software programmed to flag up suspect vibration signature changes. Ideally a team of experts would be able to interpret the changes and rule on when action would be necessary, but the Conmow report concluded that, in 2007, there was as yet insufficient experience to permit such predictions, so the usual response to data abnormality was to shut down the machine or increase the frequency of inspections.

Lubricant oil debris detection

Equipment to measure debris concentrations in lubricating oil can detect the onset of a wear failure, without the need for regular oil sampling and analysis. However, the equipment has to be installed in the lubrication pipe work used when oil is delivered to bearings and gears under pressure, so is not applicable in the case of splash lubricated gearboxes.

Nacelle vibration monitoring

Nacelle accelerators mounted in the transverse and fore-aft directions can detect rotor mass imbalance and blade pitch errors. Rotor mass imbalance can increase due to water ingress or icing. Special accelerometers, known as static accelerometers, are required, capable of responding to the low blade rotation frequency (ISET, 2006).

Pitch bearing resistance

In the case of electrical pitch drives, pitch bearing condition can be monitored by measuring the motor current, large peaks in the signal being indicative of jerking movements due to bearings sticking.

Added value

The benefit of a condition monitoring system is clearly reduced if there are significant numbers of false alarms, which can be due to the over cautious setting of alarm levels or faults with the system itself. Although the Connrow study looked at the factors determining added value, including the level of false alarms, it drew no general conclusions about the overall benefits of installing a condition monitoring system.

11.6 Support structures

Support structures for offshore wind turbines are inevitably much more massive than those for onshore turbines, because of the greater hub height relative to the foundation and the additional loading from waves. As a result, they constitute a much higher proportion of offshore turbine installed cost than is the case on land, rendering economic design that much more important.

Most offshore wind turbine designs have evolved from onshore designs, so it has been common practice to utilise the onshore tower design offshore above wave crest level and interpose an additional structure – termed the sub-structure between it and the seabed. Although the term ‘support structure’ applies to the complete structure supporting the nacelle, this section will concentrate on the sub-structure.

Five different types of sub-structure have been utilised offshore to date – monopiles, jacket structures, tripods, tripiles and gravity bases – but monopiles have proved the most popular, and are considered first.

Fatigue analysis of monopiles in the frequency domain is explored in a separate section (Section 11.6.2).

11.6.1 Monopiles

The monopile is a steel cylindrical tube driven into the seabed, with or without the assistance of excavation from within. In soft rock, a socket may be drilled out beforehand to receive the pile. In water depths of up to about 10 m at lowest astronomic tide (LAT), monopile diameters are usually in the range 4.5–5 m.

Transition piece

In view of the difficulty in maintaining the monopile accurately vertical during the driving operation, it is normal to introduce an additional cylindrical tube – termed the ‘transition piece’ – between the monopile and the base of the tower. The tower is then bolted to an integral flange at the top of the transition piece. Figure 11.35 shows a typical arrangement. (Exceptionally, however, at Scroby Sands wind farm, driving conditions were considered sufficiently favourable for the towers to be bolted directly to integral flanges at the top of the monopiles.)

The transition piece is usually sleeved over the monopile, with an overlap length 1.5 times the diameter. A gap of 50–100 mm between the tubes allows the transition piece to be set vertical, following which the annular space is grouted up. Design of the grouted joint is considered under a separate sub-heading below.

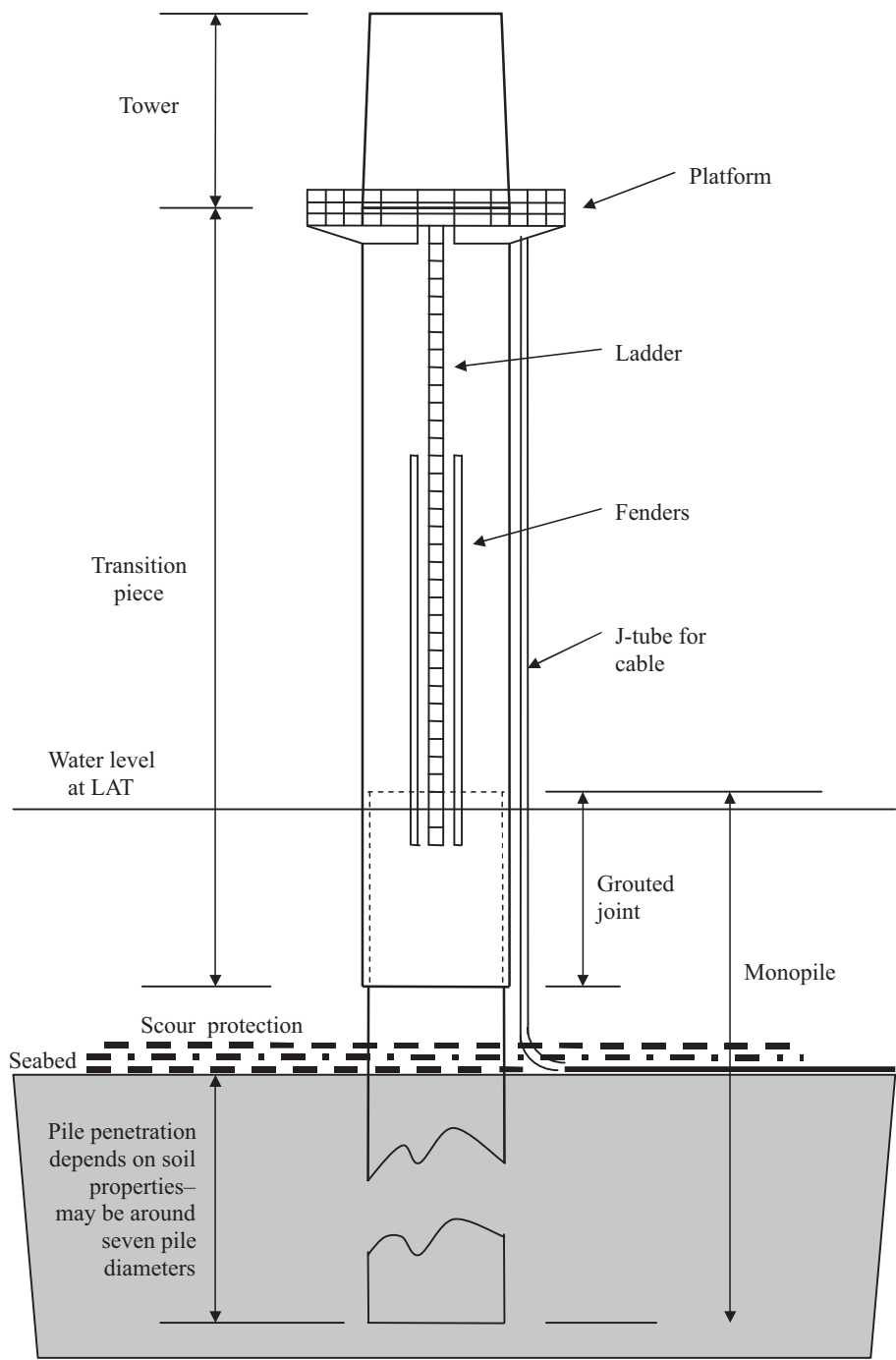


Figure 11.35 Typical arrangement of monopile and transition piece

It is normal to locate an external platform encircling the tower base at the top of the transition piece, to facilitate turbine access. Boat access is by means of a ladder fixed to the outside of the transition piece between a pair of vertical tubes which act as fenders.

The height of the platform, and thus of the top of the transition piece, is normally set so that there is an adequate clearance, known as the air gap, between the platform and the crest of the highest wave, including allowance for tidal range and storm surge. The target air gap may be 1.5 m. If, as is sometimes the case, the top of the monopile is set at 0.5 m above LAT, the length of the transition piece required comes to the sum of the sleeve length, tidal range, the storm surge, and the crest elevation of the 50-year return wave, ηH_{50} , plus 1 m. In UK waters with significant tidal range, transition pieces over 25 m long may be used, leading to weights approaching 200 tonnes.

In deeper waters, where monopiles 6 m in diameter and above are required, it may be beneficial to taper the transition piece diameter down above the grouted joint in order to reduce exposure to wave loading and match up with the smaller tower base diameter.

Cable ducts

The wind turbine power cables are routed through steel protective tubes known as J tubes, which may be located either inside or outside the transition piece/monopile. The J tubes are so called because they incorporate a 90° bend at seabed level to enable the cable to exit horizontally.

Scour

Currents and water motion due to waves will cause significant seabed erosion around a monopile if the seabed is formed of sand or other granular material. Based on a comprehensive set of experiments, Sumer and Fredsoe (2001) concluded that the maximum scour depth occurs under the action of sea current alone, and reaches a depth of 1.3 times the pile diameter, D , on average, with a standard deviation of 0.7D.

Scour of this magnitude would clearly have implications for both foundation stability and support structure natural frequency, so it is normal practice to provide (and maintain) scour protection in the form of rock armour to prevent it. This can be either laid on the seabed soon after installation of the monopile (static scour protection), or after a few weeks, when a scour pit will have developed to receive it (dynamic scour protection). The disadvantage of static scour protection laid over the seabed is that scour can occur at the edges, degrading the rock protection there, but this can be allowed for when the radial extent of protection is determined.

Some of the problems that can arise with scour protection are as follows: possible installation damage to transition piece coating, secondary structures or subsea cables; the need for occasional replenishment and possible increased environmental impact due to increased volumes of material imported to site. Some projects have been designed to accommodate scouring with appropriate allowance in the design for the resulting reductions in overturning resistance and support structure natural frequency.

A design aid for scour hole depth prediction and scour protection design, known as the Opti-Pile Design Tool (den Boom *et al.*, 2004), has been developed and calibrated against model test results and data from existing wind farms. This enables the size of rock that is stable under maximum current conditions to be calculated, together with the radial extent of protection required. As might be expected, the model tests reported by den Boom *et al.*

indicated that larger rock sizes were required when the scour protection was raised proud of the seabed level, as in the static protection, than when it was recessed beneath it.

Corrosion protection

The approach to corrosion protection varies over the height of the monopile according to the type of exposure. Three zones are defined – the atmospheric zone, the splash zone and the submerged zone, which includes the embedded portion. DNV-OS-J101 requires the steel monopile to be protected by a high quality, multi-layer surface coating in the atmospheric zone. In the splash zone, it requires extra plate thickness to be provided as a corrosion allowance, in addition to a surface coating. Corrosion rates in the range 0.3–0.5 mm per year may be assumed for structures in the North Sea.

Cathodic protection is to be provided in the submerged zone, together with a 2 mm corrosion allowance in the scour zone. Normally the cathodic protection is in the form of sacrificial anodes, which may weigh about 5 tonnes for the external protection of a 5 m diameter pile in 20 m water depth, using zinc-based anodes.

Corrosion protection of the internal surface of the monopile is not required, if, as is normal, the internal space is sealed off at the top of the transition piece by a diaphragm (fitted with an air-tight hatch) which has to be designed to absorb pressure differential. Complications may develop in the event of loss of sealing (for example around exit hole of an internal J tube). There is also risk of anaerobic corrosion in sulphur rich seabed environments, which has traditionally been addressed through adding biocide, although this is now less environmentally acceptable.

Monopile design

The sizing of the monopile is governed by three key factors:

- Resistance to extreme loads.
- Resistance to fatigue loads.
- Tuning of support structure natural frequency to avoid excitation by cyclic loading.

Normally extreme wave loads, being dominated by drag, increase linearly with diameter, whereas fatigue wave loads, being dominated by inertia loading, increase as the diameter squared, like the section modulus. On the other hand, the wind load component of monopile bending moment is virtually unaffected by monopile diameter so it would clearly be beneficial to increase monopile diameter and reduce wall thickness until the decline in buckling resistance stress precludes further reductions in cross sectional area. However, practical considerations such as compatibility with the tower base diameter – assuming a parallel sided transition piece – and manufacturing capability may limit the monopile diameter to a smaller value.

Wave fatigue loads depend on the support structure mode shape (see Equation 11.51), which in turn depends on the stiffness distribution, so design against fatigue loads is inevitably an iterative process. Accordingly it is simpler to develop an initial design based on resistance to extreme loads which can then be used to obtain an initial mode shape and set of fatigue loads. It should be noted that, when wave loading is a significant factor in the design, the extreme load bending moment distribution will exhibit a pronounced knee near wave crest level which may well be absent from the fatigue load bending moment diagram. This is

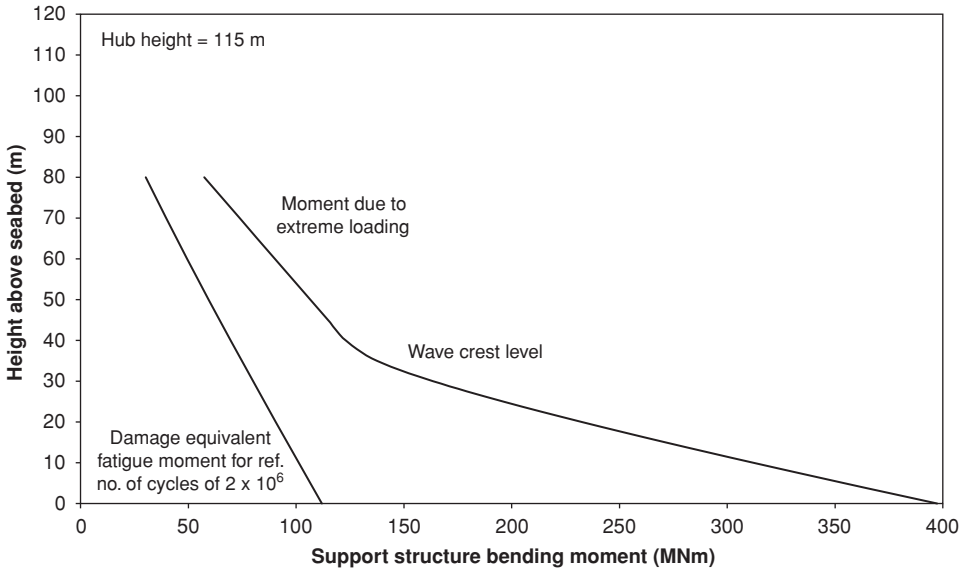


Figure 11.36 Variation of extreme and fatigue moments over height of support structure for 5 MW wind turbine

because the fatigue loads usually excite support structure resonance, for which the bending moment distribution is dominated by the tower head mass rather than the wave load itself. Consequently an onshore tower design may be unsuitable for use offshore because of wave-generated fatigue loads, even though the waves do not strike the tower itself. Example extreme and fatigue load bending moment distributions for a 5 MW turbine support structure are shown in Figure 11.36.

It is seen that extreme loads are more likely to govern the design at seabed level, whereas fatigue loads are more likely to govern at wave crest level, although, if the critical fatigue detail category is assumed to be category 90, fatigue loads govern down to seabed level in this case. In the general case, a range of factors, including the relative magnitude of extreme and fatigue loads, the critical fatigue detail category and the monopile wall slenderness, will determine whether extreme or fatigue loads govern at different heights. In practice extreme load (which could be operational given low offshore wind turbulence and gusts) tends to govern embedment depth and fatigue tends to govern other dimensions.

The other key design driver is the restriction on support structure natural frequency, and it is often best to address this early on in the process of design iteration.

Wind turbines deployed offshore are substantially larger than most of those deployed on land, and consequently nearly always operate at variable speed, because this allows power fluctuations in the wind to be absorbed by rotor acceleration rather than by rapid fluctuations of electrical power. The ratio of maximum to minimum rotational speed is usually quite wide – from 1.5 to 2.5 – with the result that the band of frequencies available for the tower natural frequency between the maximum rotational frequency and the minimum blade passing frequency is considerably narrowed. This is illustrated for a 5 MW wind turbine in Figure 11.37.

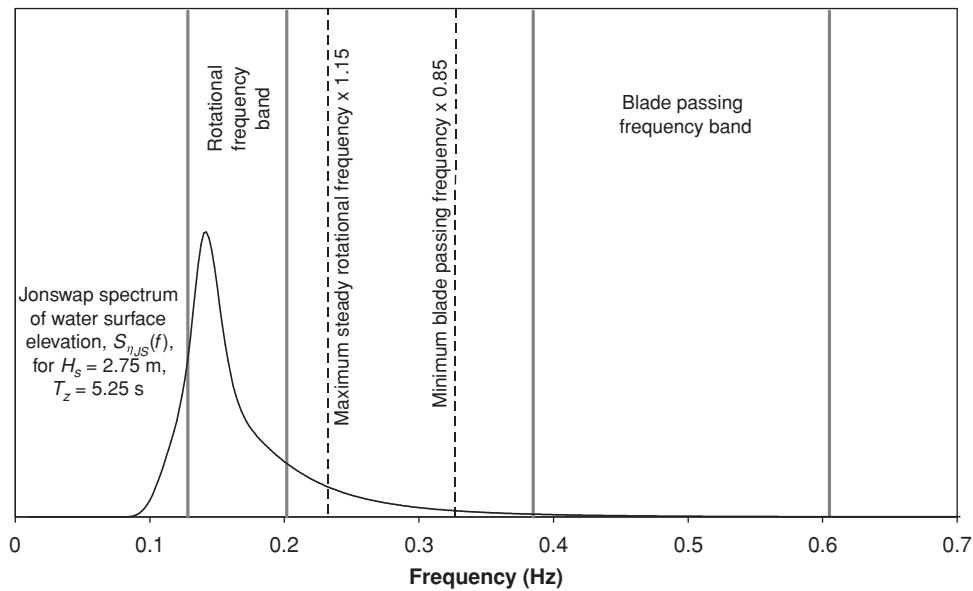


Figure 11.37 Support structure natural frequency exclusion zones for a 5 MW turbine

Given that there needs to be roughly a 15% margin between a turbine exciting frequency and the support structure natural frequency, it can be seen that in this case the range of feasible frequencies is limited to 0.23–0.33 Hz.

For a 6 m diameter monopile supporting a 5 MW turbine in water of about 20 m depth it is found that a design providing adequate strength under extreme and fatigue loading has a natural frequency just above the 0.23 Hz lower limit.

Fatigue analysis is considered further in Section 11.6.2.

Embedment

Except in the case of monopiles inserted into sockets drilled in rock, the soil in which the monopile is embedded has to be treated as a flexible medium which allows lateral movement and flexure of the pile below seabed level. This lowers the depth of effective fixity for the cantilever support structure and can have a decisive impact on natural frequency.

It is normal to analyse the interaction between pile and soil by modelling the soil as a series of non-linear lateral springs, known as *p-y* springs, acting on the pile over its embedment depth. This model was developed for determining pile ultimate resistance to lateral loads and different suites of *p-y* curves have been developed for use with clays and sands. Caution needs to be exercised when linearising the *p-y* curves in order to determine pile head stiffnesses under service loading, particularly in the case of clays, where the usual closed-form curves have infinite initial slopes. Recommended *p-y* curves and advice on their application are contained in DNV-JS-101 (2007).

Grouted joint design

The grouted joint between the monopile and transition piece must be capable of supporting the weight above and of resisting the overturning moments and torsional loads. Axial and

torsional loads produce shear at the grout/steel interfaces, whereas the overturning moment produces compression, so it is normal to treat the two types of load separately.

a) Overturning moment Moment is transferred between sleeve and pile by compression stresses in the grout, which vary approximately linearly from zero at joint mid-height to maxima at the top and bottom. Assuming the transition piece is sleeved over the monopile, the compression in the upper half of the joint is on the upwind side and the compression in the lower half of the joint is on the downwind side.

The horizontal forces between sleeve and pile generate high shear stresses in the wall of each and these approach the maximum permitted for the recommended grouted joint L/D ratio of 1.5 when the applied moment reaches the sleeve/pile moment capacity. However, the corresponding compression stresses in the grout are relatively low – of the order of 5 or 10 MPa for a fully loaded joint.

b) Axial and torsional loads Shear across the grout/steel interface due to axial and torsional loads can be resisted by the provision of shear keys in the form of weld beads or thin steel bars welded to the steel tube. However, each shear key introduces a fatigue stress raiser, which militates against their use.

In the absence of shear keys, resistance to shear at the steel/grout interfaces depends on sleeve and pile surface irregularities, which are reproduced on the mating grout surfaces. In order for relative movement between pile and sleeve to occur, the sleeve and pile must expand and contract respectively to allow a ridge in the steel to move opposite a ridge in the grout. Hence, for a depth of surface irregularity, δ , sliding generates a radial pressure of approximately $E(\delta/2R)(t/R)$ where E is the Young's Modulus of steel, R is the joint radius and t is the wall thickness of sleeve and monopile (assumed to be the same). The design shear resistance per unit area is then

$$\tau_d = \frac{\mu}{\gamma_m} E \frac{\delta}{2R} \frac{t}{R} \quad (11.45)$$

where μ is the coefficient of friction between grout and steel and γ_m is the partial safety factor for materials strength. DNV-OS-J101 recommends a maximum coefficient of friction of 0.4 and a materials factor of 3.0.

It is apparent from inspection of Equation 11.45 that the result is critically dependent on the magnitude of the surface irregularity, δ . Traditionally it was assumed that the ratio δ/R was constant, with the 2007 edition of DNV-JS-101 recommending a value of 0.00037. However, instances of joint slippage in 2009 have led to serious questioning of this assumption and the issue of interim guidance based on a fixed surface irregularity of only 0.074 mm, regardless of joint radius. This implies a dramatic reduction in δ for typical monopile diameters – for example the $\delta/R = 0.00037$ assumption would have given a δ of about 0.9 mm for a 5 m pile diameter.

Measures to increase the axial load capacity of grouted joints under consideration include the provision of elastomeric springs at the top or the introduction of a small matching taper on monopile and transition piece.

Monopiles in deeper water

Wind loads dominate monopile bending moments in shallow waters because wave loads act at a much smaller lever arm, so monopile weight would be expected to simply increase in

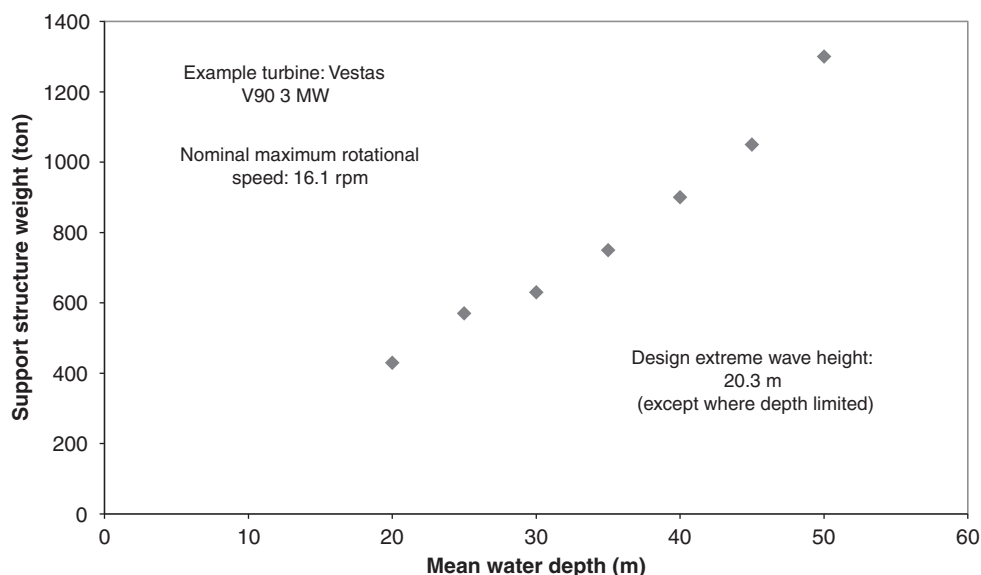


Figure 11.38 Variation of support structure weight with mean water depth. (From Van der Tempel, 2007)

proportion to monopile length at first. However, in deeper water, wave loading begins to dominate, monopile diameter has to increase and weight increases much more rapidly.

De Vries and Krolis (2007) prepared a family of outline support structures designs for the Vestas V90 3 MW wind turbine in various water depths between 20 m and 50 m, taking a design extreme wave height of 20.3 m appropriate for a North Sea site 80 km from the Dutch coast. The designs were configured to achieve a target natural frequency of 0.32 Hz and satisfy buckling strength requirements. Figure 11.38 shows the resulting variation of the weight of the support structure (comprising monopile, transition piece and tower) with water depth.

The weight for 20 m water depth is below the general trendline because, in this case, the extreme wave height is depth limited, so platform height and hub height are both reduced. To put the support structure weights in context, it can be noted that the turbine weight (rotor, hub and nacelle) for the Vestas V90 is only 112 tonnes.

The selected monopile diameters ranged from 4 m at 20 m water depth to 6 m at 50 m depth – well within the 8 m diameter capability of some existing production facilities.

11.6.2 Monopile fatigue analysis in the frequency domain

In the short term, wind and wave loading are statistically independent processes, which means that the power spectra of the loads at any cross-section of the support structure from the two sources can simply be added together to obtain the power spectrum of the total load. This section considers the treatment of wave and wind loading fatigue separately to begin with and then the combination of the two.

Wave loading in the frequency domain

On offshore wind turbine structures, the established method for calculating wave fatigue damage in the frequency domain on oil and gas platforms can be utilised. The attraction of the method is that it is computationally much more efficient than undertaking numerous simulations in the time domain followed by rain-flow cycle counting. The procedure starts with obtaining a stress spectrum for the point under consideration from the wave spectrum for each sea state by means of a transfer function, which relates the stress amplitude to the wave height at each frequency. Each stress spectrum is then converted to a stress range probability distribution (e.g. by means of the Dirlik formula, Section 5.9.3, Chapter 5), which, in combination with the appropriate $S-N$ curve, enables the fatigue damage arising from exposure to each sea state to be derived. Finally, the damages from all the sea states are summed to obtain the total fatigue damage at the point under consideration.

The method depends on a *linear* relationship between wave loading and wave height. As set out in Section 11.3.9, Morison's equation for wave loading on a cylinder contains two terms – the inertia term which is linear with respect to wave height and the drag term which is not. Fortunately, the non-linear drag term can typically be ignored when calculating fatigue loading, as inertia loading dominates for Keulegan-Carpenter numbers of less than 11 (Figure 11.22), which corresponds to a wave height to cylinder diameter ratio of less than 3.5 in deep water.

This section considers the various stages in the fatigue damage calculation in turn.

Bending moment and stress transfer functions for a monopile

a) Quasi-static behaviour For a cylindrical monopile support structure with radius r , the expression for the peak quasi-static mudline moment due to wave loading, M_0 , is obtained by integrating Morison's equation over the water depth. Hence:

$$\begin{aligned} M_0 &= C_M \rho \pi r^2 \int_{-d}^0 (\dot{u})_{\max} (z+d) dz = C_M \rho \pi r^2 \frac{H g k}{2} \int_{-d}^0 \frac{\cosh k(z+d)}{\cosh kd} (z+d) dz \\ &= C_M \rho \pi r^2 \frac{H g}{2} \tanh kd \left\{ d + \frac{1 - \cosh kd}{k \sinh kd} \right\} \end{aligned} \quad (11.46)$$

In deep water – that is, for conditions in which $d > 0.08gT^2$ – this approximates to

$$M_0 = C_M \rho \pi r^2 H g d / 2 \quad (11.47)$$

In general, M_0 is frequency dependant, as the wave number, $k (= 2\pi/L)$, depends on the wave frequency (see Equation 11.15). The transfer function giving the ratio of peak quasi-static bending moment, M_0 , to wave amplitude for different wave numbers (and hence frequencies, f) is

$$H_{M0/\eta}(f) = \frac{M_0}{H/2} = C_M \rho \pi r^2 g \tanh kd \left\{ d + \frac{1 - \cosh kd}{k \sinh kd} \right\} \quad (11.48)$$

For a monopile wall thickness of t , the bending stress is $M_0/\pi r^2 t$, so the ratio of peak bending stress to wave amplitude – the bending stress transfer function – in the absence of dynamic magnification is

$$H_{\sigma 0/\eta} = C_M \rho \frac{g}{t} \tanh kd \left\{ d + \frac{1 - \cosh kd}{k \sinh kd} \right\} \quad (11.49)$$

It is worth noting that the monopile radius does not appear explicitly in this expression, although it may affect the value of the inertia coefficient, C_M .

b) Resonant behaviour When the wave frequency is close to the natural frequency of the support structure the latter responds dynamically and its response is amplified in an analogous manner to that of a stationary blade described in Appendix A5. If, for simplicity, the seabed is assumed to be rigid, then, by analogy with Equation A5.26, the mudline moment due to wave excitation of the support structure first mode is given by

$$M_1(t) = \omega_1^2 f_1(t) \int_0^{H+d} m(y) \mu_1(y) y \, dy \quad (11.50)$$

where ω_1 is the support structure natural frequency in rad/sec, $f_1(t)$ is the first mode hub displacement, $m(y)$ is the mass distribution per unit length, $y(=z+d)$ is the height above the seabed and $\mu_1(y)$ is the first mode shape. By analogy with equation A5.3, the hub displacement is given by

$$f_1(t) = \frac{1}{k_1} \frac{\int_0^{H+d} \mu_1(y) q_1(y) dy}{\sqrt{(1 - (\omega/\omega_1)^2)^2 + 4\xi_1^2(\omega/\omega_1)^2}} \cos(\omega t + \phi) \quad (11.51)$$

where $k_1 = m_1 \omega_1^2$, $q_1(y)$ is the wave loading amplitude and ξ_1 is the first mode damping ratio. m_1 is the generalised mass, $\int_0^{H+d} m(y) \mu_1^2(y) dy$ and ω is the wave angular frequency. Hence,

$$\begin{aligned} M_1(t) &= C_M \rho \pi r^2 \frac{H g k}{2} \int_0^d \mu_1(y) \frac{\cosh ky}{\cosh kd} dy \\ &\times \frac{\int_0^{H+d} m(y) \mu_1(y) y \, dy}{m_1 \sqrt{(1 - (\omega/\omega_1)^2)^2 + 4\xi_1^2(\omega/\omega_1)^2}} \cos(\omega t + \phi) \end{aligned} \quad (11.52)$$

The generalised mass, m_1 , is usually dominated by the tower head mass, so the ratio $\int_0^{H+d} m(y) \mu_1(y) y \, dy / m_1$ approximates to the hub height above the seabed, $H+d$. Defining an effective hub height above the seabed, $H'+d$, as equal to $\int_0^{H+d} m(y) \mu_1(y) y \, dy / m_1$.

Equation 11.52 can be shortened to

$$M_1(t) = C_M \rho \pi r^2 \frac{H g k}{2} \int_0^d \mu_1(y) \frac{\cosh ky}{\cosh kd} dy \frac{H' + d}{\sqrt{(1 - (\omega/\omega_1)^2)^2 + 4\xi_1^2(\omega/\omega_1)^2}} \cos(\omega t + \phi) \quad (11.53)$$

Comparing the amplitude of the resonant moment (Equation 11.53) with the quasi-static moment (Equation 11.46), it is seen that the magnification of the mudline moment due to dynamic excitation of the first mode is

$$\begin{aligned} \frac{M_1}{M_0} &= \frac{k \int_0^d \mu_1(y) \frac{\cosh ky}{\cosh kd} dy}{\tanh kd \left\{ d + \frac{1 - \cosh kd}{k \sinh kd} \right\}} \frac{H' + d}{\sqrt{(1 - (\omega/\omega_1)^2)^2 + 4\xi_1^2(\omega/\omega_1)^2}} \\ &= \frac{\lambda(d)}{\sqrt{(1 - (f/f_1)^2)^2 + 4\xi_1^2(f/f_1)^2}} \end{aligned} \quad (11.54)$$

where

$$\lambda(d) = \frac{k \int_0^d \mu_1(y) \frac{\cosh ky}{\cosh kd} dy}{\tanh kd \left\{ d + \frac{1 - \cosh kd}{k \sinh kd} \right\}} (H' + d) \quad (11.55)$$

The ratio $\lambda(d)$ is a measure of the influence of wave loading near the root of the cantilever on first mode resonance, excluding the dynamic magnification given by the DMR, $1/\sqrt{(1 - (f/f_1)^2)^2 + 4\xi_1^2(f/f_1)^2}$. For deep water waves, the wave loading is concentrated in the region around the mean water level, so the first half of the expression for $\lambda(d)$ approximates to $\mu_1(d)/d$. Moreover, as noted above, $(H' + d)$ can be approximated by $(H + d)$. Combining these two approximations leads to

$$\lambda^*(d) = \frac{\mu_1(d)(H + d)}{d} \quad (11.56)$$

where the asterisk indicates an approximate value.

Wave loading is usually applied relatively close to the base of the support structure, so $\mu_1(d)$ is much less than the ratio $d/(H + d)$ – for example about 0.03 for a $d/(H + d)$ of 0.2 and a rigid seabed, giving a $\lambda^*(d)$ of 0.15. In practice, the seabed can rarely be considered completely rigid, and the effective pile fixity depth below the seabed, m , may be several pile diameters, leading to larger values of $\mu_1(d)$. However, in all cases, the dynamic magnification of the mudline moment due to wave loading is much less than that for the same loading applied at the tower top.

The transfer function giving the ratio of peak bending moment, M_1 , to wave amplitude for different wave frequencies, f , measured in Hz, is:

$$H_{M1/\eta}(f) = \frac{M_1}{H/2} = C_M \rho \pi r^2 g k \int_0^d \mu_1(y) \frac{\cosh ky}{\cosh kd} dy \frac{(H' + d)}{\sqrt{(1 - (f/f_1)^2)^2 + 4\xi_1^2(f/f_1)^2}} \quad (11.57)$$

It is convenient to abbreviate this to:

$$H_{M1/\eta}(f) = G_{M1/\eta}(f) \frac{1}{\sqrt{(1 - (f/f_1)^2)^2 + 4\xi_1^2(f/f_1)^2}} = G_{M1/\eta}(f) \cdot DMR \quad (11.58)$$

where

$$G_{M1/\eta}(f) = C_M \rho \pi r^2 g k \int_0^d \mu_1(y) \frac{\cosh ky}{\cosh kd} dy (H' + d) \quad (11.59)$$

Then

$$\lambda(d) = \frac{M_1/DMR}{M_0} = \frac{H_{M1/\eta}(f)/DMR}{H_{M0/\eta}(f)} = \frac{G_{M1/\eta}(f)}{H_{M0/\eta}(f)} \quad (11.60)$$

Example mudline bending moment transfer functions

Figure 11.39 illustrates the support structure for a 5 MW wind turbine considered in this example. The tower is mounted on a 6 m diameter monopile driven into the seabed. The soil properties are assumed to be such that the monopile is effectively fixed at a depth, m , of 20 m below the seabed. Hub height above the seabed is 115 m and the mean water depth, d , is 23 m.

The co-ordinate, y , defined as the distance from the cantilever root is now measured from the fixity point at a depth, m , below the seabed. This has no effect on the quasi-static mudline moment, but increases the resonant mudline moment significantly. Equation 11.52 becomes

$$M_1(t) = C_M \rho \pi r^2 \frac{H g k}{2} \int_m^{d+m} \mu_1(y) \frac{\cosh ky}{\cosh kd} dy \times \frac{\int_0^{H+d+m} m(y) \mu_1(y) (y - m) dy}{m_1 \sqrt{(1 - (\omega/\omega_1)^2)^2 + 4\xi_1^2(\omega/\omega_1)^2}} \cos(\omega t + \phi) \quad (11.61)$$

in which $\int_0^{H+d+m} m(y) \mu_1(y) (y - m) dy / m_1$ can be replaced by $(H' + d)$ as before.

Determination of the mode shape leads to a normalised deflection at mean water level, $\mu_1(d + m)$, of 0.071 and $\lambda^*(d + m) = \mu_1(d + m)(H' + d)/d = 0.071(115/23) = 0.355$.

The transfer functions for the quasi-static mudline moment, $H_{M0/\eta}(f)$, and for the first mode resonant mudline moment, $H_{M1/\eta}(f)$ together with an example Jonswap spectrum (for $H_S = 2.75$ m and $T_Z = 5.25$ sec) are shown in Figure 11.40. The support structure natural

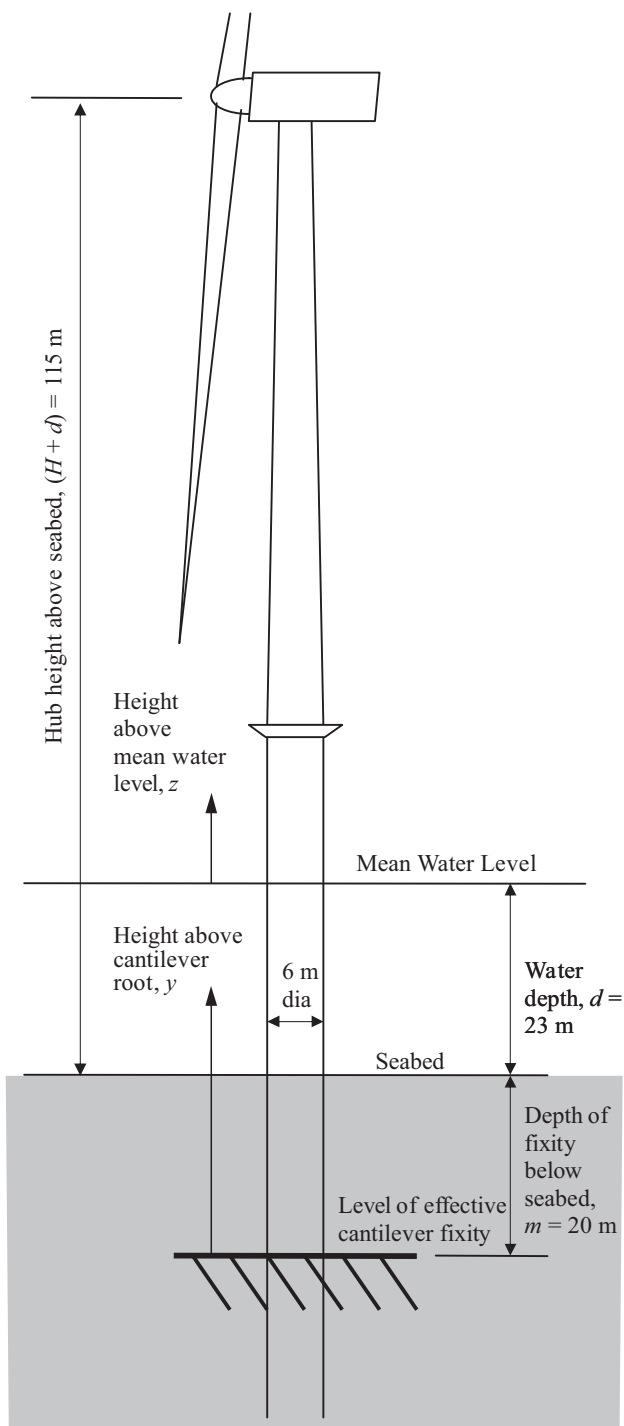


Figure 11.39 Example support structure for 5 MW turbine

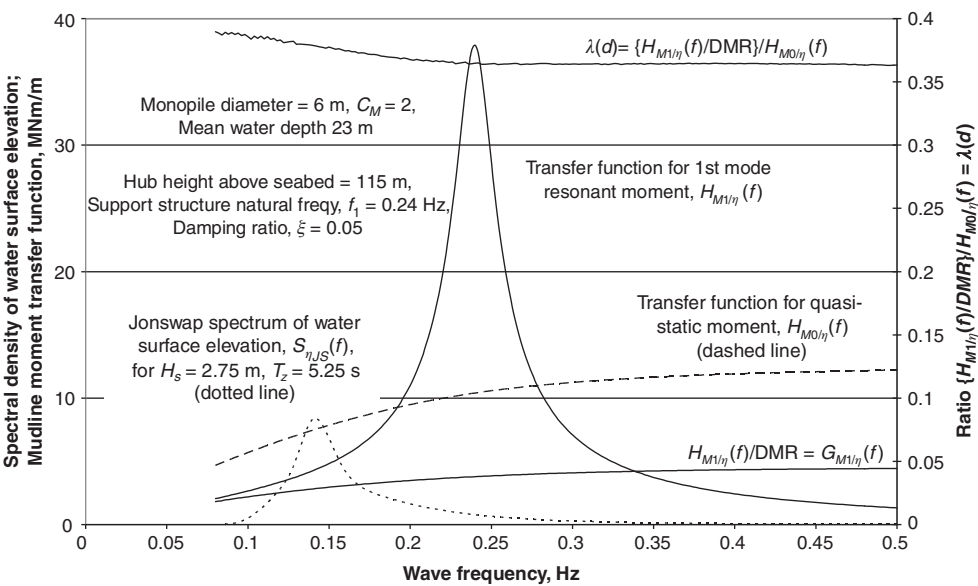


Figure 11.40 Comparison of quasi-static and resonant transfer functions for mudline moment for the example support structure of Figure 11.39

frequency, f_1 , is 0.24 Hz and the damping ratio, ξ , is taken as 5%, giving a peak dynamic magnification ratio of 10.

Clearly the damping ratio is of major importance in determining the resonance peak. The principal components of damping – viz aerodynamic, structural and that arising from soil hysteresis – are discussed in later sub-sections within this section.

It is seen that the mudline moment transfer function for quasi static loading increases with wave frequency. Thus, although water motion and, hence, wave loading are increasingly concentrated near the surface as the wave length reduces, increased particle accelerations overcome the effect, resulting in increasing overturning moments for waves of a given amplitude.

Figure 11.40 also shows the first mode resonant mudline moment transfer function divided by the dynamic magnification factor, $H_{M1/\eta}(f)/DMR = G_{M1/\eta}(f)$. The function, $G_{M1/\eta}(f)$, is very similar in form to the quasi-static transfer function. This is illustrated by the plot of the ratio of the two, $\lambda(d) = G_{M1/\eta}(f)/H_{M0/\eta}(f)$, which is seen to be insensitive to wave frequency and to be close to the $\lambda^*(d)$ value of 0.355.

Effect of diffraction

Section 11.3.9 set out how wave diffraction influences the inertia coefficient when the cylinder diameter is of significant size in relation to the wave length. Diffraction results in a rapid reduction of the inertia coefficient below the nominal value of 2.0 for D/L ratios above about 0.17, as shown in Figure 11.25. In Figure 11.40, the inertia coefficient was assumed to be constant at 2.0 for simplicity, but the effect of diffraction on the transfer function for

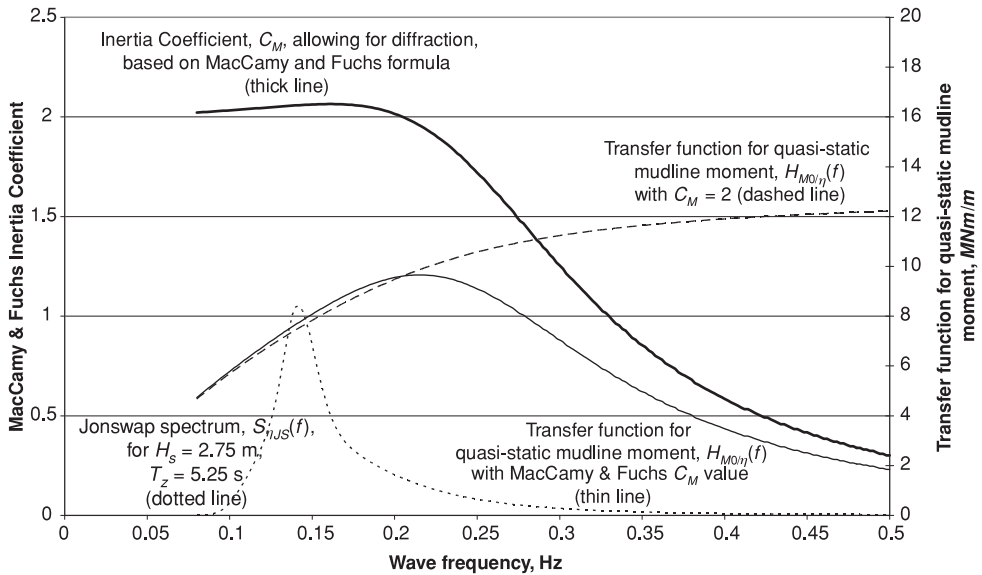


Figure 11.41 Effect of diffraction on the transfer function for quasi-static mudline moment for the example support structure of Figure 11.39

the quasi-static mudline moment is illustrated in Figure 11.41. Also shown is the Jonswap spectrum for the $H_s = 2.75$ m, $T_z = 5.25$ m sea state.

It is apparent that, for the support structure and sea state chosen, the reduction in inertia coefficient only affects the loading due to waves in the upper tail of the wave spectrum. However, as the Mac Camy and Fuchs inertia coefficient reduces to 1.8 at the support structure resonant frequency of 0.24 Hz, the resonant moment comes down somewhat when diffraction is allowed for.

Aerodynamic damping

During turbine operation, aerodynamic damping of the rotor provides significant damping of wave-induced support structure motion in the fore-aft direction. For constant blade pitch, this is easily estimated at each wind speed using the closed-form formula in Equation 5.135, but when the pitch system is active, the aerodynamic damping is heavily dependent on the control strategy and cannot be easily calculated.

Wind turbine simulations in the time domain automatically model support structure aerodynamic damping, but do so implicitly rather than explicitly. However, values of the aerodynamic damping at different wind speeds can be deduced from time domain simulations of transient free vibrations in response to a tower top step loading during turbine operation in a turbulent wind field, including the effects of blade pitching. In each case, it is necessary to subtract the tower top displacement time history from a parallel simulation without the tower top step loading, but with the same seeds, in order to determine the transient tower top displacement response to the step loading on its own. The damping ratio is then calculated from the logarithmic decrement of the first mode free vibrations.

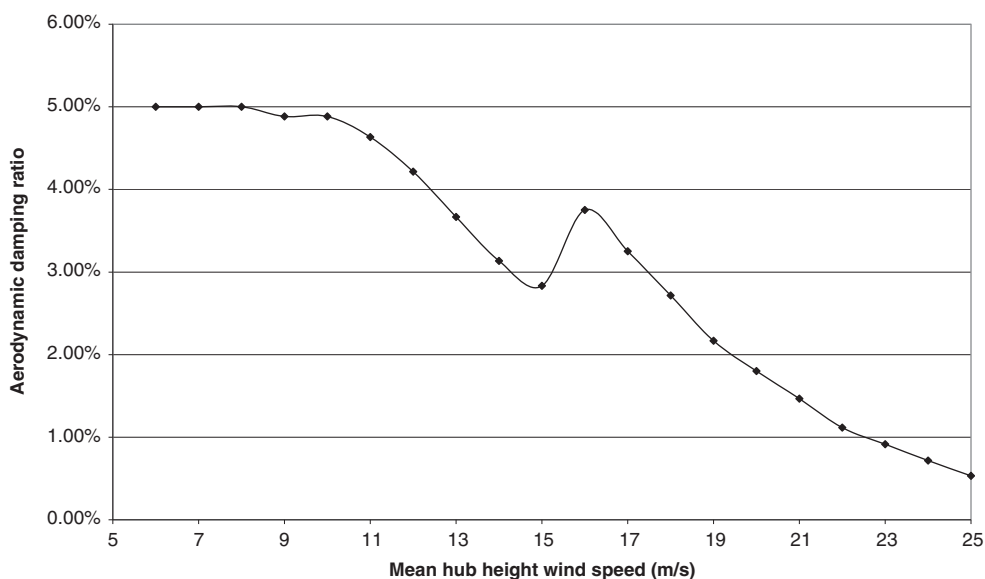


Figure 11.42 Variation of aerodynamic damping with wind speed for fixed speed pitch regulated machine in a turbulent wind

An example plot of the variation of aerodynamic damping with mean wind speed obtained using the above method by Kühn (2001) is shown in Figure 11.42. The reference turbine design investigated was a 3 MW, two-bladed, fixed speed, pitch regulated machine on a soft-soft monopile support structure, with the hub 80 m above the seabed and a water depth of 21 m, which was developed during the Opti-OWECS study.

Figure 11.42 shows reducing damping above 10 m/s as a result of increasing flow separation over the inner part of the blade until blade pitching above the rated wind speed of 13.7 m/s leads to flow re-attachment and damping recovery. However, above 16 m/s, increasingly rapid action of the pitch control system in response to tower head oscillatory motion reduces the aerodynamic damping considerably at higher wind speeds. This is because, as the tower top sways into the wind, the blades feather to curb the power increase, reducing the rotor thrust and vice versa.

It is now becoming common practice for the control system to be designed to suppress support structure oscillations at resonant frequency, which can lead to effective aerodynamic damping ratios in excess of the values for fixed pitch operation with attached flow (see Chapter 8).

Wind wave misalignment

When the wind and wave directions are not in alignment, the aerodynamic damping of wave-induced support structure motion will be reduced, as the aerodynamic damping of turbine side to side motion is very small, and this needs to be accounted for in the fatigue analysis.

Structural damping

The level of structural damping provided by the monopile and tower is very much less than that due to aerodynamic damping. For example, the Eurocode on wind loading, EN 1991-1-4,

gives a logarithmic decrement of structural damping of 0.012, equivalent to a damping ratio of 0.19% for welded steel chimneys and DNV-RP-C205 'Environmental conditions and environmental loads' gives a damping ratio of 0.15% for steel members in air.

Soil damping

Tarp-Johansen (2009) has investigated soil damping using a three-dimensional finite-element model of the seabed surrounding a monopile foundation, and concluded that the logarithmic decrement of support structure damping provided by the soil may be in the range 0.035–0.05, equivalent to a damping ratio of 0.55–0.8%. However, the pile embedment was low (4.25 diameters) resulting in larger motion in the soil and, hence, larger soil damping than would normally be the case.

In view of the difficulties of quantifying soil damping on a site-by-site basis, some designers adopt a conservative approach and simply ignore it altogether.

Damping during turbine shut-down

When the turbine is shut down, the aerodynamic damping is negligible, and support structure damping will be limited to that provided by the welded structure and the soil foundation, with a small contribution from hydrodynamic damping. The Germanischer Lloyd *Guidelines for the Certification of Offshore Wind Turbines* (2005) suggests 1% as a rough approximation to the damping ratio in these circumstances. The much reduced damping level during turbine shut-down will result in considerably greater wave excitation of support structure resonance than during turbine operation, so the percentage of machine downtime assumed in the fatigue damage calculations needs to be estimated with care.

Monopile bending moment spectra

The spectrum of the first mode resonant mudline moment is obtained by multiplying the wave spectrum by the square of the first mode resonant mudline moment transfer function:

$$S_{M1}(f) = S_{\eta JS}[H_{M1/\eta}(f)]^2 \quad (11.62)$$

The transformation of a wave spectrum into support structure bending moment spectra is illustrated in Figure 11.43 for the support structure of Figure 11.39 with the sea state modelled using the Jonswap spectrum. The sea state taken is $H_s = 2.75$ m and $T_z = 5.25$ sec, for which the peak shape factor, γ , is 2.32 and the peak frequency, f_p , is 0.141 Hz. The figure shows the Jonswap spectrum, the square of the transfer function for first mode resonant moment response and the resultant spectrum of first mode resonant mudline moment.

A complete analysis would investigate the dynamic response of the second and higher support structure modes, but excitation of these modes is minimal when inertia loading dominates.

It is evident that in this example the spectrum of the first mode resonant moment response is dominated by the resonance peak, with a much smaller peak occurring at the peak wave frequency. The latter peak is much smaller than the peak of the quasi-static spectrum (Figure 11.44), but good agreement would be expected if the spectra of mudline moment in response to the excitation of higher support structure vibration modes were added to that shown above for the first mode.

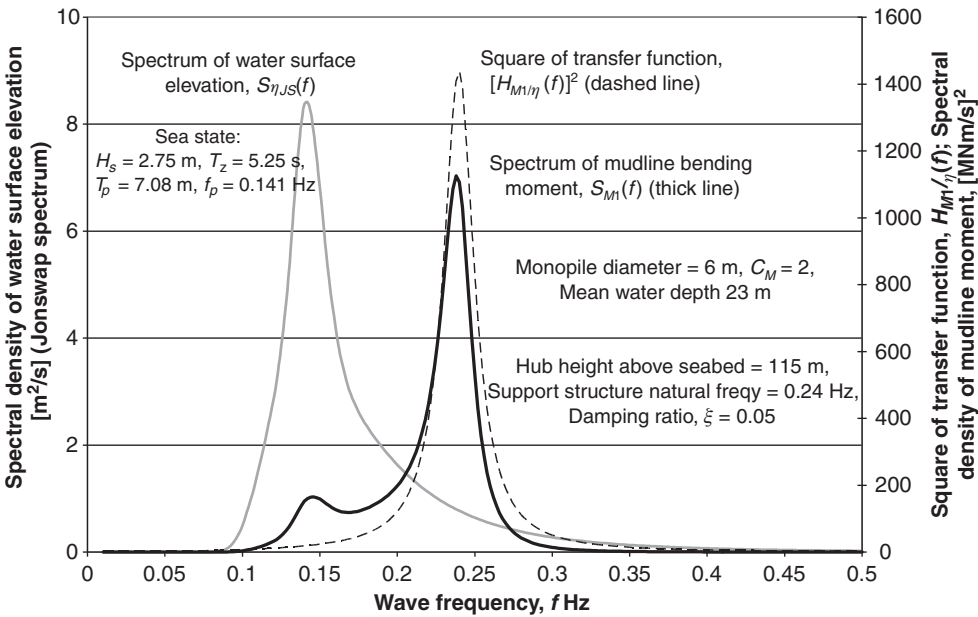


Figure 11.43 Spectra of water surface elevation and resonant mudline bending moment for the example support structure of Figure 11.39 and the $H_s = 2.75$ m/ $T_z = 5.25$ sec sea state

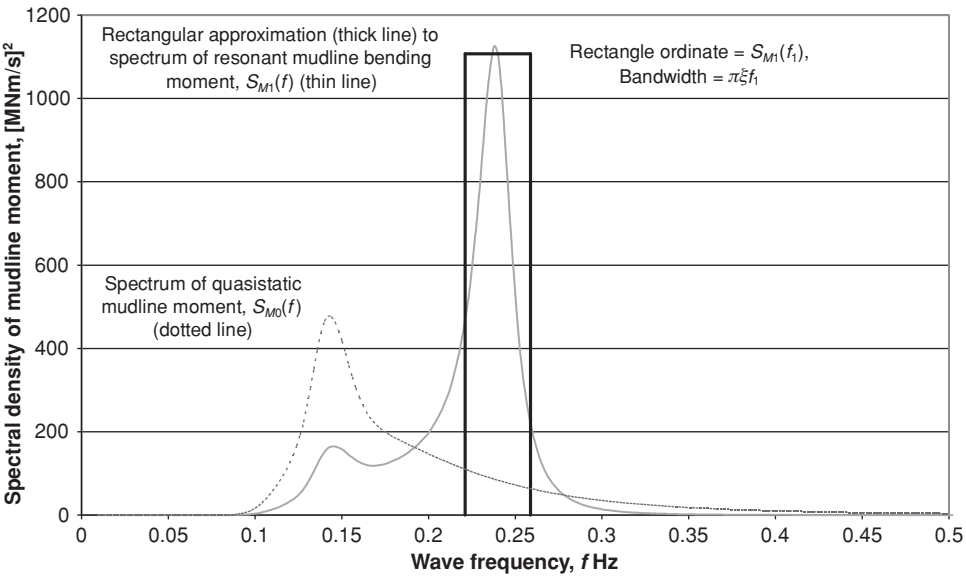


Figure 11.44 Mudline moment spectrum approximation

Approximate treatment of moment response for fatigue analysis purposes

Given that the first mode resonant response is narrow banded, the resonant moment spectrum may be approximated for fatigue analysis purposes by a rectangle of height $S_{M1}(f_1)$ and width (the bandwidth) $\pi \xi f_1$, centred on the first mode resonant frequency, f_1 Hz. This is illustrated in Figure 11.44. The area within the rectangle is

$$S_{M1}(f_1)\pi \xi f_1 = S_{\eta JS}(f_1)[H_{M1/\eta}(f_1)]^2\pi \xi f_1 \quad (11.63)$$

As the transfer function is proportional to the dynamic magnification ratio (DMR), which is equal to $1/(2\xi)$ at resonance, it follows that the area within the rectangle is inversely proportional to the damping ratio. The standard deviation of the resonant bending moment, which is equal to the square root of the area of the rectangle, is thus given by

$$\begin{aligned} \sigma_{M1} &= H_{M1/\eta}(f_1)\sqrt{S_{\eta}(f_1)\pi \xi f_1} \\ &= C_M \rho \pi r^2 g k \int_m^{d+m} \mu_1(y) \frac{\cosh ky}{\cosh kd} dy (H' + d) \sqrt{S_{\eta}(f_1) f_1 \pi / (4\xi)} \end{aligned} \quad (11.64)$$

Given that the area of overlap between the rectangular approximation to the resonant bending spectrum and the quasi-static spectrum is small, the total moment spectrum can be conservatively approximated by the summation of the two.

A further simplification is to ignore the variation of the quasi-static transfer function with frequency, and equate the quasi-static moment spectrum to the product of the wave spectrum and the square of the quasi-static transfer function at the peak wave frequency. Then

$$S_{M0}(f) = S_{\eta JS}(f)[H_{M0/\eta}(f_p)]^2 \quad (11.65)$$

and the standard deviation of the quasi-static bending moment becomes simply

$$\sigma_{M0} = \sigma_{\eta} H_{M0/\eta}(f_p) = \frac{H_s}{4} C_M \rho \pi r^2 g \tanh k_p d \left\{ d + \frac{1 - \cosh k_p d}{k_p \sinh k_p d} \right\} \quad (11.66)$$

where σ_{η} is the standard deviation of water surface elevation and k_d is the wave number corresponding to the peak frequency, f_p . Finally the standard deviation of the total mudline moment response is obtained as the square root of the sum of the squares of the quasi-static and resonant responses:

$$\sigma_M^2 = \sigma_{M0}^2 + \sigma_{M1}^2 \quad (11.67)$$

Derivation of fatigue damage

The method of calculating the fatigue damage from the moment spectrum depends on whether it can be treated as narrow-band (i.e. occurring over a narrow range of frequencies) or not. When the resonant moment dominates, the quasi-static response centred on the peak wave frequency will have little influence and the response can be treated as narrow banded, with all the load cycles occurring at the resonant frequency. It is worth noting that, provided the standard deviation of the quasi-static response is included using equation (11.66), the

narrow band assumption is conservative, because allowance for broad-bandedness *reduces* the calculated fatigue damage.

It can be shown (see Barltrop and Adams, 1991) that the damage equivalent stress range for a reference number of cycles, N_R , for narrow banded oscillatory loading is given by

$$\Delta\sigma_{eq} = \sigma_\sigma \sqrt[8]{8^m \frac{n}{N_R} \Gamma\left(\frac{2+m}{2}\right)} \quad (11.68)$$

where σ_σ is the standard deviation of the fluctuating stress ($= \sigma_M / \pi r^2 t$), m is the inverse slope of the log-log plot of the S - N curve (assumed to be a single straight line, rather than bi-linear) and n is the number of load cycles over the duration of the sea-state. For $m = 4$ and $n = N_R$, $\Delta\sigma_{eq} = 3.363\sigma_\sigma$.

If, on the other hand, the total moment spectrum is broad-banded, then the Dirlik method (Section 5.9.3, Chapter 5) can be used to derive an approximate stress range probability distribution, which is then combined with the S - N curve to obtain the fatigue damage for each sea state.

Derivation of S - N curve

The derivation of the S - N curve to be used for the damage calculation must take account of the fatigue detail, corrosion, the plate thickness and the partial safety factor for materials strength.

Influence of fatigue detail Unless there are J tube penetrations through the monopile wall or welded attachments, the critical fatigue details are likely to be the transverse butt welds. According to EN 1993-1-9 'Design of steel structures – fatigue', transverse butt welds are detail category 90, provided the weld convexity is less than 10%, or 112 if the welds are ground flush. For weld convexity between 10% and 20%, detail category 80 applies.

ISO 19902:2007 'Petroleum and natural gas industries –fixed steel offshore structures' uses a different set of S - N curves, which are identified by different letters of the alphabet and have their origins in the British Standard for steel bridges, BS 153. In the case of transverse butt welds, the applicable S - N curves are curve C, D and E for ground flush welds, welds with low convexity and welds with high convexity respectively. The curves for low weld convexity, ISO 19902 curve D and the EN 1993-1-9 curve for detail category 90 parallel each other fairly closely before the cut-off limit, as shown in Figure 11.45.

Influence of corrosion Fatigue testing of specimens with cathodic protection in seawater indicates reduced fatigue strength for stress range magnitudes corresponding to fatigue lives of less than 10^6 cycles, and this is reflected in separate S - N curves for structures in seawater 'with adequate corrosion protection' specified in ISO 19902. The curve D variant for seawater is shown in Figure 11.45.

Influence of plate thickness EN 1993-1-9 requires the fatigue strength to be reduced in the ratio $(25/t)^{0.2}$ when the plate thickness, t , exceeds 25 mm. Monopile wall thicknesses often reach 80 mm, giving a ratio of 0.792, so this is a significant consideration. The ISO 19902 reduction ratio formula, $(16/t)^{0.25}$, is more conservative.

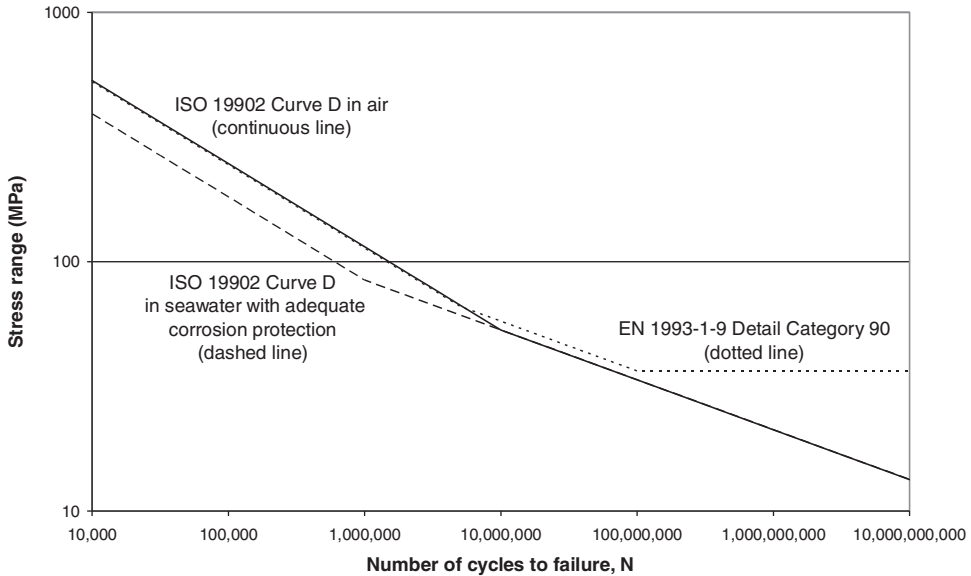


Figure 11.45 Comparison of design S - N curves for transverse butt welds without excessive reinforcement

Partial safety factor for material strength It is generally impracticable to inspect the monopile for fatigue damage, because of marine growth and the hazardous environment, so it must be designed for ‘safe-life’. On the basis that the consequence of monopile failure would be high, the partial factor for material strength, γ_m , recommended by EN 1993–1–9 is 1.35. This is higher than the corresponding value of 1.265 in IEC 61400–1, which is the product of a materials factor of 1.1 in the absence of periodic inspections and a consequence of failure factor of 1.15 for high consequence.

In ISO 19902, the partial safety factor for material strength is taken as unity and allowance made for the absence of periodic inspections and high consequence of failure by inflating the calculated fatigue damage sum (which is required to be less than unity as usual) by a fatigue damage design factor of 10. This is equivalent to a rather high partial safety factor of material strength of 1.58 for an S - N curve inverse slope of $m = 5$, which reflects the higher safety criticality of oil and gas structures.

DNV-OS-J101 ‘Design of offshore wind turbine structures’ adopts a lower value of fatigue damage design factor of three for uninspectable parts of the structure considered to have low consequence of failure. This corresponds to a γ_m of 1.25 for $m = 5$. However, no guidance is given for structural elements for which the consequence of failure is high.

Wind loading in the frequency domain

Dynamic and fatigue analysis of onshore wind turbine support structures in the frequency domain are discussed in Chapter 5, Sections 5.12.5 and 5.12.6 respectively, and similar considerations apply to offshore support structures under wind loading. Given the statistical independence of wind and wave loading, the moment spectrum due to wind loading is simply

added to the moment spectrum due to wave loading to obtain the moment spectrum due to the combined loading.

Wind loading in the time domain

As noted at the end of the preceding section, time domain simulations are preferred to frequency domain analysis for the evaluation of fatigue damage due to wind loading, because they account for both non-linear aerodynamics, non-linear control system behaviour and the combination of deterministic and stochastic load components. Methods of combining wind loading fatigue cycles obtained in the time domain with wave loading fatigue cycles obtained in the frequency domain are discussed below.

Combination of wind and wave loading fatigue spectra

a) Using simulations and rainflow counting for wind loading fatigue Kuhn (2001) has proposed calculating the fatigue damage due to wind and wave loading separately, using normal wind simulation and spectral fatigue analysis respectively, and then combining the two. The procedure is shown in the schematic below (Figure 11.46).

The damage equivalent stress range referred to in the schematic below is defined as the stress range which, when applied over a specified number of cycles, N_R , would produce the same fatigue damage as the actual calculated fatigue stress spectrum. N_R is normally taken as 10^7 cycles.

A critical step in the procedure set out in Figure 11.46 is the combination of the wind and wave damage equivalent stress ranges, and the basis for this is set out below.

It has already been noted that the damage equivalent stress range for a reference number of cycles, N_R , for narrow banded oscillatory loading is given by

$$\Delta\sigma_{eq} = \sigma_\sigma \sqrt[8]{\frac{n}{N_R} \Gamma\left(\frac{2+m}{2}\right)} \quad (11.69)$$

Hancock and Gall (1985) have proposed an empirically based modification to the above formula for the case of broad banded loading, which multiplies it by the m -th root of β , the ratio of the mean period between crests and the zero crossing period, T_c/T_z . Thus,

$$\Delta\sigma_{eq} = \sigma_\sigma \sqrt[8]{\frac{T_c}{T_z} \frac{n}{N_R} \Gamma\left(\frac{2+m}{2}\right)} \quad (11.70)$$

The variance of the combined response, $\sigma_{\sigma,c}^2$ is equal to the sum of the variances of the responses to aerodynamic and hydrodynamic loading, $\sigma_{\sigma,a}^2 + \sigma_{\sigma,h}^2$. Hence, repeated application of Equation 11.70 yields:

$$\Delta\sigma_{eq,c} = \sqrt{\Delta\sigma_{eq,a}^2 \left(\frac{T_{z,a}}{T_{z,c}}\right)^{\frac{2}{m}} + \Delta\sigma_{eq,h}^2 \left(\frac{T_{z,h}}{T_{z,c}}\right)^{\frac{2}{m}}} \quad (11.71)$$

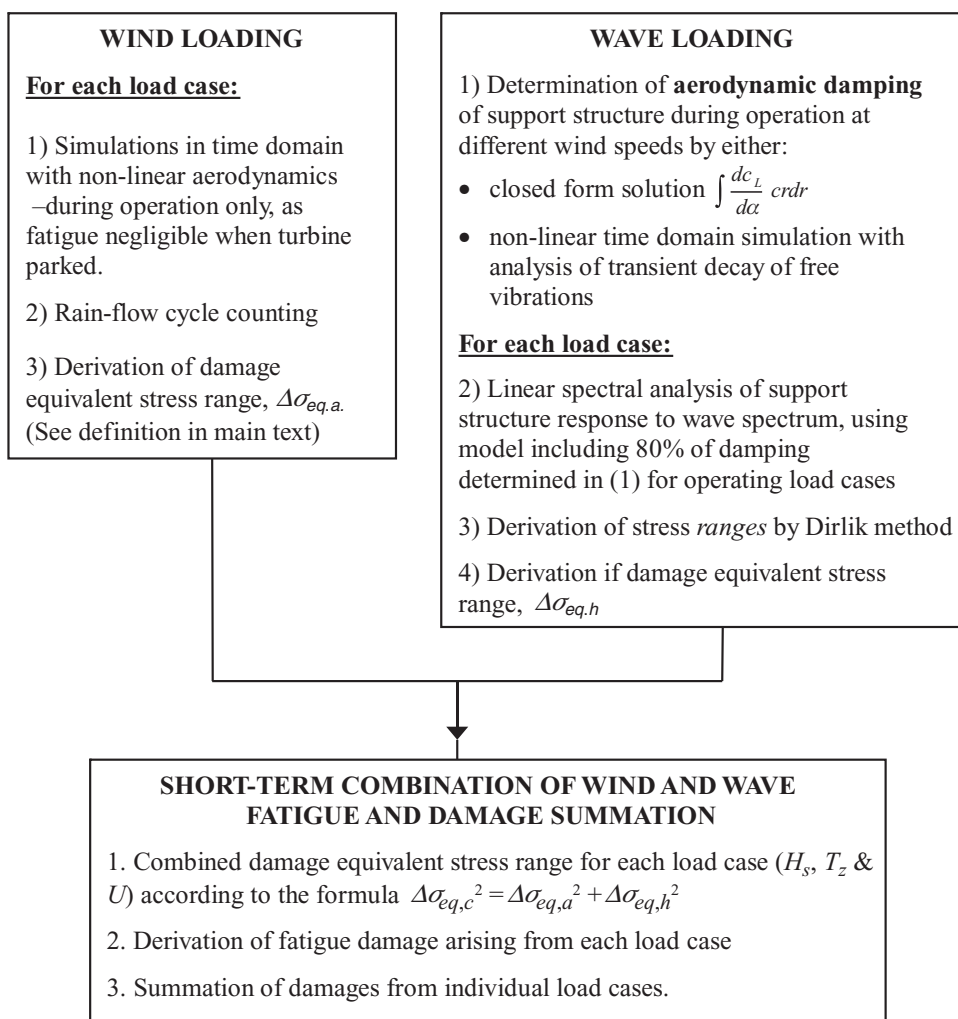


Figure 11.46 Schematic for simplified calculation of fatigue damage

If the zero crossing period weightings are omitted, this becomes simply:

$$\Delta\sigma_{eq,c} = \sqrt{\Delta\sigma_{eq,a}^2 + \Delta\sigma_{eq,h}^2} \quad (11.72)$$

Kuhn (2001) reports an investigation of the accuracy of the above simplified method, when applied to the calculation of fatigue damage at cross-sections 5.5 m below the mudline and 10.6 m below the tower top flange for the soft-soft monopile support structure of the reference design referred to above. Equivalent stress ranges obtained from full time-domain simulations of combined wind and wave loading were compared with those derived from the simplified method, using either weighted quadratic superposition, Equation 11.71, or its un-weighted equivalent, Equation 11.72, to combine the equivalent stresses from aerodynamic and

hydrodynamic loading. Both methods yielded accurate results, but, surprisingly, unweighted quadratic superposition was more accurate, with a maximum error in the equivalent stress of about 3%, compared with about 5% for weighted quadratic superposition.

b) Using simulations to obtain bending stress spectra for wind loading An alternative approach has been proposed by Van Der Tempel (2006), in which time domain simulations are used to obtain bending stress spectra due to wind loading. The key steps are as follows:

1. Carry out a time domain simulation at each mean wind speed with the tower top *fixed*, and derive a spectrum of the resulting fluctuating load applied to the tower top from its time history.
2. Divide the latter spectrum by the turbulence spectrum of the incoming wind, to obtain the transfer function between the incoming wind fluctuations and the tower top load fluctuations.
3. Obtain values of the aerodynamic damping at different wind speeds – for example, by means of time domain simulations of transient free vibrations in response to a tower top step loading (as described under the aerodynamic damping heading above).
4. Use an FE model of the support structure to derive the transfer functions between tower top loading and the required support structure bending stress for the different values of aerodynamic damping.
5. Multiply the transfer functions derived in steps 2 and 4 together to obtain the transfer function between incoming wind speed and bending stress fluctuations $H_{\sigma/U}(f, \bar{U})$ at each mean wind speed.
6. Multiply the $H_{\sigma/U}(f, \bar{U})$ transfer functions by the spectrum of the incoming wind fluctuations to obtain the spectrum of bending stress fluctuations due to wind loading at each mean wind speed.

An advantage of this approach is that the use of time domain simulations with the tower top fixed means that the results are independent of the support structure used, so do not need to be repeated for different water depths and soil conditions.

Spectra of bending stress due to combined wind and wave loading are obtained by adding the wind loading spectrum to the wave loading spectrum for each combination of mean wind speed and sea state. These combined bending stress spectra can then be used to derive the stress range probability distribution using the Dirlik method and hence the fatigue damage.

11.6.3 Gravity bases

The structural action of a gravity base supporting an offshore wind turbine is essentially the same as that of slab foundations used on land (Section 7.10.1, Chapter 7) – that is, it relies on the restoring moment provided by support structure and turbine self weight to resist overturning moments tending to tip it over. However, gravity bases deployed at sea, which are also invariably constructed in reinforced concrete (although steel or steel/concrete composite structures are feasible), are less effective in resisting overturning than those on land, because buoyancy forces reduce their effective self weight. Consequently extra ballast usually has to be added to compensate. It should also be noted that, in the case of bases founded on granular

materials, the bearing resistance is less than for the equivalent unsubmerged material on land, because of the reduced particle effective stresses.

Foundation slabs for onshore turbines can be poured *in situ*, but this option is not available at sea. Instead, the gravity base for an offshore turbine has to be constructed at a quayside, transported by barge to the wind farm site and lowered onto the seabed, which has to be prepared to receive it. Crane lifting capacity places a limit on the weight of gravity base that can be transported, so it is usual to design the structure to be hollow and fill the voids with ballast material once it is *in situ*.

Floating manufacture, towed transport and installation through controlled submersion are possible (proven in other offshore sectors) and proposals for such concepts exist. The driver for these is efficiency of fabrication and reducing delay risk and cost of installation.

Gravity bases are significantly more massive than alternative concepts so in some situations environmental impact presents a barrier to their use.

Gravity bases in the Southern Baltic

The relatively benign wave climate and negligible tidal range in the Southern Baltic render seabed preparation and gravity base transportation less challenging activities than in the more hostile conditions prevalent in the North Sea, so gravity bases have been favoured for deployment in shallow water sites close to the coasts of Denmark and Sweden.

Figure 11.47 illustrates the gravity base foundation design used to support 2.3 MW Siemens turbines, 48 in number, at the Lillgrund wind farm in Swedish waters just south of the Öresund bridge. (This information was obtained from the Lillgrund Wind Farm, owned

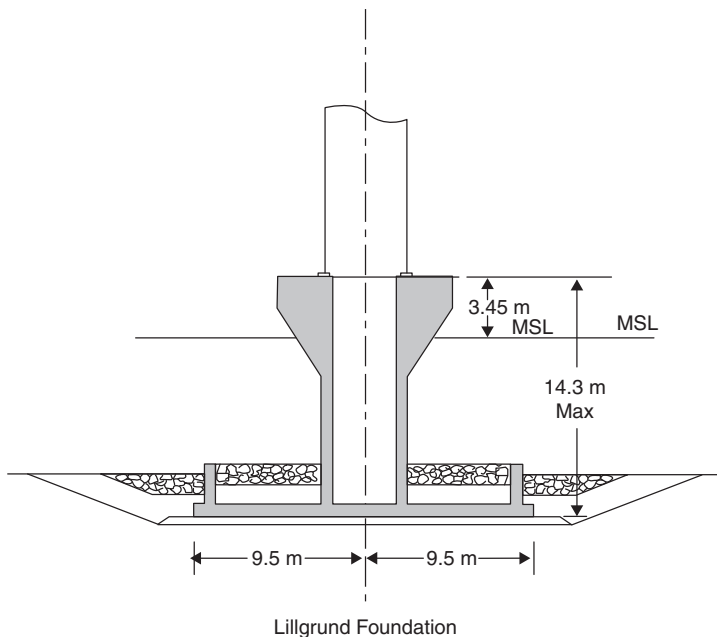


Figure 11.47 Reinforced concrete gravity base design used at Lillgrund wind farm, Sweden



Figure 11.48 Gravity bases for Lillgrund under construction on barge at quayside at Swinoujscie. (Lillgrund Pilot Project, 2008). Reproduced by permission of Vattenfall. See Plate 4 for the colour figure

and operated by Vattenfall.) The base consists of a hollow cylindrical shaft rising from a 16.45 m wide hexagonal slab (19 m across corners), which is stiffened by six radial walls bracing the shaft to an outer perimeter wall some 2.4 m high. The six-cell ballast tray thus created was backfilled with ballast rock and rock armour after foundation installation. Five variants of the design were constructed with overall heights of 10.3 m, 11.3 m, 12.3 m, 13.3 m and 14.3 m to suit water depths ranging from 4–9 m. (The large inverted cone integral with the top of the cylindrical shaft is provided to reduce ice loads.) The foundations were constructed on barges moored at a quayside at Swinoujscie on the coast of Poland, with each barge accommodating four foundations (Figure 11.48), facilitating eventual transport to the site.

The site preparation work consisted of the excavation of a shallow pit in the seabed about 2.5 m in depth, the placing of a steel frame around the perimeter of the excavation, followed by the deposition of a thin layer of crushed rock for the gravity base to rest on. The steel frame was set to level and used by divers to level the rock bed with the aid of a rotating beam mounted on a central support. After installation of each foundation by barge mounted crane (Figure 11.49), the excavation around the foundation was backfilled with filter rock over which rock armour was laid as scour protection. Finally, the void in the cylindrical shaft was filled with ballast material and a concrete slab cast above it. The weight of the tallest foundation was 1375 tonnes without ballast, increasing to 2250 tonnes with ballast.



Figure 11.49 Lowering of gravity base by floating crane during installation at Lillgrund wind farm. (Lillgrund Pilot Project, 2008). Reproduced by permission of Vattenfall. See Plate 5 for the colour figure

Similar gravity base designs were used at the other large Baltic wind farms, Rodsand 1 (Nysted) and Rodsand 2, where Siemens 2.3 MW turbines were also deployed. However, in the case of Rodsand 2, the base slab is an octagon with four short sides and four long ones, and the internal walls are arranged in a cruciform. At £350 k per MW installed, the cost of the Rodsand foundations, installed in 2010, is claimed to be half that of UK monopile foundations.

Gravity bases in the North Sea off Belgium

Until 2008, gravity base foundations had been conspicuous by their absence in the North Sea – presumably for the reasons stated above. However, in that year, six very large gravity bases were installed in deep water on Thornton Bank, 30 km off the Belgian coast, to support REpower 5 MW turbines (see Figure 11.50).

In the planning stages of the project, both monopile and gravity base foundations were considered in detail, but the large size of monopile required in the 23 m water depth, the escalation of steel costs and driveability concerns led to the selection of a gravity base. A full description of the gravity base design and its installation is given in Peire *et al.* (2009).

The gravity base configuration adopted for the deep water at Thornton Bank differs considerably from the shallow water design of the Southern Baltic. In place of a shaft and cantilevered slab, the shaft is mounted on a frustum, which provides an efficient load path to

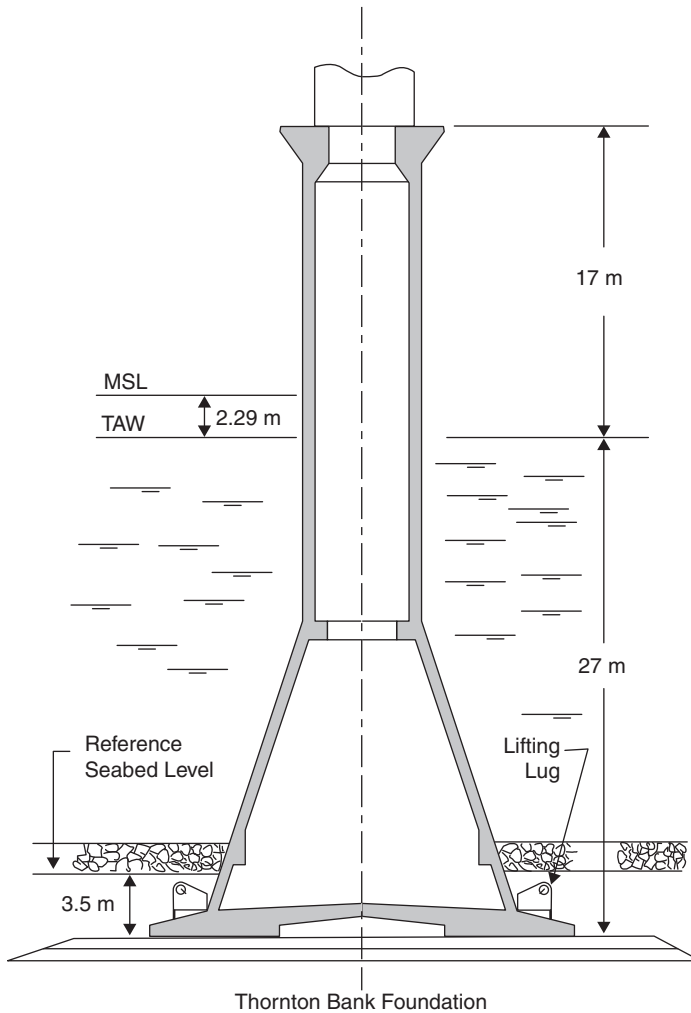


Figure 11.50 Pre-stressed concrete gravity base design used at Thornton Bank wind farm, Belgium

the base slab perimeter, without increased exposure to wave loads near the water line. As usual the weight of the unballasted structure is critical, so the shaft and frustum are pre-stressed to utilise the concrete more efficiently.

The key dimensions are as follows:

- Height (dependent on local sea bed depth): 44 m maximum.
- Shaft diameter: 6.5 m.
- Base slab diameter: 23.5 m.
- Frustum height and base diameter: 17 m.
- Wall thickness of shaft and frustum: 0.5 m.

The gravity bases are founded at a level 3.5 m below the reference seabed level, which is defined as the lowest seabed level that can be guaranteed over 30 years, after allowing for sand wave mobility and natural erosion. The top of the gravity base, which incorporates the access platform, is 17 m above TAW, the Belgian reference level for sea levels, which is 2.29 m below mean sea level.

The bases were constructed on parallel concrete beams about 1.2 m in height, cast on a quay at Ostend. On completion of each base, self propelled multi-axle modular trailers were positioned in the spaces between the concrete beams and their hydraulic suspensions extended to raise the base off the concrete beams. The trailers then transported the base very slowly to the edge of the quay, where the floating crane, Rambiz, was able to lift it clear, using a purpose built lifting frame and four large lifting lugs mounted on the base slab. After the voyage to Thornton Bank, the Rambiz lowered each base into position on a prepared bed and the hoisting cables were released, using a special hydraulically-operated pin release mechanism.

Extensive works on the seabed were necessary at each site in advance of gravity base installation. Foundation pits 50 m wide by 80 m long were dredged to a depth of about 7 m, with the longer dimension aligned NE/SW, parallel to the prevailing current direction. The final stage of dredging over the final metre depth had to be carried out in relatively calm conditions, in order to achieve a reasonably level surface. The 1.3 m thick foundation bed of gravel – consisting of a finer, filter layer followed by a coarser layer – was then deposited by the ‘Seahorse’, a dynamically positioned vessel with 18,000 tonne capacity, working in conjunction with a Remotely Operated Vehicle on the seabed equipped with surveying equipment. Finally, the surface was accurately levelled by dragging a purpose-built levelling tool across it. As a result, the inclination of the installed structures to the vertical was remarkably low – 0.1° on average.

Following installation, the pits surrounding each gravity base were backfilled with sand retrieved from the areas where the original dredgings had been deposited and the interior of the base was ballasted with the same material. Finally, rock armour was deposited around each base as scour protection.

The weight of the 44 m tall foundation without ballast was 3000 tonnes, and the volume of sand used as ballast was approximately 2000 cubic m.

11.6.4 Jacket structures

Jacket structures offer a lighter alternative to either monopiles or gravity bases in deeper waters. The basic design concept, which consists of several near-vertical tubular legs braced together by smaller tubes in each face, originates in the oil and gas industry, but is undergoing a transformation in response to the more demanding economics of the wind industry.

Figure 11.51 illustrates the arrangement of the structures with four bays of diagonal bracing installed in 2009 at the Alpha-Ventus wind farm in the North Sea. The site is located in waters 30 m deep, 45 km north of the German island of Borkum, which is close to the Dutch border. The concept was first trialled at the Beatrice Demonstrator Project in 2006, when two taller, five bay structures were deployed in waters 45 m deep in the Moray Firth, Scotland. In 2010, the installation of 30 smaller, three bay versions, but otherwise of similar design, began in waters 17–21 m deep in the Irish Sea at the Ormonde project. The structures support REpower 5 MW wind turbines at all three sites.

Jacket structures can yield considerable weight savings compared with monopiles because the primary loading – the overturning moment – is resisted at much higher lever arms, thanks

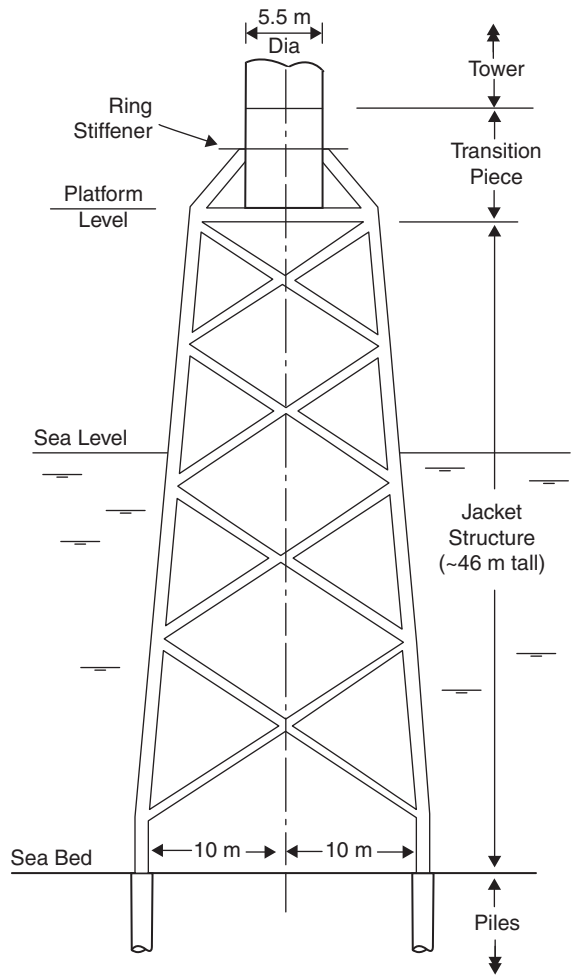


Figure 11.51 Jacket structure to support REpower 5 MW turbine at Alpha Ventus wind farm. (Access ladder, platform handrailing and J tubes omitted for clarity)

to the wide separation of the legs. In addition wave inertia loading on the structure, which is critical for fatigue damage, is smaller, because of the reduced total member cross-sectional area. On the other hand, the wave drag loading, which governs extreme loading, is increased, because of the increased total member width.

On the debit side, the fabrication and welding of the many geometrically complex joints between the bracing members and the legs carry a considerable cost penalty, even before consideration of the extra material requirements arising from the higher stress concentrations and inferior performance of the weld details in fatigue. (See section 7.9.3, Chapter 7, for discussion of fatigue design in relation to tubular towers.) In addition, the step down in width at the top of the jacket from the widely spaced legs to the tower diameter necessitates the provision of a substantial transition section. This consists of a cylindrical tower stub

member supported on raking extensions to each jacket leg, horizontal members along the sides of a square connecting the legs together and a platform at the same level integral with the base of the tower stub section. Deformation of the wall of the tower stub section by the horizontal component of the raking jacket leg extensions is prevented by a large horizontal ring stiffener in the projects mentioned above. In the case of the Beatrice support structures, the transition section weighed 163 tonnes – almost half the 360 tonne weight of the jacket itself.

An alternative approach to joint fabrication is the use of cast nodes, which can be welded to the tubes by straightforward circumferential butt welds. Weserwind (Seidel, 2007a) have developed standard cast nodes for the three types of joints in jacket structures:

- X-node for each intersection of diagonals in the faces.
- Double K node for joining the diagonals in two faces to a leg member.
- Double Y node for connecting the top of each leg to two diagonals and to the tower transition section.

In their prototype jacket design, the legs are much closer together at the top, so that they align with the tower transition section, which is a shell of revolution forming a frustum. Structurally this is a much more efficient transition arrangement, but obviously it requires significantly heavier leg members.

Piling

Jacket structures have to be attached to the sea bed in some way at each corner, in order to resist uplift generated by overturning moments. To date this has been accomplished by piling, although the use of suction buckets is under investigation.

The Beatrice jacket structures were fabricated with a pile sleeve attached to each leg. Following jacket installation, a 1.83 m diameter pile was threaded through each sleeve and driven into the seabed. Each pile and sleeve were then swaged together by expanding the pile under hydraulic pressure, deforming it into circumferential recesses in the sleeve.

A different approach was adopted at Alpha Ventus, where the piles were driven in advance. Instead of using pile sleeves, jacket leg extensions were fitted below the jacket feet and, at jacket installation, these were threaded into the piles (from which the upper part of the soil plug had previously been removed) as the jacket was lowered. With the jacket feet finally resting on the pile tops, the void between the each jacket leg extension (known as the jacket ‘Stab-in’) and its pile was grouted up to complete the anchoring of the jacket to the seabed. The arrangement is similar to that employed at the Ormonde wind farm, which is illustrated in Figure 11.52.

A critical element in this procedure was the accurate positioning of the piles in the first place, which was achieved by means of a two-part steel template laid on the seabed (Norwind, 2010). The outer element was positioned successively at four positions around a central octagonal element, kept in place by temporary piles. The outer element incorporated a pile guide and, at each position, a fresh pile was inserted and driven home.

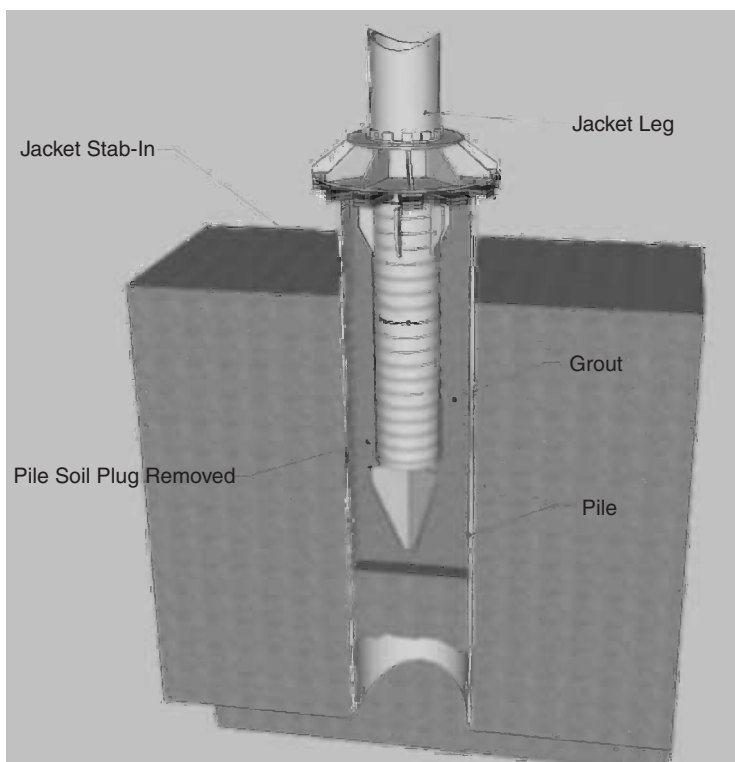


Figure 11.52 Anchorage of jacket leg to pile using concentric jacket stab-in and grouted joint – Ormonde wind farm. Reproduced by permission of Offshore Design Engineering Ltd (ode). See Plate 6 for the colour figure

Installation methods

Large floating cranes were used to place the jacket structures on the seabed at both Beatrice and Alpha Ventus wind farms. The floating crane was also used to install the pre-assembled rotor, nacelle and tower on each jacket structure at Beatrice (Figure 11.53).

The figure shows the elaborate octagonal temporary interface frames which were attached to the jacket top and tower base to facilitate the mating process. These incorporated hydraulic rams, allowing them to compensate for the relative motion of the floating crane and the jacket structure due to swell and, thus, avoid hard impact. By contrast, a jack-up barge mounted crane was used for the separate installation of tower, nacelle and rotor in the shallower waters at Alpha Ventus.

Jacket weights

The approximate weights of the jacket structure, secondary elements and piles at Beatrice and Alpha Ventus are given in Table 11.6.

It is apparent that the pile weight accounts for over one third of the total in each case and that at Beatrice the transition section approaches half the weight of the jacket structure.

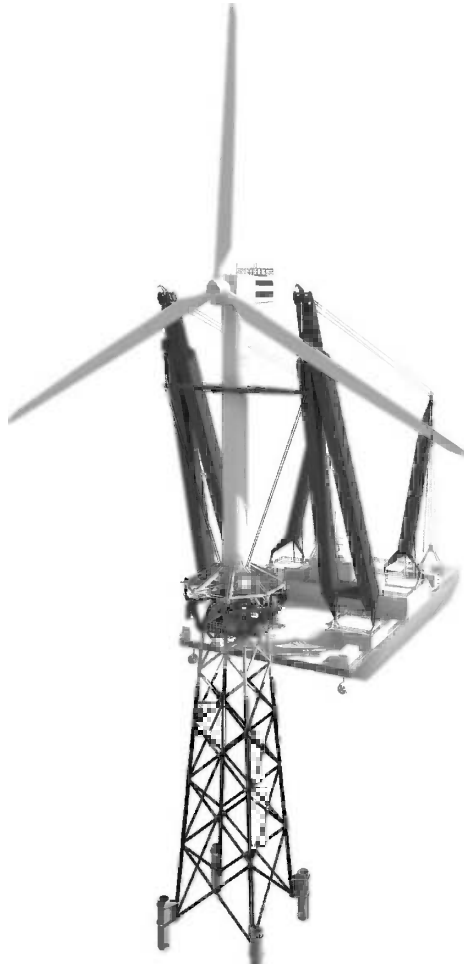


Figure 11.53 Installation of tower, nacelle and rotor assembly by floating crane at Beatrice wind farm. Reproduced by permission of Talisman Energy UK Ltd.

Comparison with monopile weights

The monopile support structure weights including tower predicted by de Vries and Krolis (2007) for the Vestas V90 3 MW turbine (Section 11.6.1) can be used to estimate the corresponding values for the REpower 5 MW using a scaling rule that preserves the relationship between the support structure natural frequency and the nominal rotational frequency at rated power. Treating the monopile and tower support structure as a uniform weightless cantilever of length L , supporting a tower head mass of M , it can be easily shown that the natural frequency is given by

$$\omega^2 = \frac{3EI}{ML^3} \text{rad/sec}$$

Table 11.6 Approximate weights of jacket structure, secondary elements and piles at Beatrice and Alpha Ventus

Wind farm site	BEATRICE	ALPHA VENTUS
Water depth	45 m	30 m
Component		
Jacket	360 t	
Transition section, including platform	160 t	
Pile sleeves	163 t	Not applicable
Total for primary structure	683 t	425 t
Boat landings	68 t	
Anodes	52 t	
Misc (e.g. J-tubes)	25 t	
Total for secondary elements	145 t	85 t
Piles	468 t	315 t
Overall total	1296 t	825 t

(Seidel, 2007a and 2007b)

so that the second moment of area required to obtain a particular natural frequency is

$$I = \frac{ML^3\omega^2}{3E} \quad (11.73)$$

This formula will also hold true for the required mudline second moment of area of geometrically similar tapering support structures, except that the number in the denominator will be different.

If it is assumed that the wall thicknesses and radii of the monopile and tower design for the V90 3 MW turbine for a given water depth are all to be scaled up in the same proportion to achieve the larger second moments of area required for the REpower 5MW turbine monopile and tower design at the same water depth, then the cross-sectional area scaling factor is equal to the square root of the second moment of area scaling factor – that is,

$$k_A = \frac{\omega_5}{\omega_3} \sqrt{\frac{M_5 L_5^3}{M_3 L_3^3}} = \frac{\omega_5}{\omega_3} \sqrt{\frac{M_5 (D + H_5)^3}{M_3 (D + H_3)^3}} \quad (11.74)$$

where the ratio ω_5/ω_3 is set equal to the ratio of nominal rotational speeds during operation at rated power, D is the water depth and H_5, H_3 are the respective hub heights of the 5 MW and 3 MW machines. The weight scaling factor, k_W , is obtained by multiplying the cross-sectional area scaling factor by the cantilever length ratio, $(D + H_5)/(D + H_3)$. Table 11.7 gives the parameter values adopted in the above expressions.

The estimated monopile plus tower weights required to support the Vestas V90 3 MW turbine in different water depths and the scaled up values for the REpower 5 MW turbine are plotted in Figure 11.54, together with the total weight of primary structure, piles and tower

Table 11.7 Turbine parameter values used to derive weight scaling factors at different water depths

Turbine	Vestas V90 3 MW	REpower 5 MW
Nominal rotational speed, ω	16.1 rpm	12.1 rpm
Tower top mass, M	112 tonnes	410 tonnes
Hub height, H	68.5 m	87 m
Monopile + tower weight scaling factor for 30 m water depth	1	2.211

(taken as 210 tonnes) for the jacket-type support structures at the Beatrice and Alpha Ventus wind farms. It is seen that the weight saving for the jacket structures is significant and more marked at the greater water depth.

Integrated jacket structure and tower

One means of eliminating the heavy transition section between the jacket structure and tubular tower is the combination of the former with a lattice tower. The lattice tower legs would then be simply continuations of the jacket legs. Lattice towers supporting upwind turbines are usually waisted to provide adequate tip clearance, but in the case of a downwind turbine, the legs could be straight all the way from the seabed to the nacelle. Access from platform level

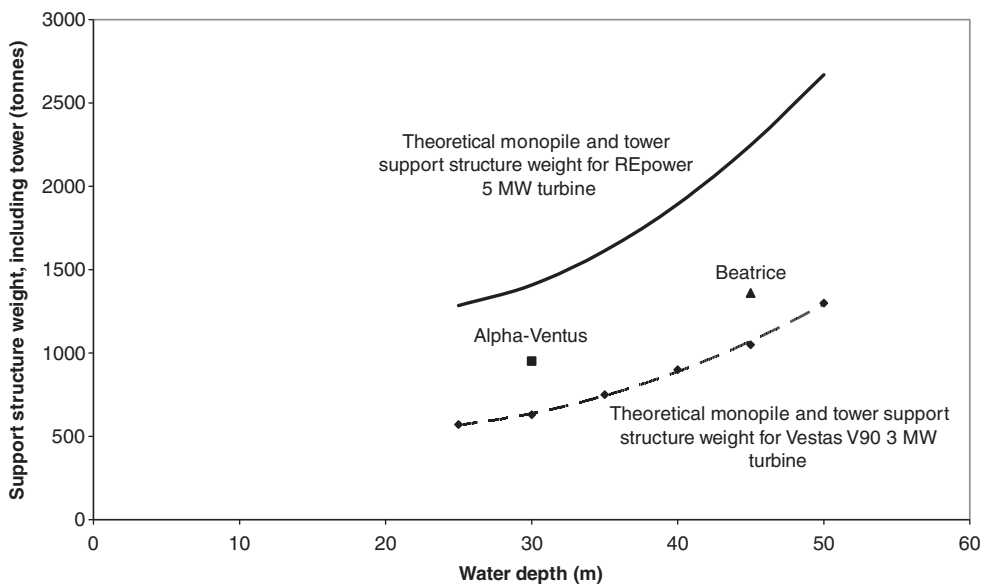


Figure 11.54 Theoretical variation of 5 MW turbine monopile support structure weight with water depth in comparison with Alpha Ventus and Beatrice jacket structure actual weights

to the nacelle would be by lift, which could be enclosed in a reinforced plastic tube to protect it from the elements.

11.6.5 Tripod structures

The chief advantages of tripod structures are that they are structurally more efficient than monopiles in resisting overturning moments and that wide spacing of the feet can avoid the need for piling if the feet are suitably ballasted. However, unlike the jacket structure, they cannot eliminate rotor thrust induced bending moments from the main members, although these moments are obviously reduced, as they are shared between several members and reduce linearly between the tripod apex and the sea bed.

A number of different forms of tripod structure are possible. The plain tripod design consists of three inclined legs meeting at a short column at the apex, with their feet connected by tie beams forming the sides of an equilateral triangle. The tower is then attached to the top of the short column.

In a variant of this, dubbed the centre column tripod (CCT), the short column is replaced by a long column extending to the seabed, where it is connected to the feet of the inclined legs by radial tie beams, in place of the tie beams between feet.

The number of members in the tripod can be reduced by adopting the flat faced tripod configuration (FFT), in which the column moves to the 90° vertex of a right-angled isosceles triangle forming the tripod base. In this case there are only two inclined legs and two tie members tying each foot back to the base of the column.

The structural benefit of the tripod in reducing column bending moments clearly increases with tripod height, but it is advisable to avoid placing the tripod apex at wave crest level to minimise wave loading. However, boat access is potentially more awkward if the apex is placed above wave crest level.

Alpha Ventus tripod structures

Six tripod structures have been installed in water 30 m deep in the North Sea to support Multibrid M5000 turbines at the Alpha Ventus site (see previous section). These are of the CCT configuration, with the tripod apex located below water level. As a consequence, the tripod base is relatively small and the tripod feet have to be secured against uplift by piling.

Table 11.8 compares an estimated tripod structure weight with that of the jacket structures at the same wind farm, based on Seidel (2007a).

It is seen that the estimated weight of the complete tripod structure including piles is over 50% greater than the corresponding figure for the jacket structure.

It is an open question as to whether larger FFT tripods would prove more economic.

11.6.6 Tripile structures

BARD has developed a tripile structure to support their own 5 MW turbines in water 25–40 m in depth. The design consists of three equispaced 3.35 m diameter vertical piles approximately 20 m apart, which are linked together by a three-arm transition structure onto which the tower is mounted. A vertical cylindrical tube projects downwards from the end of each arm and the three tubes are inserted into the tops of the three piles and grouted in the same way as monopile

Table 11.8 Comparison of tripod and jacket structure weights at Alpha Ventus

Structure type Component	TRIPOD estimated weights	JACKET weights
Primary structure, including pile sleeves, platform and transition section (jacket only)	840 t	425 t
Secondary elements (boat landings, J tubes and anodes)	85 t	85 t
Piles	370 t	315 t
Overall total	1295 t	825 t

transition pieces. The transition structure arms are of rectangular cross-section, enabling them to be fabricated easily from flat plate.

Figure 11.55 illustrates a tripod structure after installation.

The weight of the transition structure is 470 tonnes and the piles weigh up to 420 tonnes, so the total weight is comparable with that of the Alpha Ventus jacket structure (825 tonnes).



Figure 11.55 Tripile structure after installation. The yellow pile tops, which sport circumferential stripe markings, extend up to platform level with the cylinders above forming part of the three-arm transition structure. Reproduced by permission of BARD Group/Scheer. See Plate 7 for the colour figure

11.7 Environmental assessment of offshore wind farms

Offshore wind farms extend over large areas of often environmentally important, shallow waters and have the potential to make a considerable impact on the marine environment. As with any large onshore wind farm project, an Environmental Impact Assessment (EIA) is necessary before an application for consent to construct and operate an offshore wind farm is submitted. In Europe the Environmental Impact Assessment Directive (97/11/EC, 1997) requires an EIA to be undertaken and an Environmental Statement must be submitted with any proposal to develop an offshore wind farm. Detailed guidance on how this requirement is translated into UK legislation is given in CEFAS (2004).

The EIA assesses the environmental impacts of the project at all stages of construction, operation and de-commissioning. It considers measures that can be adopted for avoiding any undesirable impacts or, if this is not possible, mitigating them. The primary readership of the Environmental Statement is the authorities who must give permission for the project to proceed but also the various stakeholders with an interest in the marine environment. Coastal areas are used for a range of purposes, for example fishing, mineral extraction, shipping or recreation, and there are likely to be many groups with a keen interest in any proposed offshore wind farm development. Best practice on how the consultation should be managed is described in BWEA (2002).

The contents of any Environmental Statement are described in 97/11/EC as including a description of the project and the measures proposed to avoid and mitigate significant adverse effects. It will include the data necessary to identify and assess the main effects which the project is likely to have on the environment and an outline of the main alternatives studied by the project developer. The choices made in the development of the project must be justified taking into account the environmental effects. Both a comprehensive technical document and a shorter non-technical summary are required. Before detailed work is undertaken, including data collection and surveys, it is usual to agree a scoping report with the permitting authority that defines the extent of the investigations of the EIA.

In addition to the EIA required for an individual wind farm project, Strategic Environmental Assessments (SEA) are undertaken by national or regional governments to identify those areas where offshore wind farm development is appropriate and to identify where particular emphasis is required in the individual project EIAs (EWEA, 2009; DECC, 2009). For example, the SEA for the second round of UK offshore wind farm development identified three Strategic Areas; Greater Wash, Thames Estuary and off North-West England. It established a number of important principles. Larger wind farms (around 1000 MW) were preferred to smaller inshore projects. Wind turbines were excluded from a coastal strip of 8 km that was extended to 13 km in areas of particular landscape sensitivity. Also, wind turbines were not to be installed in areas of less than 10 m water depth due to the particular value of the natural environment of these shallow areas in terms of bird habitat, visual impact and use for recreation and inshore fishing. All three areas were close to electrical load centres and had good access to the electrical transmission system. After publication of the SEA, expressions of interest were received for the development of 7.2 GW of offshore wind energy.

EIAs for offshore wind farms start with a description of the complete proposed development and this will include the wind turbines, their layout and foundations, the array cabling, the offshore substation, the submarine transmission cable to shore, the land cable and any additional works for the land substation. Considerable work is undertaken to gather data of

the project area including surveys of the seabed, marine biology, birds and mammals, fish resources, marine traffic, the intertidal area where the power cables come ashore, and archaeology. A baseline survey for landscape and seascape visual assessment is also undertaken. It is common for a period of post-construction and commissioning environmental monitoring to be required.

Particular environmental impacts that need to be considered include:

- Coastal and sedimentary processes – sediment transport, scouring.
- Wave and tidal environment.
- Water quality – turbidity.
- Benthos – the flora and fauna of the seabed.
- Fish resources – migration patterns, nursery areas.
- Commercial fisheries.
- Marine navigation.
- Archaeology.
- Marine mammals – distribution, disturbance, displacement impacts of noise and vibration.
- Birds – distribution, disturbance, displacement, mortality.
- Designated areas (i.e. defined areas considered to be of environmental importance where particular protection policies are in place).

As with onshore wind farms, changes to the landscape and visual impact are likely to be of concern and a possible cause of objections to the wind farm development. DTI, 2005 provides detailed guidance on how Seascape and Visual Impact Assessment (SVIA) of offshore wind energy developments may be undertaken. The extent of the potential visibility of the development is described using Zones of Theoretical Visual influence (ZTV). This technique defines the areas from which one or more wind turbines, or parts of a wind turbine, are visible taking into account topography and curvature of the earth but not making any allowance for atmospheric conditions. Thus, a ZTV indicates the furthest possible extent of visibility. In order to take account of differing atmospheric and lighting conditions, the visual impact of the development is illustrated using wireframes and photomontages from viewpoints that have been agreed during the scoping phase. For large wind farms three-dimensional modelling and video-montages are used increasingly. For wind farms greater than 24 km from shore visual impacts are likely to be minor. The SVIA identifies the features of the existing seascape and the changes that the wind farm development would make to its visual character. The judgement is based on an assessment of receptor sensitivity and the capacity of the seascape to accommodate change. From the land, local residents are considered to be the most sensitive receptors with industrial activities, for example ports, the least sensitive. Similarly, out to sea, recreational boating has high sensitivity receptors while oil and gas rigs are less sensitive. The capacity of the seascape to accommodate change is evaluated considering the overall quality and sensitivity of the seascape to the wind farm as well as the value attached to the seascape. DTI, 2005 gives guidance on how these judgements can be made and the results recorded.

Noise, particularly during construction, has been an issue for some offshore wind farms that are close to shore. Installations of mono-pile foundations can be particularly noisy and restrictions may be placed on hours of work. The noise transmitted under water during piling operations can be very loud (200 dB at 1 m) and it is common practice to use a soft-start to piling operations so that fish and marine mammals leave the area.

Local scour around the turbine and substation foundations and cable routes is a common concern and scour protection (e.g. using rock dumping) may be required. This alters the surface of the seabed and can lead to changes in the benthic flora and fauna.

Ships and boats often use simple, low cost pulsed radar rather than the more sophisticated equipment commonly found in aviation radars that use the Doppler effect to distinguish between fixed and moving targets. There is less scope to use advanced digital signal processing techniques to reduce the effect of wind farms on marine radars and trials have indicated that wind farms produce blind areas in which other turbines and vessels could not be detected unless the observing vessel was moving (Howard and Brown, 2004). Also the high wind turbine structures produced large echoes and the radar response was strong enough to produce extraneous multiple and reflected echoes.

The effect of offshore wind farms on birds was considered in four main categories by Drewitt and Langston, 2006; collisions, displacement due to disturbance, barrier effects and habitat loss. Although the evidence was not complete, and a number of the wind farms studied were built away from large concentrations of birds, there was limited evidence of collision mortality. Similarly there was limited evidence of displacement due to disturbance but some barrier effect was reported from Horns Rev and Nysted wind farms. Loss of habitat from onshore installations (substations) and in shallow waters was highlighted as being of particular concern.

A comprehensive environmental monitoring programme was undertaken at the two Danish offshore wind farms at Horns Rev (160 MW, 14–20 km offshore) and Nysted (166 MW, 10 km offshore) between 2000 and 2006 (DONG Energy *et al.*, 2006). Comprehensive environmental monitoring programmes investigated the impact of the wind farms construction and operation on:

- Benthic flora and fauna, and in particular the impact of the foundations and scour protection.
- Distribution of fish around the wind farm and impact of electro-magnetic fields on fish.
- Behaviour of marine mammals and their reaction to the wind farm, including construction.
- The numbers and distribution of feeding and resting birds and the impact on migrating birds including risks of collisions.

The wind farms used hard scour protection on the sandy sediment seabed and this resulted in a more diverse surface and an increase in the biomass of the benthic communities by 50–150 times. There was little evidence of any effect of the wind farm on the local fish population or of electro-magnetic fields on fish. The main impact on marine mammals (seals and porpoises) was from pile driving during the installation of the mono-pile foundations when porpoises left the area. There was evidence that birds avoid the wind farms and migration distances were extended slightly. The modelled collisions rates of eiders at Nysted were low at 0.02%

(45 birds). In summary, the monitored environmental effect of these two large wind farms was limited.

An environmental monitoring programme was undertaken at two early UK offshore wind farms, North Hoyle and Scroby Sands. Each consisted of thirty 2 MW turbines directly connected to the 33 kV land-based power system with no offshore substation. North Hoyle wind farm is 4–5 miles off the North Wales coast and Scroby Sands 2 miles off the east coast of England. The environmental monitoring at North Hoyle, two years after commissioning, indicated small-scale changes to marine sediment composition, benthic ecology and bird distribution (DTI, 2006a). The trends could not be attributed to the wind farm and the overall conclusion was that there were no significant environmental effects due to the wind farm's construction and two years of operation.

The monitoring programme at Scroby Sands (DTI, 2006b) included bathymetric surveys and analysis around the turbines and the route of the export cables. There were also surveys of seal and little tern populations. The bathymetric surveys identified scour pits of up to 5 m depth and 60 m diameter around some wind turbines and these were back-filled with rocks. There was also evidence that the submarine cable depth of laying (initially 3 m) was changing with seabed conditions and rock mattresses were placed to protect the export cables from becoming exposed or damaged. The dynamics of scour around mono-pile foundations and scour protection with data from five offshore wind farms is reviewed in DECC (2008).

11.8 Offshore power collection and transmission

The first demonstration offshore wind farms were small and used simple, well proven wind turbines typically of 500–600 kW rating, often stall regulated with fixed speed induction generators. The power collection voltage was 10–11 kV with the submarine power collection cables, a few kilometres in length, directly connected to the on-shore electrical power system. This voltage was increased to 30–36 kV with the use of higher rated (e.g. 2 MW) variable speed turbines offshore but the connection to shore was still at the array power collection voltage. Then larger wind farms were constructed that required the use of offshore substations to increase the voltage for transmission to shore (to 132–150 kV). Offshore wind projects continue to increase in size and distance from land with one installation using a 400 MW $\pm$ 150 kV voltage source converter HVDC link to bring the power from several wind farms some 200 km to the grid connection point.

After the very early demonstration offshore wind farms, the electrical systems of offshore wind farms can be considered in three phases, corresponding to the three Rounds of UK offshore wind farm development. The division into three Rounds is due only to the administrative licensing process that was adopted by the UK but does reflect the development of offshore power collection and transmission systems (Figure 11.56).

- In Round 1, the offshore wind farms, each with 30 turbines, were of 60–90 MW capacity and less than 30 km from shore. It was then possible to use the wind farm power collection voltage of 30–36 kV, to take the power ashore using two or three cables in parallel.
- The Round 2 offshore wind farms were larger (up to 500 MW) but less than 90 km from shore. Connection to shore was possible at AC but offshore substations were needed to increase the transmission voltage up to 132–150 kV.

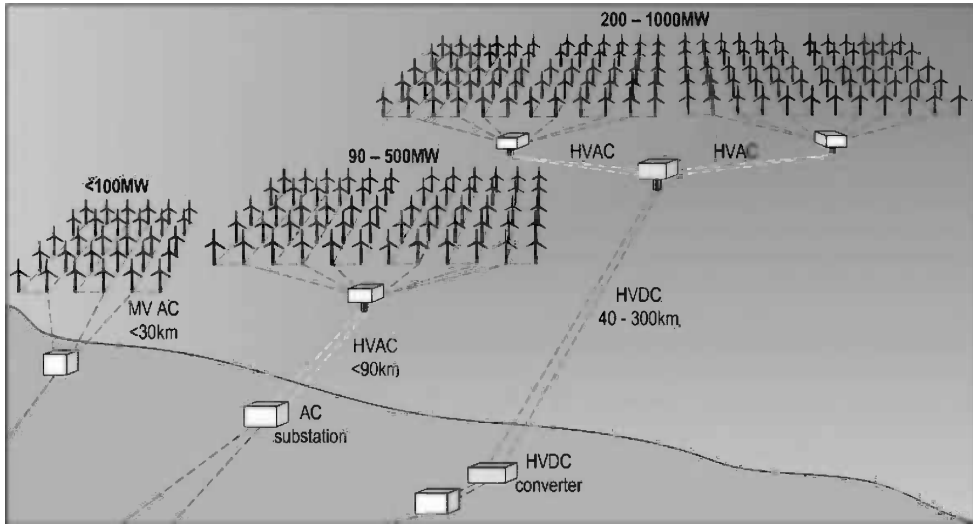


Figure 11.56 Options for offshore wind farm transmission

- Round 3 projects are now being planned and are up to 1000 MW and 300 km from shore. These are likely to use AC to collect the power from the wind farms and then multiple HVDC transmission links to bring it to shore.

In all cases a transformer and its associated switchgear is located within each turbine tower to increase the generator voltage of approximately 690 V up to 30–36 kV. The power collection array cables enter the offshore wind turbine through a J tube, attached to the transition piece of the monopile support structures, and the temperature of the short length of cable in the J tube determines its rating. The turbine transformer, often dry cast-resin type, is located in the tower together with the turbine 30–36 kV switchgear. In some larger turbines the turbine transformer has been located at the top of the tower to reduce the size, weight and losses of the pendant cable. For the largest wind turbines, consideration is also being given to increasing the voltage of the array power collection system to 72 kV and the generator/variable speed equipment to either 3.3 kV or 6 kV.

11.8.1 Offshore wind farm transmission

To obtain the power transfer capacity required, Round 1 UK offshore wind farms used multiple 33 kV cable circuits in parallel each taking up to 30–40 MW (600–800 Amps).² The wind turbines were connected in array strings and the three-phase submarine power collection cables extended to the shore where they were joined to land underground cables at a transition joint before entering a 33/132 kV substation.

² A small number of the more remote Round 1 offshore wind farms (e.g. Barrow) used an offshore substation but most avoided this cost by using multiple power transmission cables at the power collection voltage.

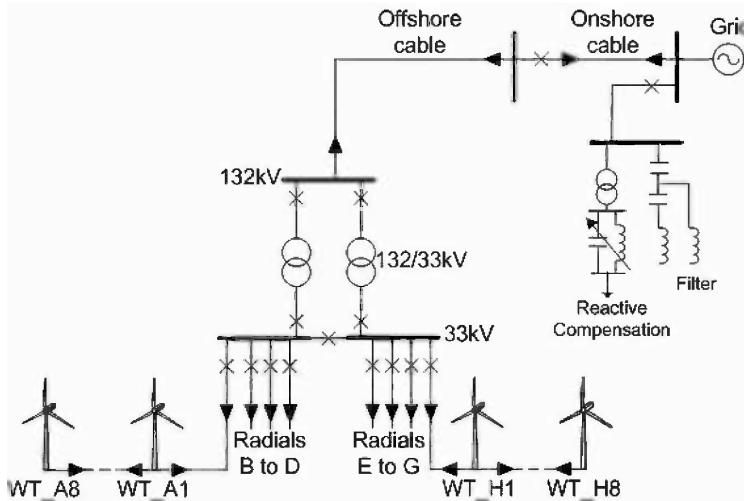


Figure 11.57 Typical UK Round 2 offshore wind farm power collection and transmission

Figure 11.57 shows a typical Round 2 offshore wind farm electrical system. Radial 33 kV XLPE³ array power collection cables with a capacity of 30–40 MW were used to connect up to eight turbines to an offshore sub-station. The conductor diameter of these cables was often tapered away from the sub-station. The sub-station was used to connect the array cables and transform the collection voltage up from 33 kV to 132 kV for transmission to shore. Sub-stations of up to 90 MVA capacity typically used one offshore transformer. Larger offshore sub-stations had multiple transformers to reduce the weight of individual items of equipment and allow the wind farm to continue to operate (at less than rated output) with one transformer out of service. There was normally an auxiliary diesel generator on the offshore substation platform. The power collection circuits used three-core XLPE cable for ease of laying and to reduce environmental impact.

The transmission circuits to shore also used three-core XLPE cables, usually at 132–150 kV but the use of 220 kV has also been considered. Submarine AC cables (800 mm² Cu) have capacities of approximately 200 MVA at 132 kV and 330 MVA at 220 kV. Once on land, a transition joint was used to connect the three-core submarine cables to three single-core underground cables. In some cases there was also an overhead line section before the cables connected to an onshore substation. The onshore sub-station was often located tens of kilometres from shore and included reactive compensation and harmonic filters.

Specialist equipment is required for handling and laying offshore cables both within the wind turbine array and for transmission of the power to shore with 220 kV submarine high voltage cable weighing up to 100 kg/m. Figure 11.58 shows the Lillgrund 3 core, 400 mm² Cu, 130 kV offshore transmission cable of length 7 km from the offshore substation (Figure 11.59) to land. Note the fibre optic communications cores within the cable.

³XLPE, Cross Linked Poly Ethylene insulated cables.

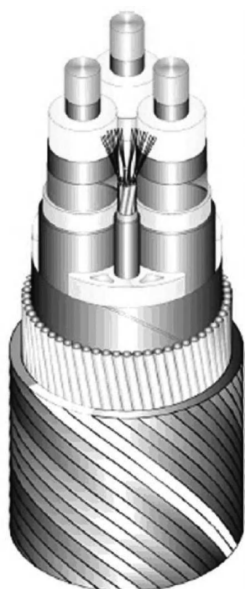


Figure 11.58 130 kV submarine transmission cable used at Lillgrund offshore wind farm. Lillgrund Pilot Project, 2008. Reproduced by permission of Vattenfall; cable manufactured by ABB HVC in Karlskrona, Sweden



Figure 11.59 Offshore sub-station at Lillgrund. Lillgrund Pilot Project, 2008. Reproduced by permission of Vattenfall. See Plate 8 for the colour figure

Round 3 offshore wind farm projects have yet to be constructed but the largest projects (e.g. Dogger Bank has a total wind turbine capacity of 9000 MW with a cable route length of 200 km) are likely to use HVDC transmission.

Comprehensive electrical system studies are required for the design of wind farm electrical offshore systems. For the Lillgrund project the following studies were undertaken (Lillgrund Pilot Project, 2008): cable sizing, load flow, short circuits, harmonics, grounding, voltage fluctuations, reliability, arc faults, dynamic simulation, protection coordination, insulation coordination, electromagnetic interference, lightning protection.

On land, electrical transmission systems deliver electrical energy to customers, some of whom have very high value loads and so need a reliable supply of power. Redundant circuits are installed to ensure that a single electrical fault, or sometimes two independent faults, does not result in loss of supply. In contrast the function of a wind farm offshore transmission system is to deliver wind generated electricity. The high costs of equipment offshore, particularly of submarine cables, and relatively modest value of the electrical energy mean that, as with onshore wind farms, total redundancy of circuits cannot be justified. Instead, partial redundancy can be provided for components with a high chance and/or a high impact of failure. An example is where two transformers, each of 50 or 70% of the wind farm capacity are installed. In this case, if one transformer fails only a small fraction of the energy is curtailed, as the wind farm will only operate at its full output a small fraction of the year.

A cost-benefit analysis was undertaken to determine the most cost-effective architecture for the connection of UK Round 2 offshore wind farms (Djapic and Strbac, 2008). This study balanced the costs of the offshore transmission system including those of; the submarine transmission cable(s), the offshore platform (including transformers, reactive compensation and switchgear), onshore circuits and capitalised costs of corrective maintenance against the capitalised costs of expected constrained energy due to equipment failures and maintenance as well as the cost of losses over the project life. The results showed that, similarly to land based wind farms, circuits for the transmission of offshore wind energy should be designed without explicit redundancy. Equipment failures in transmission circuits offshore require much longer repair times than those on-shore (e.g. the mean time to repair of an offshore transformer was assumed to be six months) but the cost of offshore equipment is so high that no redundancy in either transformers or submarine cables could be justified. Large offshore wind farms require multiple AC cable circuits due to the limited rating of submarine cables. Also the wind farm will be operating at its full output for only a fraction of the year. Thus, although the transmission link is designed with no redundant capacity its security is quite high. A calculation of a 600 MW offshore wind farm with 3×132 kV submarine cables indicated that failure of one cable would only result in 8% of the annual wind farm energy being curtailed. It may even be cost-effective to choose a transmission link capacity that is slightly lower than the simple summation of the full power ratings of all the wind turbines if they are dispersed over a wide area.

High voltage AC cables have a limited useful route length due to their capacitance and the reactive power they generate. Hence, for the large remote wind farms of Round 3 it becomes more cost effective to use HVDC if the cable route length exceeds 60–90 km (the precise break-even point when HVDC transmission becomes more cost-effective than AC transmission depends on wind farm rating and commercial factors influencing the price of equipment).

11.8.2 Submarine AC cable systems

Large offshore wind farms have very extensive networks of submarine AC cable for power collection and transmission to shore. The 30–36 kV power collection circuits as well as the 132–150 kV transmission to shore cables are unusual in being very long and without any significant load connected to them. Similar cable systems are not found in land based public electricity supply systems. The electrical systems of very large, land based wind farms, such as those found in parts of China and the USA, change from underground cable within the wind farm to overhead line as soon as is practicable in order to reduce cost.

The high voltage submarine cables use solid XLPE insulation with a high dielectric value. Hence, they have much higher shunt (parallel) capacitance than overhead lines that use air for their insulation. Thus, offshore wind farm cable circuits are unique and require particular consideration.

At 50/60 Hz, the high shunt capacitance with the high system voltage results in generation of reactive power (VARs) that reduces the capacity of the circuit to transmit real power and also increases the network voltage. The no-load reactive power generation of a cable is given by:

$$Q = 2\pi f l C V^2$$

where f is the system frequency, V is the rated voltage, and l is the length of cable. C is the capacitance of the cable per unit length.

Figure 11.60 shows the per-phase approximate equivalent circuit of a 1 km length of 132 kV cable. It will generate approximately 1 MVar when no power is transmitted through it. This reactive power increases the voltage at the wind farm and also uses part of the thermal rating of the cable. The voltage rise is controlled by reactive power compensation equipment, usually a fixed shunt reactor on the offshore platform and a STATCOM or SVC at the on-shore substation.

In addition to reactive power generation at fundamental frequency (50 or 60 Hz), the shunt capacitance of the large cable networks has important effects at harmonic frequencies.

Figure 11.61 shows the calculated impedance, seen from the 132 kV busbar, of the network of Figure 11.62 at harmonic frequencies. There are a number of series and parallel resonances caused by the interaction of the cable capacitance and inductive reactance of the cables and transformer. The resonances appear as low impedances (series resonance) and high impedances (parallel resonance). The harmonic resonant frequencies vary in frequency and magnitude depending on the number of cables and turbines in service (King and Ekanayake, 2010).

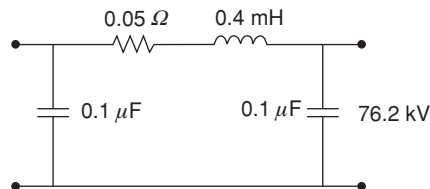


Figure 11.60 Per-phase approximate equivalent circuit of 1 km of 132 kV cable

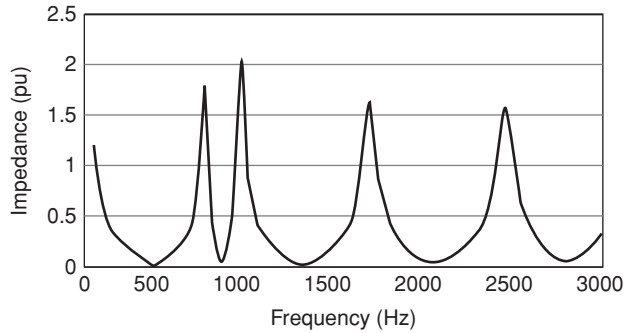


Figure 11.61 Impedance of the network of Figure 11.60 seen from 132 kV busbar

The voltage of all land based power systems contains harmonics, caused by the currents drawn by non-linear loads. Hence harmonic filters may be required at the on-shore substation to ensure that harmonic resonances within the wind farm network are not excited by this pre-existing distortion. In addition variable speed wind turbines using power electronic converters and power electronic reactive compensation equipment, such as STATCOMs, inject harmonic currents into the cable networks. If these harmonics are amplified they may cause significant voltage stresses in the offshore network and may also create complex interactions with the power electronic control systems causing unpredicted equipment malfunction.

Transient over-voltages in offshore wind farm networks are generally caused by circuit breaker operation unlike in land based overhead line systems where lightning is the main cause (Lars Liljestr nd, 2007). Vacuum circuit breakers are often used to control the MV networks in offshore wind farms, due to their high reliability and low maintenance requirements. In some cases vacuum circuit breakers can give multiple pre-strikes and re-ignitions and hence high-frequency transient currents. The extensive cable systems in offshore wind farms combined with a large number of step-up transformers results in many voltage reflections. The surge impedance of cables is less than 40Ω , whereas for overhead lines it is typically $300\text{--}400\Omega$ and a lower surge impedance results in a higher rate of rise of the transient over-voltage. Thus, the extensive cable systems of offshore wind farms can lead to high transient over-voltages and demanding requirements for the circuit breakers. It may also be necessary to fit surge arrestors to protect the wind turbine transformers from switching over-voltages.

Figure 11.63 shows a simulation of the voltage of the 33 kV collection cable of an offshore wind farm after it is energised by a vacuum circuit breaker.

Longer lasting temporary over-voltages may occur if the main circuit breaker at the onshore connection point disconnects the wind farm from the rest of the transmission network, so that the operating wind farm and HV cable become isolated (Akhmatov, 2006; Wiechowski, 2008). Depending on the protection system of the wind turbines, they may continue to inject active current into the isolated network, charging its capacitance. Once the turbines disconnect, the energy stored in the long AC cable capacitance and the shunt reactor inductance can oscillate with a low frequency, leading to high overvoltages lasting for several seconds. If this case is considered likely, overvoltage protection which is able to absorb the energy from the cable/reactor system may be required.

Although there are relatively few large offshore wind farms in operation at present, there has been a high rate of transformer failures reported. In two offshore wind farms the turbine

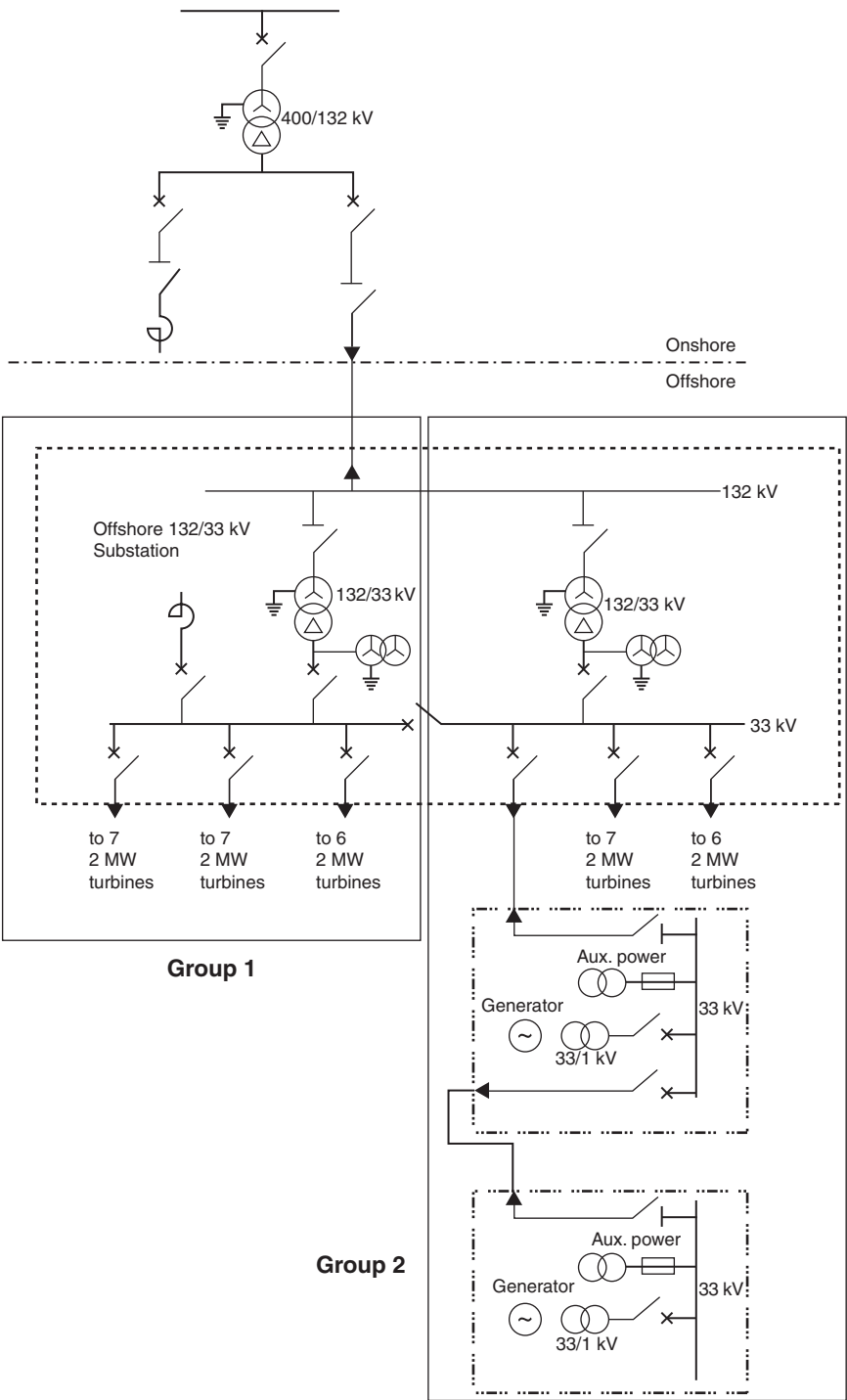


Figure 11.62 Typical offshore wind farm AC connection

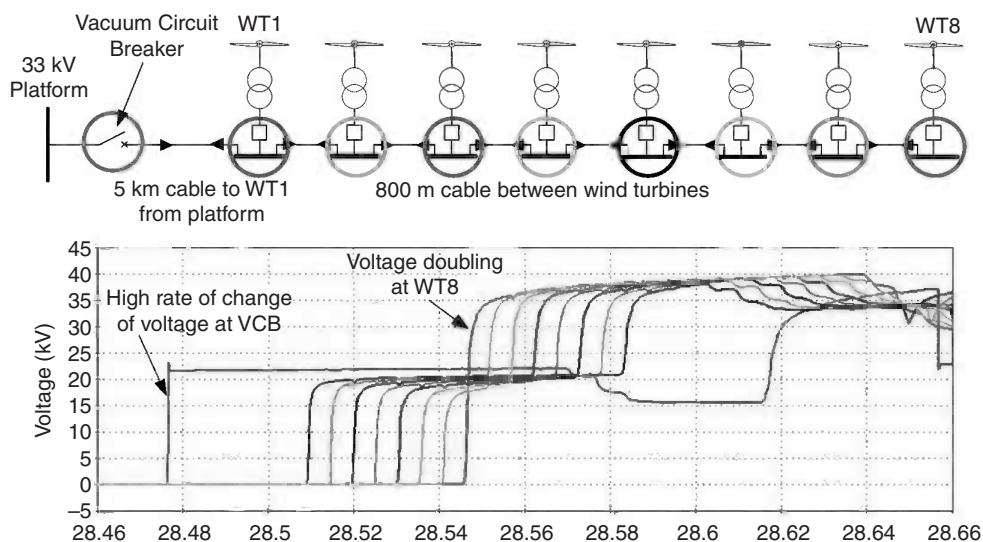


Figure 11.63 Voltage propagating through a wind turbine power collection radial circuit during energisation. Reproduced by permission of R. King

transformers were replaced after a large number of failures and in another offshore wind farm, a failure of the main offshore transformer led to a four and a half month outage of the entire wind farm. The reasons for these failures were complex and the transient performance of these very extensive cable systems is still the subject of research. However, it is possible that high frequency, high voltage switching transients made a significant contribution to these insulation failures. Hence, proper coordination of surge-protective devices with the insulation strength of electrical power equipment is essential for protecting the offshore electrical system from over-voltages (IEEE PES Wind Plant Collector System Design Working Group (2009)).

11.8.3 HVDC transmission

For cable route lengths above 50 km, HVDC transmission becomes a viable option (Koch and Retzmann, 2010). Above around 100 km HVDC is the only option since there will be a length limit of HVAC transmission due to the high levels of reactive power generated. For wind farms above around 200 MW in size, HVDC transmission tends to have the lowest number of cables connecting the wind farm to shore.

There are two distinct HVDC technologies, Line-Commutated Current Source Converters (CSC) and Self-Commutated Voltage Source Converters (VSC), Table 11.9. CSC technology is based on line-commutated devices such as thyristors that are switched on by a pulse and switch off when the current flowing through them reduces to zero. The converters have a requirement to be connected to a strong AC system, since the AC system voltage forces the current to commute from one phase to another. The current must always flow in the same direction through a thyristor which means that in order to change the direction of the power flow the voltage polarity of the converters must be reversed.

Table 11.9 Comparison between HVDC CSC and HVDC VSC

HVDC CSC	HVDC VSC
Line-Commutated Current-Sourced Converter (LCC/CSC)	Self-Commutated Voltage-Sourced Converter (SCC/VSC)
Turn-on capability only (Thyristors)	Semiconducting switches with Turn-on and Turn-off capability (IGBTs)
Low switching frequency (50/60 Hz)	High switching frequency (~1000 Hz)
Lower losses (~0.8% per terminal at full power)	Higher losses (~1.8% per terminal at full power)
Commutation within the converter is driven by the AC voltages (requires a minimum short-circuit power). A STATCOM is required offshore to provide this AC voltage reference if LCC is to be used offshore.	Do not require a commutating system voltage.
High space requirements, mainly due to filter requirements (to limit harmonics and to provide reactive power control)	Low space requirements
Converters absorb reactive power	Independent control of Active and Reactive power
In service since 1954	In service since 1997
Installed capacity >70 GW	Installed capacity ~1 GW
Highest voltage; ±800 kV DC	Highest voltage; ±400 kV DC
Largest scheme; 6400 MW (overhead line)	Largest scheme; 400 MVA (cable)

HVDC VSC uses semiconductor switches that can be turned on and off in a controlled manner which means that HVDC VSC is able to operate when there is no local generation and it can control the AC system voltage and frequency. The majority of HVDC VSC converters in operation at present switch between two or three voltage levels to create the AC terminal voltage of the converter. However, in order to reduce switching losses, Modular Multilevel Converter (MMC) designs for HVDC VSC are now becoming commercially available.

Whilst conventional CSC HVDC is well developed with lower capital costs and losses compared to the VSC technology, there is a significantly greater foot-print of the harmonic filters. VSC technology offers greater flexibility in terms of real and reactive power control and can be used with variable speed wind turbines (see Figure 11.64). VSC has been used for the only HVDC connection to an offshore wind farm commissioned to date.

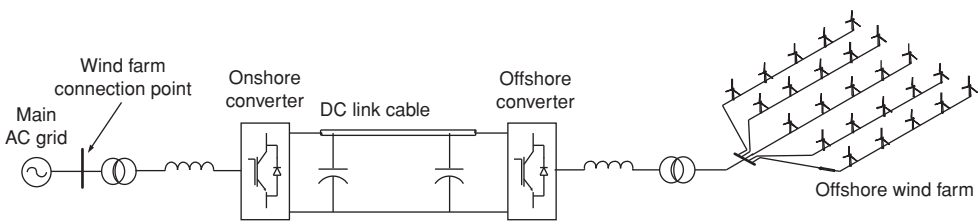


Figure 11.64 VSC HVDC transmission from an offshore wind farm

11.9 Operation and access

Access to wind farms offshore and hence maintaining high availability of the turbines is more difficult and expensive than for onshore wind farms. In many onshore wind farms, the tracks that are installed for access by large cranes during construction are maintained for the life of the project. Even if these have to be removed after construction then personnel access to the tower bases using light vehicles is straightforward and the large cranes can be re-mobilised and the access tracks reinstated if that is necessary. In contrast, the construction of offshore wind farms is undertaken using specialist vessels that, once the installation is complete, can be difficult and expensive to re-mobilise. Offshore wind turbines typically contain a light hoist for smaller items within the nacelle and may also include provision for the installation of a larger internal crane for major items of the drive trains. A gearbox or generator can then be replaced without mobilising a crane vessel. Replacing a damaged blade is likely to require a crane vessel (Amec Energy, 2004).

Access by personnel is required for routine maintenance, fault diagnosis and rectification with each wind turbine likely to require 10–20 person access visits per year. In addition to fault rectification, planned visits will be made for six and twelve month servicing of the turbines as well as various inspections for insurance and safety requirements. A minimum of a team of three is required for work to be undertaken on an offshore wind turbine. Twiddell and Gaudiosi (2009) estimate a total number of person access visits to offshore wind farms by 2010 of 200,000 per year. The early experience of the Horns Rev wind farm confirmed the need for frequent personnel access while there were 10,000 transfers between boat and tower during the construction and first two years of operation at North Hoyle wind farm (DTI, 2006a). Such frequent personnel access obviously requires safe and reliable means by which the turbines can be visited.

A number of access techniques have been tried. At Horns Rev frequent use was made of helicopters that did not land but lowered maintenance staff onto the top of the nacelle. More commonly, ladder and boat fender systems are used whereby a boat presses against tubular steel fenders attached to the wind turbine foundation and the technician transfers to and from a ladder. A number of other methods for transfer from the boat to the wind turbine have been investigated including dynamically controlled platforms. Catamarans are used in some wind farms as despite being more expensive than single hull boats they are more stable and have greater deck space.

UK experience indicates that transfers are possible with a significant wave height of up to 1.5 m using a ladder and boat fender system, depending on the wind and sea conditions. This then allows access to the turbines for 50–80% of the year (EWEA, 2009). At Scroby Sands (DTI, 2006b) the number of days when visits to the wind farm were not possible dropped from 143 (39% of the year) in the first year of operation to 76 (21%) in the second year as experience was gained and different access arrangements used. However, access during the winter months of high wind speed and hence revenue generation is limited by the wind and sea conditions and so for many wind farms there are continuing efforts to increase access up to 90% of the year, for example through improvement of access systems using catamarans and dynamically controlled transfer platforms. Higher rates of access are likely to be prohibitively expensive and, therefore, there are also continual efforts to improve the availability of the wind turbines through the use of SCADA (Supervisory Control and Data Acquisition) and remote condition monitoring (see Section 11.4). A number of simulation models to assess maintenance strategies of offshore wind farms have been developed. These

Table 11.10 Early operation of North Hoyle and Scroby Sands offshore wind farms

	North Hoyle	Scroby Sands
Capital cost	£1.35 m/MW	£1.26 m/MW
1 st year	July 2004–June 2005	January 2005–December 2005
Availability	84%	84%
Output	188 GWh	153 GWh
Capacity factor	36%	29%
2 nd year	July 2005–June 2006	January 2006–December 2006
Availability	91%	75%
Output	179 GWh	129 GWh
Capacity factor	34%	25%

DTI, 2006a and 2006b

use probabilistic representations of the weather, and hence access to the turbines, as well as failure of components and repair times.

Table 11.10 shows the performance of two early UK offshore wind farms during the initial two years of their operation. Both wind farms received a capital grant from the UK Government and so recorded the initial operation of the wind farm as well as undertaking an environmental monitoring programme (DTI, 2006a and 2006b).

Each wind farm of 30 wind turbines was close to shore and personnel access was with a ladder and boat fender system. The capital costs were typical of the rather low costs of offshore wind farms at that time. Subsequently costs of offshore wind farms rose considerably (Gross *et al.*, 2010).

On both sites mono-pile foundations were used. At North Hoyle a drive-drill installation technique was used with the mono-pile driven through the upper sand and clay layer and then a drill into the underlying rock. The sandy seabed conditions at Scroby Sands did not require any drilling. In both wind farms, the piles, towers and nacelles were installed mainly by jack-up vessels that used legs extending to the sea floor to provide a stable platform. The nacelle was installed with two blades attached and the third blade fitted attached to the hub once the nacelle was in position. At Scroby Sands, the assembly of the towers and turbines took approximately 12 hours with two turbines being installed every three–four days.

At both sites the availability was below the level that had been planned. There was a planned three-day grid outage at North Hoyle and at Scroby Sands one of the submarine-land cable joints failed resulting in one of the 33 kV circuits being out of commission for 12 weeks. A number of bearings were replaced *in situ* on both the generators and gearboxes at North Hoyle while at Scroby Sands all the generators were replaced and replacement of some gearbox bearings was required. At Scroby Sands additional remote monitoring was installed in order to increase availability.

The operation and maintenance costs to the project operators were £10–13 per MWh although as this was during the warranty period a number of costs relating to availability and unscheduled maintenance were borne by the manufacturer (estimated at one third of the operating costs). This agrees with the estimate of the Opti-EWECs project (van Bussel *et al.*, 2010) that operation and maintenance costs of offshore wind farms might be some 25% of the levelised costs of energy.

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Appendix A11

Table A11.1 Offshore wind farms from 1991–2010

Year of com- missioning	Wind farm		Offshore Substation	Distance to shore (km)	Depth of water	Wind turbine rating (MW)	Foundation	Reference
	missioning	Capacity (MW)						
1991	5	Vindeby, DK	N	1.5 to 3	2–4 m	0.45 (11 × Siemens Bonus)	Gravity	1
1994	2	Lely, NL	N	0.8	3–4 m	0.5 (4 × Nedwind)	Monopile	2
1995	5	Tunø Knob, DK	N	3	4–7 m	0.5 (10 × Vestas)	Gravity	2
1996	16.8	Irene Vorrink, NL	N	0.8	2–3 m	0.6 (28 × Nordtank)	Monopile	3
1998	2.8	Bockstigen, SE	N	4	5–6 m	0.5 (5 × WindWorld)	Monopile	2
2000	4	Blyth, UK	N	1.6	5m	2 (2 × Vestas)	Monopile	4
2001	40	Middelgrunden, DK	N	2	3–6 m	2 (20 × Siemens Bonus)	Gravity	5
2001	10.5	Utgrunden, SE	N	7	6–15 m	1.5 (7 × GE)	Monopile	2
2002	160	Horns Rev I, DK	Y	14–20	6–14 m	2 (80 × Vestas)	Monopile	6
2002	10	Yttre Stengrund, SE	N	5	6–8 m	2 (5 × NEG Micon)	Monopile	2
2003	10.6	Frederikshavn, DK	N	3.2	1–4 m	4 turbines 2.3 to 3 MW (Nordex, Vestas, Bonus)	Monopile, bucket	7
2003	60	North Hoyle, UK	N	6.5–8	5–12 m	2 (30 × Vestas)	Monopile	8
2003	166	Rodsand 1, Nysted, DK	Y	10	6–9 m	2.3 (72 × Siemens Bonus)	Gravity	9
2003	23	Samsø, DK	N	4	10–13 m	2.3 (10 × Siemens Bonus)	Monopile	2
2003	17	Rønland, DK	N	0.1	0–2 m	2 to 2.3 (4 × 2 MW and 4 × 2.3 MW)	Monopile	7

(Continued)

Table A11.1 (Continued)

Year of com- missioning	Wind farm Capacity (MW)	Location	Offshore Substation	Distance to shore (km)	Depth of water	Wind turbine rating (MW)	Foundation	Reference
2004	25	Arklow Bank, IE	N	10	1–5 m	3.6 (7 × GE)	Monopile	10
2004	60	Scroby Sands, UK	N	2.5	0–8 m	2 (30 × Vestas)	Monopile	11
2004	1.2	Setana, JP	N	0.7	10 m	0.66 (2 × Vestas)		2
2005	90	Kentish Flats, UK	N	9–10	3–5 m	3 (30 × Vestas)	Monopile	12
2006	90	Barrow, UK	Y	7	12–16 m	3 (30 × Vestas)	Monopile	13
2006	108	Egmond aan Zee, NL	N	10–18	15–18 m	3 (36 × Vestas)	Monopile	14
2007	10	Beatrice, UK	N	25	35–50 m	5 (2 × Repower)	Jacket	15
2007	90	Burbo Bank, UK	N	6.4–7.2	0–6 m	3.6 (25 × Siemens)	Monopile	16
2007	110	Lillgrund, SE	Y	10	4–13 m	2.3 (48 × Siemens)	Gravity	17
2008	30	Kemi Ajos I + II, FI	N	2.6–6	1–7 m	3 (10 × WinWinD)	Gravity	2
2008	194	Lynn and Inner Dowsing, UK	N	5	6–11 m	3.6 (54 × Siemens)	Monopile	18
2008	120	Princess Amalia, NL	Y	23	19–24 m	2 (60 × Vestas)	Monopile	19
2008	30	Thornton Bank I, BE	Y	27–30	13–19 m	5 (6 × Repower)	Gravity	20
2009	60	Alpha Ventus, DE	Y	45	28–30 m	5 (6 × Repower, 6 × Multibrid)	Jacket, Tripod	21
2009	2	Hywind, NO	N	10	220 m	2 (Floating wind turbine, 220 m deep water)	Floating	22

(Continued)

Table A11.1 (Continued)

Year of commissioning	Wind farm		Offshore Substation	Distance to shore (km)	Depth of water	Wind turbine rating (MW)	Foundation	Reference
	Capacity (MW)	Location						
2009	209	Horns Rev II, DK	Y (with separate accommodation platform)	31	9–17 m	2.3 (91 × Siemens)	Monopile	23
2009	90	Rhyl Flats, UK	N	8	4–11 m	3.6 (25 × Siemens)	Monopile	24
2010	180	Robin Rigg, UK	Y	11	0–12 m	3 (60 × Vestas)	Monopile	25
2010	173	Gunfleet Sands, UK	Y	8.5	2–13 m	3.6 (48 × Siemens)	Monopile	25
2010	102	Shanghai Donghai Bridge, CN	N	8–13	10 m (Average)	3 (34 × Sinovel)	Gravity	26
2010	6	Xianshui Intertidal Project, CN	N	3.5	0–4 m	2 (3 × Shanghai Electric)		26,2
2010	7.2	Avedøre Holme, DK	N		2 m	3.6 (2 × Siemens)	Gravity	2
2010	30	Vindpark Vanern, SE	N	1.4	1–22 m	3 (10 × WinWinD & Dynawind)	Gravity	2
2010	21	Sprogø, DK	N	3.5	6–16 m	3 (7 × Vestas)	Gravity base	2
2010	12	Frederikshavn, DK	N	10.6	1–4 m	6 (test site)	Monopile	27
2010	48	EnBW Baltic 1, DE	N	4.5	20–42 m	2.3 (21 × Siemens)	Monopile	2
2010	207	Rødsand II, DK	Y	16	6–12 m	2.3 (92 × Siemens)	Gravity	28
2010	300	Thanet, UK	Y	4	14–23 m	3 (100 × Vestas)	Monopile	29
2010	14	Kamisui, JP	N	12	5 m	2 (7 × Subaru)	Monopile	2
2010			N	0.1				

(Continued)

Table A11.1 (Continued)

Year of commissioning	Wind farm		Offshore Substation	Distance to shore (km)	Depth of water	Wind turbine rating (MW)	Foundation	Reference
	Capacity (MW)	Location						
2011	30	Rudong Intertidal Project, CN	N	3.5		1.5 to 3 (3 of 16 turbines – 2 × UP82-1500 and 1 × Mingyang – installed by the end of 2009.)		26
2011	315	Sheringham Shoal, UK	Y	17–23	14–23 m	3.6 (88 × Siemens)	Monopile	30
2011	400	BARD Offshore 1, DE	Y (HVDC)	100 (200km total export cable)	39–41 m	5 (80 × BARD)	Tripile	2
2011	184	Walney Phase 1, UK	Y	14	19–23 m	3.6 (51 × Siemens)	Monopile	2
2011	45	Weiwei Offshore wind farm, phase 1, CN (2)			1.5 (30×)			2
2011	165	Belwind, BE	Y	45	16–31 m	3 (55 × Vestas)	Monopile	2
2012	500	Greater Gabbard, UK	Y (2 - Gabbard and Galloper)	34	4–37 m	3.6 (140 × Siemens)	Monopile	31
2012	150	Ormonde, UK	Y	10	17–21 m	5 (30 × Repower)	Jacket	28
2012	92	Tricase, IT		20	92 m	2.4 and 3.5 (24 × Blue H, floating turbines in 118 m water depth)	Floating	33,2
2012	184	Walney Phase 2, UK	Y	14	19–23 m	3.6 (51 × Siemens)	Monopile	2
2012	248	Lincs, UK	Y	8	7–12 m	3.6 (69 × Siemens)	Monopile	2

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