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INTERNATIONAL STANDARD

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Wind turbines –
Part 4: Design requirements for wind turbine gearboxes

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CONTENTS

INTERNATIONAL ELECTROTECHNICAL COMMISSION

WIND TURBINES –

Part 4: Design requirements for wind turbine gearboxes

FOREWORD

1) The International Electrotechnical Commission (IEC) is a worldwide organization for standardization comprising all national electrotechnical committees (IEC National Committees). The object of IEC is to promote international co-operation on all questions concerning standardization in the electrical and electronic fields. To this end and in addition to other activities, IEC publishes International Standards, Technical Specifications, Technical Reports, Publicly Available Specifications (PAS) and Guides (hereafter referred to as "IEC Publication(s)"). Their preparation is entrusted to technical committees; any IEC National Committee interested in the subject dealt with may participate in this preparatory work. International, governmental and non-governmental organizations liaising with the IEC also participate in this preparation. IEC collaborates closely with the International Organization for Standardization (ISO) in accordance with conditions determined by agreement between the two organizations.
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International Standard IEC 61400-4 has been prepared by IEC technical committee 88: Wind turbines, in co-operation with ISO technical committee 60: Gears.

It is published as a double logo standard.

This first edition cancels and replaces ISO 81400-4 published in 2005. It constitutes a technical revision of ISO 81400-4 with extended content and changes in all pertinent sections.

This edition includes the following significant technical changes with respect to the previous edition:

a) extension of the scope to wind turbines above 2 MW rated power;
b) considerations for converging differing approaches to reliability in gear, bearing and wind turbine standards;
c) a new clause on wind turbine loads specific to drivetrains;
d) new clause on testing and validation of new gearbox designs;

e) updated bearing selection tables for different locations in a wind turbine gearbox;

f) expanded design considerations on the use of bearings based on avoiding standard failures;

g) a new clause on considerations and requirements in the design and analysis of gearbox structural elements;

h) updated considerations and requirements on lubricants and lubrication systems.

The text of this standard is based on the following documents of IEC:

FDIS	Report on voting
88/438/FDIS	88/441/RVD

Full information on the voting for the approval of this standard can be found in the report on voting indicated in the above table. In ISO, the standard has been approved by 11 P-members out of 12 having cast a vote.

This publication has been drafted in accordance with the ISO/IEC Directives, Part 2.

A list of all parts in the IEC 61400 series, published under the general title *Wind turbines*, can be found on the IEC website.

The committee has decided that the contents of this publication will remain unchanged until the stability date indicated on the IEC web site under "http://webstore.iec.ch" in the data related to the specific publication. At this date, the publication will be

- reconfirmed,
- withdrawn,
- replaced by a revised edition, or
- amended.

A bilingual edition of this document may be issued at a later date.

IMPORTANT – The 'colour inside' logo on the cover page of this publication indicates that it contains colours which are considered to be useful for the correct understanding of its contents. Users should therefore print this document using a colour printer.

INTRODUCTION

IEC 61400-4 outlines minimum requirements for specification, design and verification of gearboxes in wind turbines. It is not intended for use as a complete design specification or instruction manual, and it is not intended to assure performance of assembled drive systems. It is intended for use by experienced gear designers capable of selecting reasonable values for the factors, based on knowledge of similar designs and the effects of such items as lubrication, deflection, manufacturing tolerances, metallurgy, residual stress and system dynamics. It is not intended for use by the engineering public at large.

Any of the requirements of this standard may be altered if it can be suitably demonstrated that the safety and reliability of the system is not compromised. Compliance with this standard does not relieve any person, organization, or corporation from the responsibility of observing other applicable regulations.

WIND TURBINES –

Part 4: Design requirements for wind turbine gearboxes

1 Scope

This part of the IEC 61400 series is applicable to enclosed speed increasing gearboxes for horizontal axis wind turbine drivetrains with a power rating in excess of 500 kW. This standard applies to wind turbines installed onshore or offshore.

This International Standard provides guidance on the analysis of the wind turbine loads in relation to the design of the gear and gearbox elements.

The gearing elements covered by this standard include such gears as spur, helical or double helical and their combinations in parallel and epicyclic arrangements in the main power path. This standard does not apply to power take off gears (PTO).

The standard is based on gearbox designs using rolling element bearings. Use of plain bearings is permissible under this standard, but the use and rating of them is not covered.

Also included is guidance on the engineering of shafts, shaft hub interfaces, bearings and the gear case structure in the development of a fully integrated design that meets the rigours of the operating conditions.

Lubrication of the transmission is covered along with prototype and production testing. Finally, guidance is provided on the operation and maintenance of the gearbox.

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

IEC 60050 (all parts), *International Electrotechnical Vocabulary*
Available at <http://www.electropedia.org>

IEC 61400-1:2005, *Wind turbines – Part 1: Design requirements*

IEC 61400-3, *Wind turbines – Part 3: Design requirements for offshore wind turbines*

IEC/TS 61400-13:2001, *Wind turbine generator systems – Part 13: Measurement of mechanical loads*

IEC 61400-22:2010, *Wind turbines – Part 22: Conformity testing and certification*

ISO 76, *Rolling bearings – Static load ratings*

ISO 281:2007, *Rolling bearings – Dynamic load ratings and rating life*

ISO 683 (all parts), *Heat-treatable steels, alloy steels and free-cutting steels*

ISO 1328-1, *Cylindrical gears – ISO system of accuracy – Part 1: Definitions and allowable values of deviations relevant to corresponding flanks of gear teeth*

ISO 4287, *Geometrical Product Specifications (GPS) – Surface texture: Profile method – terms, definitions and surface texture parameters*

ISO 4288, *Geometrical Product Specifications (GPS) – Surface texture: Profile method – rules and procedures for the assessment of surface texture*

ISO 4406, *Hydraulic fluid power – Fluids– Method for coding the level of contamination by solid particles*

ISO 5725-2, *Accuracy (trueness and precision) of measurement methods and results – Part 2: Basic methods for the determination of repeatability and reproducibility of a standard measurement method*

ISO 6336 (all parts), *Calculation of load capacity of spur and helical gears*

ISO 6336-1:2006, *Calculation of load capacity of spur and helical gears – Part 1: Basic principles, introduction and general influence factors*

ISO 6336-2:2006, *Calculation of load capacity of spur and helical gears – Part 2: Calculation of surface durability (pitting)*

ISO 6336-3:2006, *Calculation of load capacity of spur and helical gears – Part 3: Calculation of tooth bending strength*

ISO 6336-5:2003, *Calculation of load capacity of spur and helical gears – Part 5: Strength and quality of materials*

ISO 6336-6:2006, *Calculation of load capacity of spur and helical gears – Part 6: Calculation of service life under variable load*

ISO/TR 10064-3, *Cylindrical gears – Code of inspection practice – Part 3: Recommendations relative to gear blanks, shaft centre distance and parallelism of axes*

ISO 12925-1, *Lubricants, industrial oils and related products (class L). Family C (Gears) – Part 1: Specifications for lubricants for enclosed gear systems*

ISO/TR 13593, *Enclosed gear drives for industrial applications*

ISO/TR 13989-1, *Calculation of scuffing load capacity of cylindrical, bevel and hypoid gears – Part 1: Flash temperature method*

ISO/TR 13989-2, *Calculation of scuffing load capacity of cylindrical, bevel and hypoid gears – Part 2: Integral temperature method*

ISO 14104, *Gears – Surface temper etch inspection after grinding*

ISO 14635-1:2000, *Gears – FZG test procedures – Part 1: FZG test method A/8,3/90 for relative scuffing load-carrying capacity of oils*

ISO 15243:2004, *Rolling bearings – Damage and failures – Terms, characteristics and causes*

ISO/TS 16281:2008, *Rolling bearings – Methods for calculating the modified reference rating life for universally loaded bearings*

AGMA 9005, *Industrial Gear Lubrication*

ANSI/AGMA 925-A02, *Effect of lubrication on gear surface distress*

ANSI/AGMA 6001-E10, *Design and selection of components for enclosed gear drives*

ANSI/AGMA 6123, *Design manual for enclosed epicyclic gear drives*

ASTM E1049-85, *Standard practices for cycle counting in fatigue analysis*

DIN 471, *Circlips (retaining rings) for shafts: Normal type and heavy type*

DIN 472, *Circlips (retaining rings) for bores: Normal type and heavy type*

DIN 743:2000, *Shafts and axles, calculations of load capacity, Parts 1,2, 3*

DIN 3990-4, *Calculation of load capacity of cylindrical gears: calculation of scuffing load capacity*

DIN 6885-2, *Parallel Key Geometries*

DIN 6892, *Mitnehmerverbindungen ohne Anzug – Passfedern – Berechnung und Gestaltung* (available in German only)

DIN 7190, *Interference fits – Calculation and design rules*

DIN 51517-3, *Lubricants: Lubricating oils – Part 3: Lubricating oils CLP; Minimum requirements*

EN 12680-3:2003, *Ultrasonic examination. Spheroidal graphite cast iron castings*

3 Terms, definitions and conventions

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in IEC 61400-1:2005 and IEC 60050-415 as well as the following apply.

NOTE The definitions in this standard take precedence.

3.1.1
bearing manufacturer
legal entity supplying bearings for the wind turbine gearbox, and who is responsible for the design and the application engineering of the bearing

Note 1 to entry: Typically, the bearing supplier will also manufacture the bearing.

3.1.2
certification body
entity that conducts certification of conformity of the wind turbine gearbox in accordance with IEC 61400-22

3.1.3
characteristic load
load value having a prescribed probability of not being exceeded

Note 1 to entry: See also 3.1.5, design load.

3.1.4
design lifetime
specified duration for which strength verification shall be performed

Note 1 to entry: Some serviceable components and wear parts may have a lower design lifetime than the one specified for the entire gearbox.

3.1.5
design load
load for which the strength of any component has to be documented

Note 1 to entry: It consists of the characteristic load multiplied by the appropriate partial safety factor for load.

Note 2 to entry: See also IEC 61400-1 and Clause 6.

3.1.6
double-row bearings
rolling bearings with two rows of rolling elements

3.1.7
equivalent load
load which when repeated for a specified number of cycles causes the same damage as the actual load variation if a specified life exponent applies

Note 1 to entry: When applied to load ranges, the equivalent load does not take the mean-stress level of the load cycles into account.

3.1.8
extreme load
that design load from any source, either operating or non-operating, that is the largest absolute value of the respective load component

Note 1 to entry: This component can be a force, a moment, a torque or a combination of these.

3.1.9
gearbox manufacturer
the entity responsible for designing the gearbox, and specifying manufacturing requirements for the gearbox and its components

Note 1 to entry: In reality, several legal entities may be involved in this process, which is not further reflected in this standard.

3.1.10
interface
defined boundary of the gearbox that is either a physical mount to another wind turbine subcomponent or a path of exchange such as control signals, hydraulic fluid, or lubricant

3.1.11
load reserve factor
LRF
ratio of the design load to the maximum allowable load on a specific component

Note 1 to entry: *LRF* can be determined separately for both the ultimate and fatigue strength calculation.

3.1.12
local failure
failure which occurs when at a critical location, the maximum allowable strain is exceeded

3.1.13
locating bearing
fixed bearing
bearing supporting axial forces in both directions

3.1.14
lubricant supplier
legal entity supplying lubricants for the wind turbine gearbox through either the wind turbine manufacturer, the gearbox manufacturer, or the wind turbine owner

Note 1 to entry: The lubricant supplier is responsible for the performance of the lubricant and the blending specifications, but will not necessarily produce any of the components, or blend the final product.

3.1.15
maximum operating load
highest load determined by the design load cases used in fatigue analysis as defined in IEC 61400-1, including partial load safety factor as applicable in accordance with IEC 61400-1

3.1.16
nacelle
turbine structure above the tower that holds the drivetrain, generator, other subcomponents, and parts of the controls and actuation systems

3.1.17
non-locating bearing
floating bearing
bearing supporting only radial load

3.1.18
paired bearings
two bearings of the same type at the same location

Note 1 to entry: These can be arranged so that their radial capacities complement and their axial capacities are opposite (e.g., two TRB or two ACBB in face-to-face or back-to-back arrangement), or they can be two bearings in tandem to increase both radial and axial load carrying capacities (see C.7).

3.1.19
rainflow matrices
representation of fatigue loads using a two dimensional matrix containing counts of cycle occurrence within sub-ranges of cyclic means and amplitudes

Note 1 to entry: See A.4.3.

3.1.20
time series
set of time sequences of loads, describing different operational regimes of the wind turbine

Note 1 to entry: These time series together with their corresponding occurrences specify the load history during the entire design lifetime.

3.1.21
wind turbine manufacturer
entity responsible for specifying the requirements for the gearbox designed in accordance with this standard

Note 1 to entry: Typically, the wind turbine manufacturer will design, manufacture and market the wind turbine.

3.1.22
wind turbine owner
entity who purchases and is responsible for operating the wind turbine

Note 1 to entry: In reality, the owner may contract different legal entities to operate, service and maintain the wind turbine. This distinction is not further reflected in this standard.

3.2 Conventions

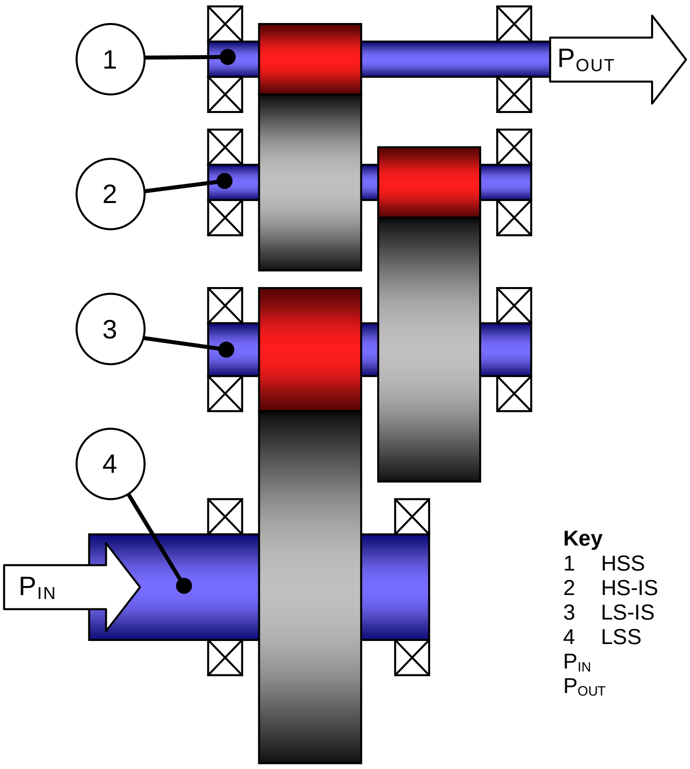
3.2.1
bearing position designations
the following abbreviations can be used to define bearing positions (shaft designations are defined in 3.2.2):

- RS: rotor side (normally upwind)
- GS: generator side (normally downwind)

In case of paired bearings the following can be used:

- IB: inboard (pointing inwards related to the shaft)
- OB: outboard (pointing outwards related to the shaft)

3.2.2
shaft designations – examples for typical wind turbine gearbox architecture
Figure 1 shows the designations of shafts in 3-stage parallel shaft gearboxes. In 4-stage gearboxes, the intermediate shafts are called “low speed intermediate shaft”, “medium speed intermediate shaft”, and “high speed intermediate shaft”.



Key

1	HSS	High-speed shaft
2	HS-IS	High-speed intermediate shaft
3	LS-IS	Low-speed intermediate shaft
4	LSS	Low-speed shaft
P_{IN}		Power input
P_{OUT}		Power output

IEC 2205/12

Figure 1 – Shaft designation in 3-stage parallel shaft gearboxes

Figure 2 shows the designations of shafts in 3-stage planet/helical hybrid gearboxes with one planet stage.

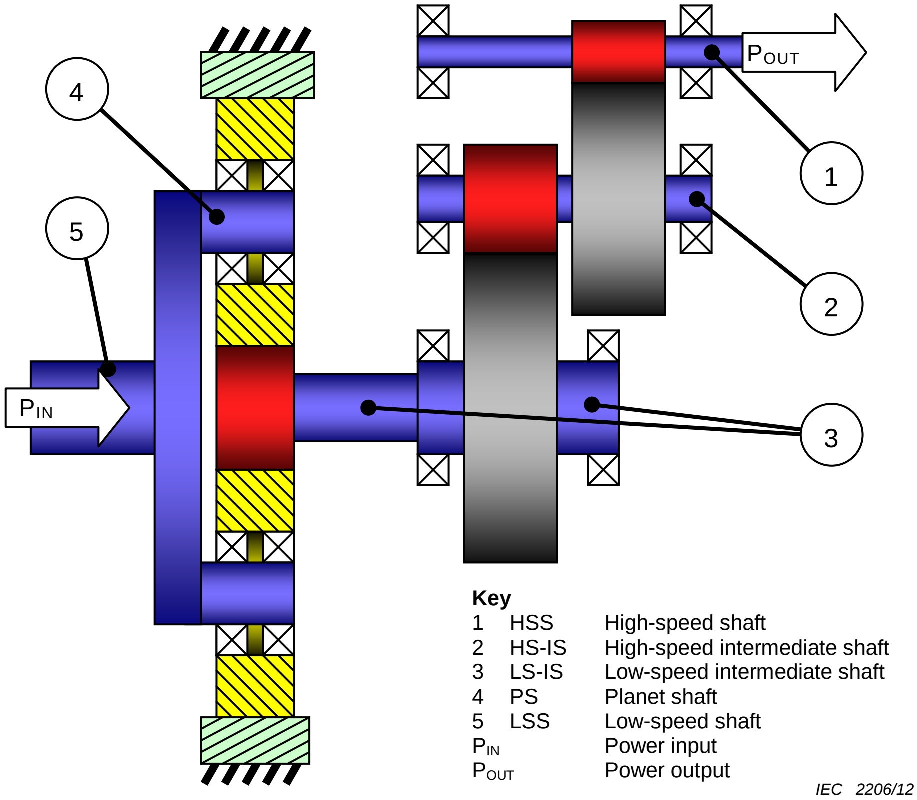
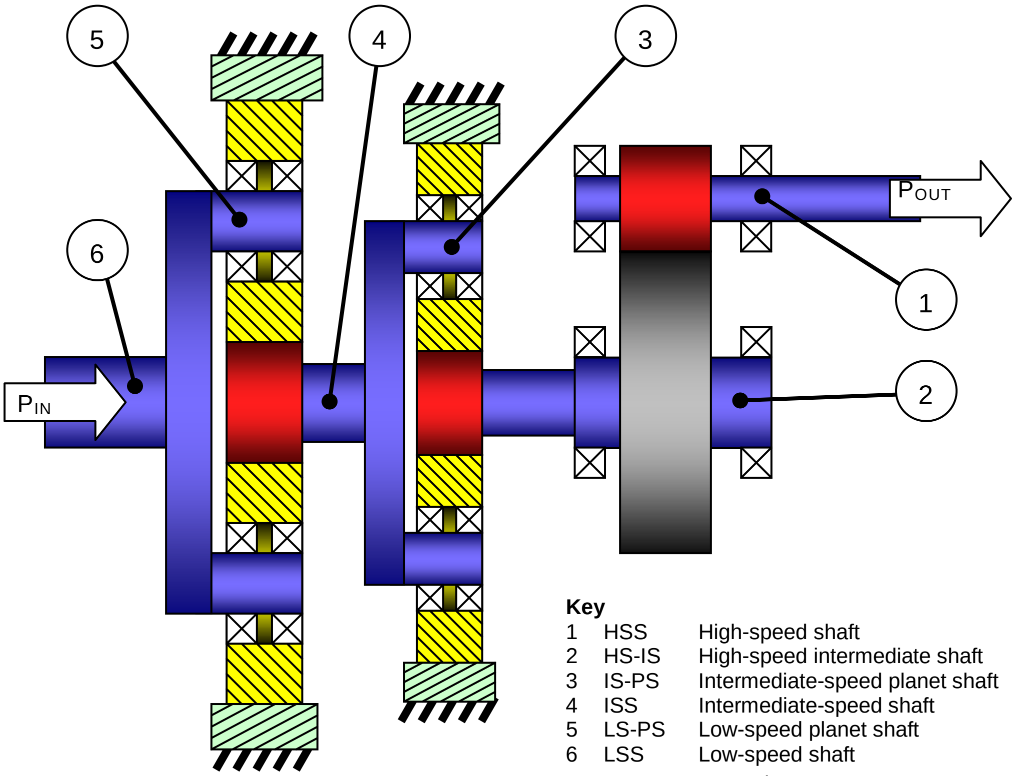


Figure 2 – Shaft designation in 3-stage gearboxes with one planet stage

Figure 3 shows the designations of shafts in 3-stage planet/helical hybrid gearboxes with two planet stages.



Key

1	HSS	High-speed shaft
2	HS-IS	High-speed intermediate shaft
3	IS-PS	Intermediate-speed planet shaft
4	ISS	Intermediate-speed shaft
5	LS-PS	Low-speed planet shaft
6	LSS	Low-speed shaft
P_{IN}		Power input
P_{OUT}		Power output

IEC 2207/12

Figure 3 – Shaft designation in 3-stage gearboxes with two planet stages

4 Symbols, abbreviations and units

4.1 Symbols and units

This standard uses equations and relationships from several engineering specialties. As a result there are, in some cases, conflicting definitions for the same symbol. All the symbols used in the document are nevertheless listed in Table 1, but, if there is ambiguity, the specific definition is presented in the clause where they are used in equations, graphs or text.

Table 1 – Symbols used in the document

Symbol	Description	Unit
a	acceleration	mm/s^2
a	semi-major axis of the Hertzian contact ellipse	mm
a_1	life adjustment factor for reliability	–
A_3	ultimate strain	%
A_5	ultimate strain	%
b	semi-minor axis of the Hertzian contact ellipse	mm
c	stiffness	N/mm Nm/rad
c	unit stress	MPa/N MPa/Nm
C	bearing basic dynamic load rating	N
CLI_i	consumed life index of the i^{th} load level	%
c (s)	uniaxial elastic unit stress at location s	MPa/N MPa/Nm
$C_{\delta L}$	bearing elastic constant	–
C_0	bearing basic static load rating	N
c_{ij} (s)	elastic stress tensor at location s for a uniaxial unit load	MPa/N MPa/Nm
$c_{ij,k}$ (s)	elastic stress tensor at location s for unit load k	MPa/N MPa/Nm
$c_{ij,m}$ (s)	mean value of the elastic stress tensor at location s	MPa/N MPa/Nm
C_T	contact truncation factor	–
d	damping increment	
D	accumulated damage	–
D_i	damage caused by cycle i	–
D_w	diameter of a rolling element in a bearing	mm
D_{pw}	pitch diameter of the rolling element set in a bearing	mm
e	bearing constant, limiting value for ratio of axial to radial loads, F_a / F_r	–
e_C	lubricant contamination factor	–
$e_{\sigma ij}$ (s,t)	elastic stress tensor at location s and time t	MPa
$e_{\sigma ij,a}$ (s)	local amplitude stress tensor	MPa
$e_{\sigma a,eq}$ (s)	equivalent stress amplitudes at location s	MPa
$e_{\sigma ij,m}$ (s)	mean local stress tensor	MPa
E	modulus of elasticity (Young's modulus)	MPa
F	force	N
$f_{\Sigma\delta}$	in-plane deviation of housing or carrier	mm
$f_{\Sigma\gamma}$	resulting mesh misalignment caused by deviations of the shaft alignment to the ideal axis projected into the face width	mm
$f_{\Sigma\beta}$	out-of-plane deviation of housing or carrier	mm
F_a	bearing axial load	N
$f_{H\beta}$	helix slope deviation of a gear	mm
f_{ma}	gear mesh misalignment	mm
F_r	bearing radial load	N
G_r	radial operating clearance of a bearing	mm

Symbol	Description	Unit
J	mass moment of inertia, indexed by the respective axis x, y, z	kg m^2
k	inclination exponent of the S/N curve	–
k	load sharing factor for the maximum loaded rolling element	–
K_γ	mesh load factor	–
$K_{F\alpha}$	transverse load factor (bending stress)	–
$K_{F\beta}$	face load factor (bending stress)	–
$K_{H\alpha}$	transverse load factor (contact stress)	–
$K_{H\beta}$	face load factor (contact stress)	–
K_{lc}	ratio of maximum contact pressure to contact pressure for line contact without misalignment	–
K_m	ratio of maximum contact pressure with misalignment to maximum contact pressure without misalignment	–
K_v	dynamic factor	–
l	length	mm
L	load value	N or Nm
$L_{10mr,i}$	modified reference rating life at load level i and 10 % failure probability, in 10^6 revolution	rev.
L_a	load amplitude	N or Nm
$L_{a,eq}$	equivalent load amplitude	N or Nm
L_e	elastic load limit	N or Nm
L_{h10}	basic rating life with 10 % failure probability	h
$L_k\ (t)$	time depending load component k of a time series	N or Nm
L_m	mean load in a bin	N or Nm
L_{nmr}	modified reference rating life at failure probability n, in 10^6 revolutions	rev.
L_{nr}	reference rating life at failure probability n, in 10^6 revolutions	rev.
L_p	total yield limit or plasticized load limit	N or Nm
LRF_f	load reserve factor against fatigue load	–
LRF_u	load reserve factor against ultimate load	–
L_{we}	effective roller length	mm
m	mass	kg
m	number of bins in a load spectrum	–
M	moment	Nm
m_n	normal tooth module	mm
n	failure probability	%
n	rotational speed	min^{-1}
$n_{eq,j}$	equivalent speed in the j^{th} bin of a load spectrum	min^{-1}
n_i or n_j	number of cycles in the i^{th} or j^{th} load level	–
N	number of cycles in characteristic stress-life curve	–
N_D	number of cycles at knee in characteristic stress-life curve for test specimens, from constant amplitude tests	
N_i	endurable number of cycles at the i^{th} load level, derived from the S/N-curve	–
N_i	number of times that cycle i occurs during the entire design lifetime	–
N_L	reference number of cycles	–
$n_{pl,\sigma,GF}$	section factor for global failure local stress in relation to R_P	–
$n_{pl,\sigma,LF}$	section factor for local failure local stress in relation to R_P	–

Symbol	Description	Unit
n_{ref}	reference number of cycles	–
p	exponent in bearing life equation	–
P	dynamic equivalent bearing load	N
$P_{i,j}$	load level of the i^{th} or j^{th} bin of a load spectrum	N
p_0	bearing contact pressure for point contact	MPa
P_0	static equivalent bearing load	N
P_{el}	generator electrical power	kW
p_{line}	approximated bearing contact pressure for line contact	MPa
p_{max}	approximated maximum contact pressure for line contact	MPa
Q	single roller maximum load for a clearance free bearing	N
q_i	time- or cycle- or revolution share on the i^{th} load level	–
Q_{oil}	oil volume	l
R	stress ratio	–
r_{12}	rolling element radius in plane of rotation axis	mm
r_{22}	raceway cross sectional groove radius	mm
R_a	arithmetic mean roughness	µm
R_m	ultimate tensile or compressive strength	MPa
R_p	yield strength (yield point or offset yield point at 0,2 % plastic strain)	MPa
R_z	mean peak-to-valley roughness (as specified in ISO 4287 / ISO 4288)	µm
s	motion	mm
s	variable of the location (position)	–
S	contact osculation	–
S_0	(bearing) static safety factor	–
S_B	safety factor for scuffing	–
S_F	safety factor for tooth breakage	–
S_H	safety factor for pitting	–
t	time as variable	s
VI	viscosity index	–
v_t	pitch line velocity	m/s
X_0	bearing constant, static radial load factor	–
Y_0	bearing constant, static axial load factor	–
Y_{NT}	life factor for tooth-root stress for reference test conditions	–
Y_{Sg}	stress correction factor for gears with notches in fillets	–
Z	total number of rolling elements in a bearing row	–
Z_{NT}	life factor for contact stress for reference test conditions	–
α	angle of rotation	°
α_0	bearing nominal contact angle	°
γ_f	partial safety factor for load	–
γ_m	partial safety factor for material	–
γ_n	partial safety factor for consequence of failure	–
δ	deflection	mm
ε_{lim}	elastic plastic limit of notch strain	%
Θ_L	bearing misalignment shaft slope angle	°arc-min
κ	viscosity ratio	–

Symbol	Description	Unit
μ	auxiliary Hertzian parameter	–
ν	auxiliary Hertzian parameter	–
ν	actual kinematic viscosity	mm^2/s
ν_1	reference kinematic viscosity	mm^2/s
ν_{40}	kinematic viscosity at 40 °C	mm^2/s
ρ_{11}	curvature factor with respect to body 1 in plane 1	–
ρ_{12}	curvature factor with respect to body 1 in plane 2	–
ρ_{21}	curvature factor with respect to body 2 in plane 1	–
ρ_{22}	curvature factor with respect to body 2 in plane 2	–
σ	stress (true stress)	MPa
σ_a	amplitude of occurring stress cycle	MPa
σ_A	design fatigue strength of component at N_D cycles	MPa
$\sigma_{a,R}$	value of σ_a for stress cycle with minimum/maximum ratio R	MPa
$\sigma_{A,R}$	value of σ_A relevant to loading cycles with minimum/maximum ratio R	MPa
σ_D	characteristic fatigue strength of test specimen at N_D cycles	MPa
$\sigma_{D,R}$	value of σ_D from tests with minimum/maximum ratio R	MPa
σ_{el}	linear elastic stress	MPa
σ_{FE}	allowable stress number (bending)	MPa
$\sigma_{H\,\mathrm{lim}}$	allowable stress number (contact stress)	MPa
σ_I	maximum principal stress	MPa
σ_{III}	minimum principal stress	MPa
$\sigma_{ij,\mathrm{pre}}(s)$	local pre-stress tensor at location s	MPa
σ_{lim}	limiting stress level	MPa
σ_{max}	maximum elastic stress	MPa
σ_{prin}	maximum absolute principal stress	MPa
$\Sigma\rho_{\mathrm{line}}$	curvature sum for line contact	–
$\Sigma\rho_{\mathrm{point}}$	curvature sum for point contact	–
τ	auxiliary Hertzian parameter	–

4.2 Abbreviations

Abbreviations are given in Table 2.

Table 2 – Abbreviations

Abbreviation	Description
ACBB	angular contact ball bearing
AGMA	American Gear Manufacturers Association
ANSI	American National Standards Institute
ASTM	American Society for Testing and Materials
CEC	Commission of the European Communities
CRB	cylindrical roller bearing
CRTB	cylindrical roller thrust bearing
DGBB	deep groove ball bearing

Abbreviation	Description
DIN	Deutsches Institut für Normung
DLC	design load cases as used in IEC 61400-1
DR ACBB	double-row angular contact ball bearing
DR CRB	double-row cylindrical roller bearing
DR FCCRB	double-row full complement cylindrical roller bearing
DR TRB	double-row tapered roller bearing
EXT	extreme load (matrices)
EHL	elasto-hydrodynamic lubrication
FCCRB	full complement cylindrical roller bearing
FEA	finite element analysis
FMEA	failure mode and effect analysis
FPCBB	four-point contact ball bearing
FZG	"Forschungsstelle für Zahnräder und Getriebebau" TU Munich
GS	generator side (normally downwind)
HS-IS	high-speed intermediate shaft
HSS	high-speed shaft
IEC	International Electrotechnical Commission
ISO	International Organization for Standardization
LDD	load duration distribution (histogram)
LRD	load revolution distribution (histogram)
LS-IS	low-speed intermediate shaft
LS-PS	low-speed planet shaft (or axle)
LSS	low-speed shaft
NPT	national pipe thread
PAG	poly-alkylene-glycol or polyglycol, synthetic lubricants
PAO	poly-alpha-olefin, fully paraffinic synthetic lubricant based on synthesized hydrocarbons
PS	planet shaft (or axle)
PTO	power take-off, additional output shafts driving auxiliary equipment such as oil pumps
RFC	rain flow count
RMS	root mean square
RS	rotor side (normally upwind)
SRB	spherical roller bearing
SRTB	spherical roller thrust bearing
TCT	total contact temperature method (Blok's method)
TIFF	tooth interior fatigue fracture
TORB	toroidal roller bearing
TRB	tapered roller bearing
VG	viscosity grade
WTG	wind turbine generator (system)

5 Design for reliability

5.1 Design lifetime and reliability

The objective of the design of a wind turbine gearbox is to achieve high availability and a reliability that is sufficient to limit maintenance and repair cost throughout the design life. The design life should at least be the same as for the wind turbine. The design lifetime of a wind turbine is defined in IEC 61400-1 to be at least 20 years for wind turbine classes I to III.

IEC 61400-1 defines component classes as a function of potential consequences of failures. The gearbox is a class 2 component which is a "non fail-safe" structural component whose failure may lead to the failure of a major part of a wind turbine. A wind turbine gearbox is assembled from torque transmitting parts such as gears, pinions, shaft and couplings, machinery elements such as bearings, supporting structural elements such as torque arms or housings, and bolted connections. The various components are designed using component specific design standards such as ISO 76 and ISO 281 for rolling element bearings or ISO 6336 series for gears. These standards generally cover different applications and do not stipulate specific safety factors or design life that may be needed to meet the requirements of specific applications. These relevant component standards use different measures for reliability, and this makes it difficult to arrive at a common reliability level throughout a system such as a wind turbine gearbox.

When comparing different designs under identical environmental conditions, the system reliability of a gearbox design is influenced by many parameters, such as:

- the number of components;
- the target design reliability of each individual component;
- the uncertainty of material properties at the given size range and target reliability;
- the probability of general material deviations such as flaws, voids or inclusions;
- the capability of the employed manufacturing processes for making parts compliant with the specifications;
- the robustness of the design against variation in the environmental conditions, or the compliance and ability of the design to adapt to such variation; and
- the robustness of the design against variation in material and manufacture.

Assessing the system reliability of a gearbox requires consideration of the shape of the probability density functions and the ratio of calculated life to design lifetime. In many instances, the system reliability may be approximated by the reliability of the component with the lowest design life. These parameters are not independent from each other. For example, the probability of material and manufacturing deviations typically increases with size. Because distributed power path drivetrains such as planetary gear sets will typically use smaller components, the individual component reliability may increase. At the same time, these gearboxes contain a larger number of components which may have a detrimental influence on system reliability. The selection of the gearbox concept, and decision for the appropriate application specific minimum safety factors or design life for each component, shall take all these aspects into consideration. Experience will be the strongest driver for levelling the reliability between various components such that the required system reliability is achieved.

Scheduled maintenance and even scheduled replacement of components may be applied to increase the availability and reliability of the gearbox. Components with a design lifetime less than that specified for the entire gearbox shall be serviceable or replaceable. The expected design life of these components shall be specified as part of the operations and maintenance documentation. Condition monitoring systems may further increase the availability as they may detect failures in due time to plan for repair.

5.2 Design process

Designing a drivetrain for a wind turbine is an iterative process that integrates input from the key suppliers such as the wind turbine manufacturer, gearbox manufacturer, bearing manufacturer and lubricant suppliers. It is strongly recommended that all relevant parties in the design process be included as early as possible to obtain the best possible design. The design process flow chart shown in Figure 4 may be used as a starting point.

During the design process each group should be involved in a critical systems analysis (such as failure modes and effects analysis, FMEA) to identify design assumptions which strongly influence the correct design, manufacture and operation of the gearbox. A documented systematic process as agreed between the gearbox manufacturer and the wind turbine manufacturer should be applied to identify and weight items that impact risk and reliability. For design changes, only the changed components and their impact on the system should be addressed. The system analysis should include:

- relevant design load cases;
- system architecture;
- interfaces;
- interface requirements;
- functional requirements for each element;
- potential failure modes;
- potential consequences of failure;
- limited life components;
- lubrication and cooling;
- manufacturing quality demands;
- service and maintenance;
- monitoring;
- transportation;
- installation; and
- commissioning.

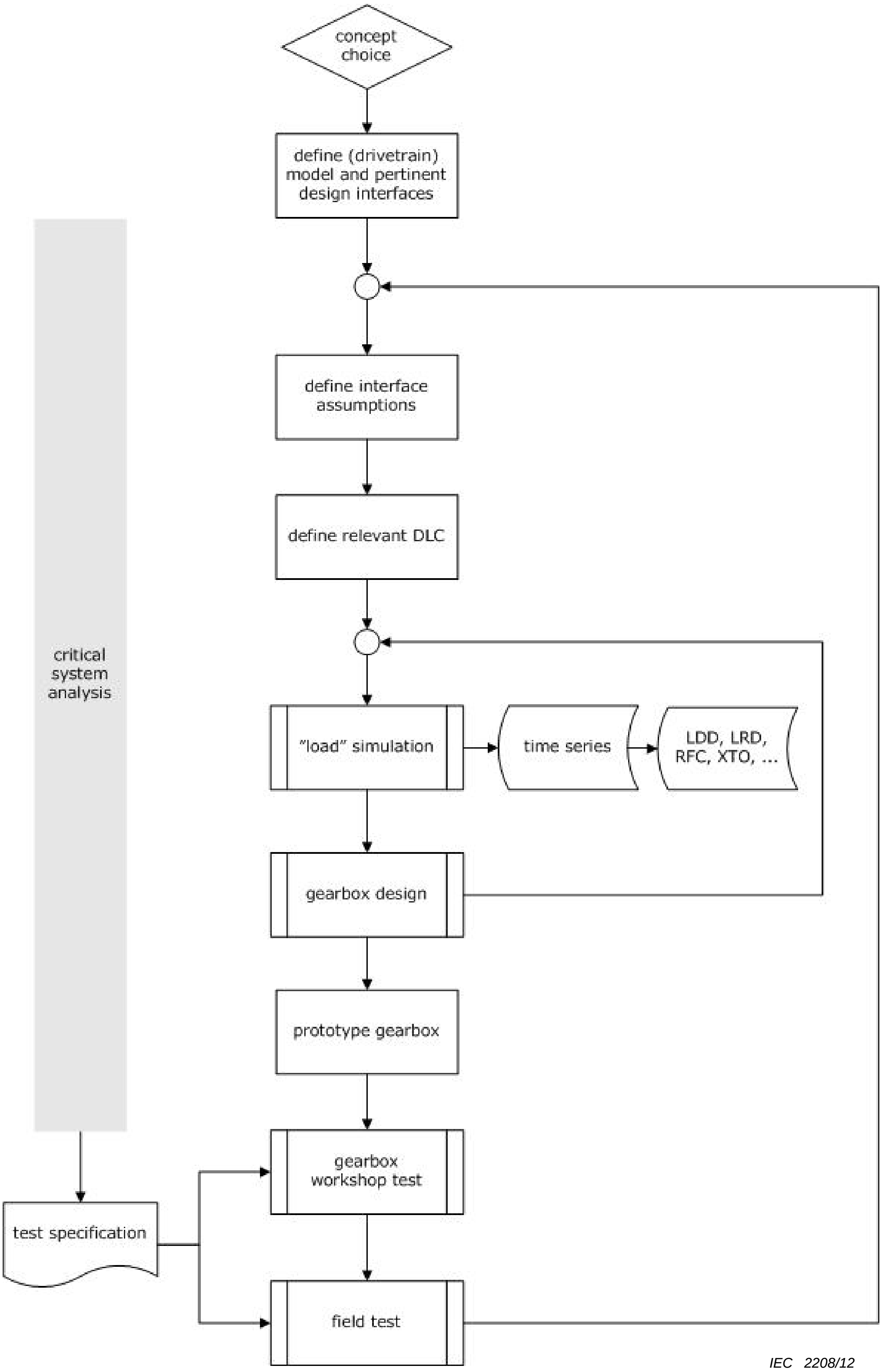


Figure 4 – Design process flow chart

The first step of wind turbine design process includes selecting the rotor design, the drivetrain configuration, the torque and power limiting methodology and control system architecture. The rotor converts wind to torque and other loads, but these design choices can significantly define the drivetrain initial conceptual design. An initial analysis of the rotor and control design results in a first iteration of the gearbox concept, the design envelope and the operational characteristics including the system loads.

This gearbox concept forms the basis for establishing a model of the drivetrain including the relevant interfaces and interface assumptions as noted in 6.1.2. Based on this description, relevant design load cases (DLC) can be defined per IEC 61400-1, and more detailed load calculations will be performed resulting in design loads at the defined interfaces (see 6.2.2). If the loading of the wind turbine is significantly influenced by the control and regulation systems, the impacts of control actions on loads should be taken into account in this analysis. The resulting design loads normally include time series simulations for many conditions. These can then be processed into extreme load tables, fatigue spectra in the form of rainflow matrices or load duration histograms, and other descriptions of operational characteristics (see 6.3).

With the loads and operational characteristics defined and specified, the initial gearbox design can take place. Since the actual design details of the gearbox affects turbine dynamics, reasonable estimates and assumptions need to be made for some of the motions, deflections and other dynamic response at the interfaces. These could be derived, for instance, from previous wind turbine designs. These estimates should be verified in subsequent testing, and the results shall be included in further simulations and design analysis.

The initial design results in one or more prototype gearboxes. The prototype design shall be tested intensively in a test rig such as at the gearbox manufacturer as described in 8.3.

The test specification is then based on the design loads as well as a design validation taking into account the uncertainties in the design identified jointly by wind turbine, gearbox and bearing manufacturers. The workshop prototype test results should be published in a test report that can be included in the wind turbine design evaluation module as per IEC 61400-22 Design Evaluation. If tests of subcomponents or subsystems are relevant (for instance, bearing solutions, lubrication systems, etc.) they normally take place in workshop tests parallel to the design process.

All testing shall be followed by an integrated design review process that uses test result feedback to verify, or iterate on, the design.

After the workshop prototype test (and resultant design review) a field test in a wind turbine shall take place. The test specification for the field test campaigns specific to the gearbox shall include inputs from the gear and bearing manufacturers, but it is strongly recommended that the field test only include items that cannot be accomplished in a workshop test. A report summarizing the result of this field test should be included in the type testing module as per IEC 61400-22.

If both workshop tests and field test results are acceptable, a test specification for the standard serial production test of the gearbox type can be defined. A preliminary production test plan should be considered to identify requirements for serial production testing (see 8.5).

5.3 Documentation

All steps in the specification, design and verification process will produce data and documentation that should be tracked, categorized and cross-referenced. A summary table for the documentation is contained in Annex F.

5.4 Quality plan

The methods and processes used in manufacturing the gearbox elements shall be documented as part of a quality plan.

6 Drivetrain operating conditions and loads

6.1 Drivetrain description

6.1.1 General

The first step in designing and specifying a wind turbine gearbox is to provide a detailed description of the interfaces between the gearbox and the wind turbine (see 6.1.2). The wind turbine manufacturer shall then describe what loads, motions and processes are transferred across these interfaces (see 6.1.3). Finally, a summary of the load calculations for these interfaces shall be provided (see 6.3).

6.1.2 Interface definition

All interfaces shall be accurately defined and described, and the interface conditions shall be documented.

The gearbox to be designed under this standard will normally be physically connected to the following wind turbine components:

- wind turbine rotor shaft (also referred to as the main shaft) to the gearbox low speed shaft;
- generator on the high speed shaft, usually by means of a coupling;
- nacelle mainframe via torque arm;
- blade control transfer mechanism (e.g., slip rings, actuator, hydraulics);
- mechanical brake, typically connected to the high speed shaft.

The common interfaces of these wind turbine components and the gearbox should match properly and support, transfer or otherwise tolerate forces, moments, and motions across their defined boundaries. The information required at these interfaces to properly specify the design of the gearbox changes with different wind turbine configurations.

In addition, some systems in the gearbox need to act across other types of interfaces, including:

- external lubrication system components such as reservoirs, pumps, coolers and filters;
- monitoring systems including sensors for temperature, oil condition, vibration, etc.;
- control actions such as heating.

The physical locations of each interface within the drivetrain structure shall be defined (see Annex A for guidance). Examples include:

- gearbox low speed shaft;
- gearbox high speed shaft;
- gearbox mounting;
- brake mounting surfaces;
- other actuators mounted to the gearbox;
- ports for lubrication and cooling system(s);
- sensors.

As part of the interface definition, information shall be supplied to accurately describe the geometry of the interfaces, for example:

- dimensional design envelope for the gearbox;
- placement of drivetrain components;

- geometry and dimensions of matched components such as couplings, mounting details, shafts, bolted interfaces and lubrication system.

Recommended interface definitions for common wind turbine configurations are found in A.3.

6.1.3 Specified requirements across interfaces

For each interface, the following information shall be supplied:

- forces and moments;
- rotational speed;
- motions / accelerations;
- deflections;
- temperatures;
- lubricant flow, temperature and pressures;
- misalignment or alignment allowances (relative to mountings).

Some of this information may not be available until the wind turbine design – including the gearbox – is completed. Therefore, reasonable assumptions need to be made for preliminary gearbox design calculations, for example based on previous wind turbine designs. Any assumption made in this process shall be documented as input for the final system verification. It is recommended that a sensitivity study be performed early in the design process to understand possible consequences of incorrect assumptions.

6.2 Deriving drivetrain loads

6.2.1 Wind turbine load simulation model

For each interface, the following shall be supplied:

- description of the drivetrain model used in the calculation of global wind turbine loads;
- assumptions for a set of initial design parameters which describe the gearbox components in the simulations, for example:
  - transmission ratio,
  - masses,
  - centre of gravity,
  - mass moments of inertia,
  - stiffness,
  - damping properties;
- the reference system of co-ordinates and reference points used for exchanging interface information (see A.3);
- information needed for describing the pertinent operating response and reactions at each of these interfaces (see A.4), for example:
  - forces,
  - moments,
  - rotational speeds,
  - motions,
  - offsets,
  - (mis)alignments,
  - deflections,
  - accelerations,

– temperatures,
– lubricant flow, temperature and pressure.

6.2.2 Wind turbine load calculations

The pertinent operating conditions at each of the agreed interfaces shall be determined in accordance with IEC 61400-1 or IEC 61400-3.

IEC 61400-1 or IEC 61400-3 define a minimum set of design load cases (DLC) that shall be considered in the design of a wind turbine. Additional DLC relevant for the design of the gearbox and its components shall be included. For example:

- situations resulting in axial motions at low loads;
- situations including generator switch operations, for example for 2-speed generators;
- situations resulting in torque reversals;
- situations resulting in acceleration and deceleration of the drivetrain including those caused on the high speed side (e.g., caused by brake events, grid loss or grid frequency variation);
- situations at reduced rated power possibly resulting in torque reversals (e.g. noise reduction operation, block control operation);
- situations at wind speeds below cut-in (e.g. parked);
- situations caused by asymmetric loads from mechanical brake (normal or fault);
- situations resulting from actions at periodic maintenance (e.g. emergency stop system tests);
- situations representing the load variations during grid events (e.g. low voltage ride through);
- situations resulting in high frequency inputs from the generator or high speed coupling (e.g. short-circuit, clutch slip and reconnection).

It may be necessary to perform more complex dynamic analysis of the drivetrain than currently available in the wind turbine aero-elastic design codes to properly address some of these specific situations. See 6.5 for further guidance on this.

Annex A contains more information on the consideration of gearbox-specific load situations. This annex also provides guidance on special requirements for the simulation models necessary to obtain realistic load assumptions for the wind turbine gearbox.

6.2.3 Reliability of load assumptions

The fatigue and ultimate load functions for a given component shall be compared with the related resistance functions as described in IEC 61400-1 or IEC 61400-3 taking into account the proper partial safety factors for loads, materials and consequence of failure.

The loading shall be verified through load measurements on a wind turbine as described in IEC 61400-22 and in 8.4.

6.3 Results from wind turbine load calculations

6.3.1 General

The summary of the load calculation shall be specified and at least comprise:

- the description of the DLC relevant to the drivetrain design;
- the duration of occurrence;

- the information on the load calculation model including a description of the drivetrain model employed in the load simulation.

Documentation of the wind turbine load calculations shall include time series as well as post-processed results as described below. Examples of documentation of load calculations are provided in Annex A.

6.3.2 Time series

Time series from the wind turbine load calculations relevant for the gearbox design shall be provided.

Such time series are derived from numerical simulations. They represent the loads experienced by the wind turbines under the reference site conditions defined in IEC 61400-1 or IEC 61400-3. The uncertainties are reflected by the partial safety factor for loads defined in IEC 61400-1 or IEC 61400-3.

Fatigue loads derived from these time series will typically be representative for normal operation. Isolation of single events or specific sequences may however only be indicative of the real behaviour. It is therefore recommended that any evaluation or post-processing of the time series should be mutually agreed between wind turbine manufacturer, gearbox manufacturer and bearing manufacturer (where applicable).

Extreme values from wind turbine load calculations during normal operation will in some cases be derived through extrapolation methods applied to the fatigue load calculations, as described in IEC 61400-1 or IEC 61400-3. Hence, time series will not be available for these extrapolations but only the extreme value of the parameter.

6.3.3 Fatigue load

6.3.3.1 General

Fatigue loading of a wind turbine gearbox is described in load cases specified in IEC 61400-1 or IEC 61400-3, such as power production load cases, start-up and braking procedures. The frequency of occurrence of each load case shall be checked carefully (see also 6.2.2). If the magnitude, duration and frequency of occurrence of ultimate design load cases may cause low cycle fatigue damage, they shall be included as part of the fatigue load spectrum and provided as time series.

6.3.3.2 Rainflow cycle counts (RFC)

Rainflow cycle counts (RFC) shall be determined using commonly accepted methodologies (see for example Downing and Socie (1982) or Matsuishi and Endo (1968) or ASTM E1049-85). An example of the presentation of rainflow count tables is shown in A.4.3.

The documentation of the RFC shall identify:

- which DLC have been considered in the evaluation;
- the frequency of occurrence for the different DLC;
- whether range (peak to peak) or semi-range (amplitude) is used to define cycle magnitudes.

6.3.3.3 Load duration distribution (LDD, LRD)

The load duration distribution is derived from the simulated time series. ASTM E1049-85 describes suitable methods. Load distributions may be expressed in time-at-level (LDD) or in revolutions-at-level (LRD). For turbines with variable rotor speed, LDD or LRD shall include a third dimension of shaft speed. Examples for the presentation of LDD or LRD are shown in Annex A.

The specified load level of each bin shall represent the highest absolute level of load represented in that bin. Bin width need not be uniform. Some bins may be negative load. The load spectrum shall also include loading from idling and stopped time.

The load spectrum durations may not add up to the design life even when idling is included. Some DLC and random extreme events may even cause the total simulation time to exceed the design life. It is also not necessary that the simulated life meets the design life, equivalent loads can be extrapolated with the respective S/N slope.

In the event of reducing the amount of bins for specific component calculations (see also 7.3.8.3 for bearings), the methodology used shall be neutral or conservative with respect to fatigue damage.

The documentation of the LDD shall identify:

- which DLC have been considered in the evaluation;
- the frequency of occurrence for the different DLC.

The wind turbine manufacturer may specify values for nominal torque and nominal rotational speeds. Especially for variable speed turbines, these values are arbitrary selections that may suit as reference, but are not suitable for any design calculations.

6.3.4 Extreme loads

It shall be stated under what conditions extreme loads occur (e.g., rotating or non-rotating situations, power production, and extrapolation). Extreme design loads shall be specified in tables, see Annex A for examples. These loads can be forces, moments, and torques.

Maximum load reversals and accelerations should be included in statistical summaries and identified separately with supporting time series where possible.

The partial load safety factors shall be included according to IEC 61400-1 or IEC 61400-3 and this value shall be stated in any load specification (see Annex A for examples).

6.4 Operating conditions

6.4.1 General

The operating conditions specified in the requirement specification for the gearbox shall cover the entire product lifecycle including wind turbine assembly, transport, installation, commissioning, and service (see also IEC 61400-1 or IEC 61400-3).

6.4.2 Environmental conditions

The wind turbine manufacturer shall specify in which environment the gearbox is supposed to operate, at least including:

- temperature ranges;
- relative humidity;
- contamination such as salt, dust, etc.;
- exposure to sun light and precipitation;
- if heat exchangers are within the scope of supplies:
  - ambient temperature outside the nacelle,
  - temperature at the cooler,
  - air quantity for the cooler,
  - air flow around the gearbox for cooling,

– ambient temperature inside the nacelle during operation and standstill (including survival temperature),
– air density.

For extreme environmental conditions the frequency of occurrence and their duration should also be specified.

6.4.3 Operating strategies

Operating strategies that influence the operating conditions of the gearbox shall be documented, including:

- start-up conditions including corresponding loads and speeds for all climate conditions;
- monitoring, warning limits, alarm limits, and alarm handling;
- temperature inside the nacelle.

6.5 Drivetrain analysis

At a minimum, drivetrain analysis shall be performed to verify the simplified WT aero-elastic model representation of the gearbox (to confirm the torsional stiffness value of the drivetrain supplied to the WT manufacturer by the gearbox manufacturer), to verify component specific gearbox loads due to dynamic amplification within the gearbox and to assess the influence of boundary (interface) conditions (see 6.1.2) on internal gearbox loading. Potential dynamic amplifications at resonances may be determined by modal analysis, time domain calculations, frequency domain calculations or any other equivalent method. Analysis shall also include the fundamental forcing and natural frequencies of the drivetrain as part of the wind turbine in addition to other frequencies such as gear mesh frequencies, rotational frequencies of the gearbox shafts and their harmonics.

Documentation of the drivetrain analysis shall, at a minimum, include:

a) a Campbell diagram including the main excitations in the system and relevant natural frequencies of the wind turbine system, the drivetrain (as part of the wind turbine), gear mesh frequencies, shaft frequencies and relevant harmonics;
b) stiffness, mass, inertia, and damping values of significant internal components, such as gears, shafts and bearings, and the complete drivetrain as used in the analysis;
c) an evaluation of results in terms of excitability of eigen modes.

A time domain dynamic simulation model of the drivetrain is useful in analyzing transient dynamic loading occurring within the gearbox. The data necessary to create a dynamic drivetrain model should be shared between gearbox and wind turbine manufacturers to ensure accurate representation of the drivetrain. Additionally, it is also useful to apply selected aero-elastic model time series to the dynamic drivetrain model to assess the influence of dynamic boundary (interface) conditions (see 6.1.2) on drivetrain loading. Any dynamic simulation models should be adequately verified to ensure representative behavior of the as-built drivetrain.

7 Gearbox design, rating, and manufacturing requirements

7.1 Gearbox cooling

The required cooling capacity shall be documented. Adequate cooling capacity shall be provided to remove the heat generated within the gearbox under the operating conditions specified in 6.4. The cooling capacity shall be confirmed in accordance with 8.3.4.

7.2 Gears

7.2.1 Gear reliability considerations

The load capacities of gears are calculated, according to the ISO 6336 series, using allowable stress numbers, some of which were derived from gear tests on back-to-back test rigs (surface durability) and from pulsator tests (tooth root strength). Information on some of the specific test gears can be found in ISO 6336-5.

Influences not covered by the reference tests are taken into account by influence factors, which were usually based on tests or experience with different gear macro and micro geometries, lubricants etc. These influences are described in ISO 6336-2 and ISO 6336-3.

The allowable stress numbers listed in ISO 6336-5 are based on 99 % survival probability of the test gears. Gear reliability does not just depend on the allowable stresses, it also depends on effects such as:

- operating conditions;
- manufacturing processes;
- quality control;
- effectiveness of lubrication system (oil distribution, oil cleanliness, cooling);
- material properties.

Because these effects influence each other in varying amounts, specific values for gear reliability are not determined using the ISO 6336 series. The minimum safety factors required by this standard are intended to take these effects into account. The values are derived from experience in wind turbine gearboxes.

7.2.2 Gear rating

7.2.2.1 Pitting resistance and bending strength

Gear rating shall be in accordance with ISO 6336 series. Miner's rule shall be applied in accordance with ISO 6336-6 to calculate safety factors using the load spectrum supplied by the wind turbine manufacturer. Life factors, Z_{NT} and Y_{NT} at the reference cycle number, N_L, shall be determined using the curve(s) intersecting at 0,85 at 10^{10} cycles using a log-log relationship. Pitting and bending fatigue lives shall be a minimum number of hours specified by the wind turbine manufacturer but not less than the specified design life.

In the early design stage, a gear calculation using an equivalent torque is allowed. For additional information on this, see ISO 6336-6.

All external gear teeth should be ground on the flanks only. Grinding notches on the tooth flanks or in the root fillets may reduce the bending strength of the teeth. A grinding notch is a form discontinuity produced by a grinding tool between the start of active profile and the tooth root that increases tooth root stress. With proper cutter design, heat treatment, and grinding procedures, grinding notches can be avoided. See 7.2.7.1 for additional details.

The minimum safety factor for pitting resistance shall be S_H = 1,25. The minimum safety factor for bending strength shall be S_F = 1,56.

7.2.2.2 Scuffing

Scuffing calculations are based on two different methods, Blok's total contact temperature (flash temperature) method and the integral temperature (bulk temperature) method.

ISO/TR 13989-1 and ANSI/AGMA 925-A02 are based on the total contact temperature method and yield similar results although ISO/TR 13989-1 provides a safety factor and

ANSI/AGMA 925-A02 provides a percentage of risk. ISO/TR 13989-2 is based on the integral temperature method. DIN 3990-4 includes both methods.

Evaluation of scuffing shall be performed using either the ISO/TR 13989-1 or ANSI/AGMA 925-A02 method and ISO/TR 13989-2. DIN 3990-4 may be used as an alternative to ISO/TR 13989-1 and ISO/TR 13989-2. The worst case results of total contact temperature method and integral temperature method shall be applied. The maximum value for the risk of scuffing is 5 % for ANSI/AGMA 925-A02. The minimum safety factor for ISO/TR 13989-1, ISO/TR 13989-2 and DIN 3990-4 is 1,3.

Scuffing rating shall be performed at the maximum operating load and rated speed. The gear bulk and gear mesh temperatures shall be calculated at rated load and rated speed and at the highest gearbox operating temperature at which the turbine controller either reduces power production or shuts down the turbine.

Time series from DLC for ultimate load analysis should be scanned for sequences where the torque exceeds the maximum operating load for a duration that could be sufficient to cause scuffing. If such events occur, this load level should be used in the scuffing rating.

NOTE Experimental results using ISO 14635-1 suggest that a minimum duration of 0,3 s is required before scuffing occurs.

The scuffing capacity of the oil shall be determined in accordance with ISO 14635-1. One stage lower than the fail load stage shall be used for the scuffing analysis. If no scuffing occurs in test load stage 12, class 12 can be used in scuffing analysis.

7.2.2.3 Micropitting

Micropitting is influenced by factors such as lubricant film thickness, material and microstructure, surface roughness and texture, contact geometry, load distribution and operating conditions. Influence parameters of the lubricant are the lubricant viscosity, chemical and physical properties of the base oil and the additives. Currently, no standardized calculation method is available for determining the risk of micropitting. However, ISO/TR 15144-1 suggests a method for assessing its occurrence. Therefore, a review of the parameters influencing micropitting is recommended.

7.2.2.4 Static strength

Static strength shall be calculated in accordance with the ISO 6336 series at the extreme torque using the static life factors Y_{NT} and Z_{NT}. The minimum safety factors shall be $S_F > 1,4$ for root bending strength, and $S_H > 1,0$ for surface durability. Additional consideration needs to be given to internal tooth fracture and possible yielding of gear teeth.

7.2.3 Load factors

7.2.3.1 General

The load factors for the calculation of pitting resistance and bending strength according to 7.2.2.1 shall be derived in accordance with the ISO 6336 series. The following subclauses provide application rules for this standard.

7.2.3.2 Dynamic factor, K_v

The dynamic factor, K_v, significantly affects gear rating. K_v shall be calculated in accordance with method B of ISO 6336-1:2006. If the calculation indicates a value of $K_v < 1,05$, $K_v = 1,05$ minimum shall be used unless proven by measurements. Measurement data shall be appropriately extrapolated to verify the value for the specified production tolerances (see 8.3).

7.2.3.3 Mesh load factor, K_γ

The mesh load factor K_γ accounts for deviations in load splitting e.g. in gearboxes with dual or multiple load paths or planetary stages. For a planetary stage, the default values given in Table 3 apply (for accuracy grades according to Table 4 or better) depending on the number of planets.

Table 3 – Mesh load factor K_γ for planetary stages

number of planets	**3**	**4**	**5**	**6**	**7**
mesh load factor K_γ	1,10	1,25	1,35	1,44	1,47

Lower values than given in Table 3, if used, shall be verified by simulation and measurement as described in 8.3.3.

7.2.3.4 Load distribution factors

7.2.3.4.1 Face load factor, $K_{H\beta}$

$K_{H\beta}$ shall be determined by numerical analysis according to method B of ISO 6336-1:2006. The influence of production variation on shaft parallelism and tooth alignment of pinion and gear should be included in the value of mesh misalignment. For fatigue calculations, the mesh misalignment shall be calculated in accordance with Equation (1). The extreme value of the mesh misalignment is calculated with Equation (2).

$$f_{\mathrm{ma}}=\sqrt{f_{H\beta 1}^{\ 2}+f_{H\beta 2}^{\ 2}+f_{\Sigma\gamma}^{\ 2}} \tag{1}$$

$$f_{\mathrm{ma}}=f_{H\beta 1}+f_{H\beta 2}+f_{\Sigma\gamma} \tag{2}$$

where

f_{ma} is the mesh misalignment of a gear set installed in the housing;

$f_{H\beta 1}$ is the helix slope deviation of the pinion;

$f_{H\beta 2}$ is the helix slope deviation of the gear;

$f_{\Sigma\gamma}$ is the resulting mesh misalignment caused by deviations of the shaft alignment to the ideal axis projected into the face width.

NOTE This method for determining f_{ma} is not as per ISO 6336-1.

The resulting mesh misalignment caused by shaft misalignment to the ideal axis $f_{\Sigma\gamma}$ considers all manufacturing variations affecting relative shaft parallelism that influence the mesh misalignment of the gear pair. At least the following influences shall be considered (and individually documented) when calculating $f_{\Sigma\gamma}$:

- out-of-plane deviation, $f_{\Sigma\beta}$, of the housing or planet carrier per ISO/TR 10064-3, including the influence of sleeves or bushings installed in the housing (where applicable);
- in-plane deviation, $f_{\Sigma\delta}$, of the housing or planet carrier per ISO/TR 10064-3, including the influence of sleeves or bushings installed in the housing (where applicable);
- influence of planet carrier alignment tolerance, including the range of planet carrier bearing clearance and deflection;
- variation of bearing clearance and deflection under load;
- variation of deflection of housing and planet carriers under load.

Where no statistical data are available, the resulting value for $f_{\Sigma\gamma}$ may be determined as RMS of these individual influences. Care should be taken that the independently assessed values are added to $f_{\Sigma\gamma}$ with the correct sign.

7.2.3.4.2 Numerical contact analysis

The design process of a wind turbine gear shall employ numerical analysis of the gear face load distribution in helix direction (see 7.2.3.5.3) and profile direction (see 7.2.3.5.2) at the same time, providing full information of the local loading in the entire contact area. Such a numerical contact analysis of the gear face load distribution shall at least account for:

- influence of adjacent meshes;
- influence of bearing operating clearances;
- influence of deflections of shafts, housings, carrier(s) and bearings;
- influence of local discontinuities in the stiffness at the extremities of the contact area.

Additionally, maximum operating loads, extreme loads and tolerance combinations using Equation (2) shall be checked with their resulting contact stress. Special care shall be taken to avoid stress risers at the extremities of the contact area.

If the contact analysis indicates a value of $K_{H\beta} < 1{,}15$, then a value of $K_{H\beta} = 1{,}15$ shall be used in the rating calculation. The result of the calculation model for the load distribution shall be verified by testing as described in 8.3.

7.2.3.4.3 Transverse load factors, $K_{H\alpha}$ and $K_{F\alpha}$

The transverse load factors $K_{H\alpha}$ and $K_{F\alpha}$ take into account the effects of an uneven distribution of load over several tooth pairs engaging at the same time. If the gears have a tooth accuracy as specified in Table 4, the value of 1,0 may be used for the transverse load factors $K_{H\alpha}$ and $K_{F\alpha}$.

7.2.3.5 Tooth modification

7.2.3.5.1 General

Profile and helix modification shall be used to minimize detrimental effects of tooth variations, bending and torsional deflections of teeth, shafts, bearings, housing, and manufacturing and assembly tolerances. Proper profile and helix modification increases load capacity and reduces noise. The design load for the profile and helix modifications should be the load level that contributes most to surface fatigue.

7.2.3.5.2 Profile modification

The design point for profile modification should be chosen carefully since these modifications can be designed for only one load and over modification is detrimental. This is particularly critical since the loads on wind turbine gears are variable. The design modification shall account for effects of all loads, scuffing risk, manufacturing variations, sound and contact ratios at low and varying loads.

7.2.3.5.3 Helix modification

Since the loads on wind turbine gears are variable, the design point for the helix modification must be chosen carefully because the helix modification can be designed for only one load, and over modification is detrimental. The design modification shall account for effects of all loads, manufacturing and operating deviations, load distribution in helix direction and sound.

7.2.4 Gear materials

All gear materials and processing shall as a minimum meet the requirements of ISO 6336-5. Allowable stress numbers $\sigma_{H\,\mathrm{lim}}$ and σ_{FE} shall be chosen for the selected quality grade in accordance with ISO 6336-5.

The effective case depth shall be designed in accordance with ISO 6336-5. Lower values than the recommended values to avoid case crushing according to ISO 6336-5 may be used, if verified by calculation and measurement. Heat treatment distortion and resulting variation of case depth from grinding shall be evaluated for hardened and ground gears.

Surface hardened internal gears have better wear properties than through-hardened gears.

7.2.5 Subsurface initiated fatigue

In addition to pitting failures (see 7.2.2.1), gear teeth may fail due to other subsurface initiated fatigue phenomena. The origin of these failures may occur in the case or in the transition between case and core (compare case crushing in ISO 10825).

Subsurface damage may also occur as tooth interior fatigue fracture (TIFF). TIFF is a fatigue failure caused by contact stresses and resulting in complete or partial tooth fracture where the crack origin is beneath the hardened layer and below the transition zone.

There is no standardized method established for determining the risk of such failures. However, a method for assessment of sub-surface strength is in DNV Classification Note 41.2, Subclause 2.13.

7.2.6 Gear accuracy

Geometric accuracy of gear elements shall be specified in accordance with ISO 1328-1. The maximum grades used shall be per Table 4. Gear accuracy may change when a gear wheel is assembled to its shaft. Therefore, the above accuracy specifications shall apply to the assembled gears. Where maximum accuracy is required, grinding after assembly with shafts should be considered.

Table 4 – Required gear accuracy

Gear type	Maximum accuracy grade (per ISO 1328-1)
External gear	6
Internal gear	7[a]

[a] Grade 8 may apply for run-out and total cumulative pitch deviation of nitrided internal gears due to process capability limitations.

7.2.7 Gear manufacturing

7.2.7.1 Method of manufacturing

The method and processing of the gear elements shall be specified.

All external gear teeth shall be cut to provide an adequate undercut to avoid grinding notches. Where protuberance cutters are used these shall have a minimum tool tip radius of 0,25 × m_n The final resulting tooth root geometry shall be used in strength calculations.

All gear teeth should have radii or chamfers at the tips of the teeth and over the full contour of the edges of the teeth. The value of chamfer or radii should be specified on the drawings.

If grinding notches do occur their effect on the tooth bending strength shall be evaluated. The preferred evaluation method is to analyse the worst-case notch using FEA or other methods to determine the reduction in tooth bending strength considering the most severe combination of notch location, depth, and radius. If no FEA or other similar method is available, the effect of the worst-case notch shall be evaluated using the Y_{Sg} factor per 7.3 of ISO 6336-3:2006.

During the design process, the boundary conditions for the maximum removed material, shape and location shall be properly documented such that the required life is still achieved. All gears shall be inspected after the production process to the design boundary limit conditions and the results recorded in the quality documents. If the results are out of the boundary conditions or reduce the bending safety factor below the limits specified in 7.2.2, the gear shall be rejected.

7.2.7.2 Gear tooth surface roughness

Wind turbine gears require smooth tooth surfaces to ensure adequate load capacity. Smooth surfaces are especially important with regard to micropitting resistance. For external gears maximum flank surface roughness shall be R_a = 0,8 μm. Maximum flank surface roughness for internal gears shall be R_a = 1,6 μm. B.2.6 includes recommended limits for gear tooth surface roughness for reducing the risk of micropitting.

On ground surfaces, the value of R_a is approximately 1/6 of the value for R_z. The above requirements may hence be translated for ground surfaces to $R_z \leq 4,8$ μm for external gears and $R_z \leq 9,6$ μm for internal gears.

Active flanks of gear teeth shall not be shot-peened as a final operation.

7.2.7.3 Surface temper inspection after grinding

Surface temper shall be controlled. Surface temper inspection after grinding shall be specified by a sampling plan. Sampling rate shall be based on effectiveness of process control and rejection rate on similar parts.

Any sampling plan less than 100 % must be supported by a formal, documented process capability study demonstrating that the specific processes and grinding machine procedures to be used are capable of producing parts free of rejectable grinding temper.

A well established, reliable inspection method is surface temper etch inspection in accordance with ISO 14104. Drive flanks shall meet level B1 or better. Non-drive flanks shall meet level B2 or better. All other inspection methods shall be agreed to between the wind turbine manufacturer and the gearbox manufacturer.

7.3 Bearings

7.3.1 General

This clause is valid for rolling element bearings. If plain bearings are to be used calculation methods and requirements should be agreed to between wind turbine, bearing and gearbox manufacturers.

7.3.2 Bearing reliability considerations

Bearing static safety and rating life calculations have been standardized in ISO 76 and ISO 281.

Bearing rating life is based on testing of a large amount of relatively small bearings operating under very well defined conditions. The characteristic fatigue strength is determined at 90 % survival probability. There is a limited number of tests for larger size bearings. The size of the bearing is amongst others considered through a statistical consideration related to the increase in the stressed volume. Methods have been derived based on extended testing to account for lubrication conditions such as lubrication film thickness, additives and contamination.

Advanced calculation methods are available to account for pressure distribution as well as deflections and tolerances of shafts and housings. Applying these methods gives a more realistic prediction of actual bearing life. Depending on operating conditions, this can result both in an increase or decrease of the calculated bearing life.

Bearing rating life requirements as specified by this standard are intended to take the uncertainties into account for wind turbine gearboxes.

Bearing life may be limited by several failure modes of which only few are possible to account for through calculations. Since some of the failure modes become more critical as the size increases (e.g. skidding or adhesive wear), bearings should not be unnecessarily oversized and the bearing size and arrangement should be carefully selected for the actual application.

7.3.3 Bearing steel quality requirements

ISO 76 and ISO 281 do not define specific bearing steel quality requirements to meet the bearing ratings. Bearing steel quality for wind turbine gearboxes shall meet ISO 683 requirements. Important aspects of steel quality are:

- chemical composition;
- steel cleanliness;
- steelmaking process;
- heat treatment and micro-structure.

7.3.4 General design considerations

7.3.4.1 Bearing selection and failure risk

Rolling bearings are typically selected for an application by calculation of the rating life against subsurface fatigue, and static safety against plastic deformation at extreme loads. Calculation methods for these criteria are widely available. However, in wind turbine gearboxes some other failure modes have been observed, which are not covered by these two selection criteria.

To minimize the risk of such failure modes, proper bearing selection and good design practices shall be adhered to. Some design principles are conflicting. For example, increased rolling element diameter will increase the static capacity and lower the risk of plastic deformation, but it may increase the risk of skidding and adhesive wear.

Figure 5 shows how bearing selection is influenced by different design criteria. The limiting sector between the principle curves displays the safe area for typical criteria. It should be noted that Figure 5 is not intended for design purposes. The relative positions of the different limiting curves can vary greatly for different bearing designs.

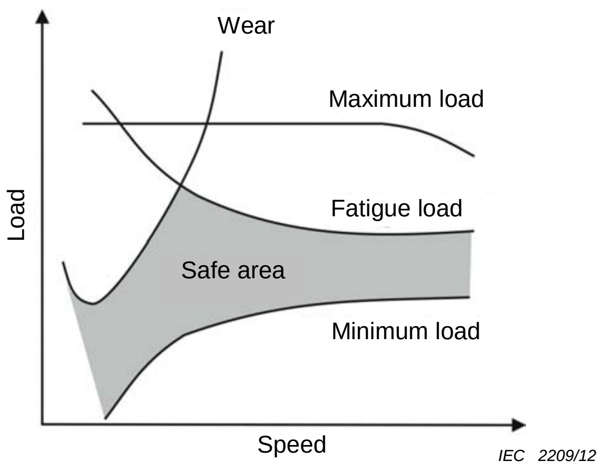


Figure 5 – Examples of bearing selection criteria

In addition to rating life requirements, the bearing selection shall consider other aspects such as:

- shaft assembly;
- suitability of the chosen bearing type for the actual bearing position;
- misalignment;
- low-load conditions;
- centrifugal forces;
- vibration;
- debris resistance;
- thermal expansion;
- oil supply and oil drain conditions;
- system eigenfrequencies;
- load sharing.

It is of vital importance to consider the interactions between the different bearings since they are influencing the dynamics of the complete shaft arrangement.

In the bearing selection process the complete load spectrum and time series shall be considered. The way the data has been considered for the specific design and the different failure modes shall be documented.

When designing the application, potential impact of electrical current flow, water/sea water ingress etc. should be considered and if possible eliminated.

Annex C provides guidance for selecting appropriate bearing types and arrangements for a wind turbine gearbox. These recommendations are based on experience. The guidelines shall not replace a detailed analysis of each bearing and arrangement during the design phase. Special care shall be taken before using bearing types or arrangements not shown or described as “limited experience”.

Bearing recommendations are valid assuming:

- the bearing is appropriately sized in accordance to this standard and the recommendations of the bearing manufacturer;
- the load spectrum for the bearings includes all external and internal loads;
- the bearing is adequately lubricated in accordance with this standard and the recommendations of the bearing manufacturer;
- the design of the adjacent components is in accordance with this standard and the recommendations of the bearing manufacturer.

ISO 15243 gives a general overview of bearing failure modes, many of which have historically been observed in wind turbine gearboxes. Bearing damage may occur in combination of the below listed failure modes.

7.3.4.2 Subsurface initiated fatigue (5.1.2 of ISO 15243:2004)

The resistance against subsurface initiated fatigue causing flaking, spalling or pitting as defined in ISO 15243 can be calculated in accordance with ISO 281.

No generally accepted calculation methods are currently available for the prediction of other forms of subsurface initiated fatigue which may occur, such as in conjunction with the formation of irregular white etched structures or hairline cracks. Subsurface initiated fatigue may be affected by load, number of load cycles, and loading rate (see ISO/TR 1281-2).

7.3.4.3 Surface initiated fatigue (5.1.3 of ISO 15243:2004)

Surface initiated fatigue is observed in wind turbine gearboxes in the form of surface distress, such as micropitting. ISO 281 life calculations include the effect on life from surface distress due to particle contamination and lubricant viscosity. Other causes of surface distress are not accounted for in the ISO 281 life analysis. Therefore a review of the parameters influencing surface distress is recommended.

Surface distress is influenced by factors such as lubricant film thickness, material and microstructure, surface roughness and texture, contact geometry, load distribution and operating conditions. Skidding and chemical reaction with the lubricant and/or moisture can also have an effect. Influence parameters of the lubricant are the lubricant viscosity, chemical and physical properties of the base oil and the additives.

7.3.4.4 Adhesive wear (5.2.3 of ISO 15243:2004)

Adhesive wear is a transfer of material from one surface to another with frictional heating and sometimes tempering or rehardening of the surfaces. It can be observed as skidding damage, which occurs because of:

- a loss of traction on the roller;
- excessive sliding of the whole roller set, when the load on the bearing falls below the bearing minimum load at high speed;
- high angular acceleration; or
- torsional vibration.

Skidding damage is commonly influenced by bearing size, type, and clearance or preload, and by size and location of the bearing load zone. Adhesive wear can also occur on guiding flange faces due to insufficient lubrication.

7.3.4.5 Moisture corrosion (5.3.2 of ISO 15243:2004)

Moisture corrosion is affected by ingress of atmospheric or sea water.

7.3.4.6 Frictional corrosion (5.3.3 of ISO 15243:2004)

Frictional corrosion can be observed in the form of false brinelling on the bearing raceways. This failure mode is typically caused by micro movements over a long period of time, e.g. when a wind turbine is parked with a brake applied.

Another form of frictional corrosion that may be found is fretting corrosion. This type of corrosion is a wear process caused by the combination of corrosion and the abrasive effects of corrosion product debris. This occurs at the contact area between two materials under load and subject to minute relative motion by vibration or some other force. Fretting corrosion typically occurs at interference fits that allow for micro movements due to improper fit selection or elastic deformation of one or both of the bodies joined by the fit. Both types of frictional corrosion will be augmented by the availability of moisture.

7.3.4.7 Excessive voltage (5.4.2 of ISO 15243:2004)

Damage by excessive voltage can occur as lightning damage, when lightning protection is insufficient. Excessive voltage can lead to local overheating of the material, which can cause rehardening or craters in the raceway visible to the naked eye.

7.3.4.8 Current leakage (5.4.3 of ISO 15243:2004)

Current leakage in wind turbine gearboxes starts with a large number of small craters, typically on the raceways. Over a period of time fluting results and is typically associated with bearings in proximity to the generator. Fluting damage is caused by electrical conduction through bearing materials and lubricant often caused by improper grounding of electrical components.

7.3.4.9 Overload (5.5.2 of ISO 15243:2004)

Plastic deformation and overload has been observed in rare cases in wind turbine gearboxes. When the actual bearing load does not exceed the static safety factor specified in 7.3.8.2 the effects of plastic deformation will be negligible.

7.3.4.10 Indentation from debris (5.5.3 of ISO 15243:2004)

Debris may come from a variety of sources including external contamination, gear and bearing wear, cage wear, or debris from failed components. Indentations from debris, whether initial or secondary damage, can lead to premature fatigue failure. Therefore, it is crucial to maintain oil cleanliness levels as specified in 7.6.9.

7.3.4.11 Indentation from handling (5.5.4 of ISO 15243:2004)

Indentation from handling can be easily avoided when good manufacturing practices are employed (see 7.3.6.4).

7.3.5 Bearing interface requirements

7.3.5.1 Specification

The design of the bearing arrangement for a wind turbine gearbox happens across a company interface between the gearbox manufacturer and the bearing manufacturer. For managing this critical interface without loss of relevant information, the gearbox manufacturer shall issue a detailed requirement specification for the bearing arrangement. The following clauses describe the minimum required content of this specification.

7.3.5.2 Interface definitions

For the chosen bearing arrangement, the following shall be specified:

- the reference system of co-ordinates and reference points used for exchanging interface information;
- information required to describe the interfaces between gearbox and bearings, for example:
  - arrangement and dimensions of gears and bearings in the gearbox;
  - dimensions, dimensional tolerances, form tolerances, surface hardness and surface roughness of shaft and housing journal diameters;
  - dimensions, dimensional tolerances, form tolerances and roughness of abutment diameters and support shoulders on shafts, housings and covers;
- stiffness of bearing environment;
- lubricant supply flows, pressures and temperatures as well as lubricant drains.

All interface definitions and the initial design assumptions shall be documented for review by the certification body.

7.3.5.3 Operating conditions

For each of the defined interfaces, information shall be provided for describing the pertinent operating conditions and reactions including their dynamic excitations, for example:

- forces;
- moments;

- rotational speeds;
- motions;
- offsets;
- (mis)alignments;
- deflections;
- accelerations;
- temperatures – nominal, maximum and minimum.

7.3.6 Bearing design issues

7.3.6.1 Cages

Bearing cages that both guide and separate the rollers (such as SRB) shall be steel or brass. For bearing cages that only separate the rollers, other cage material may be considered, if they are proven to be resistant to ageing due to operating temperature and lubricant for the design lifetime. The suitability of other cage materials shall be agreed to between the purchaser, the wind turbine manufacturer and the gearbox manufacturer and shall be based on thorough evaluation of field experience.

7.3.6.2 Operating internal clearance

The initial bearing internal clearance shall be selected to accommodate heavy interference fits and temperature differences. The appropriate selection shall be checked at worst tolerance combination at start-up condition. Radial clearance should be selected to minimize misalignment of the gear mesh. This is especially important for the low speed shaft of gearboxes that support rotor forces.

The operating clearance used in the advanced rating life calculation shall include the effects of the following:

- bearing initial clearance;
- fitting tolerances of shaft, housing and bearing rings;
- smoothing of fit for the specified surface finishes;
- temperature gradient between inner and outer ring (see Table 5);
- thermal expansion coefficient of the materials;
- operating temperatures (see Table 6).

Static deflection of the mounting may affect the clearance. Statistical tolerance analysis should be used to determine operating clearance. If this is not available, then 2/3 of the sum of all extremes shall be used. In addition, extreme tolerance combinations shall be checked.

Table 5 provides typical temperature differences between inner and outer ring observed under constant operation. Higher gradients may occur, for example, during start-up conditions, with additional air cooling during high winds or in cold climate conditions. These extremes may be relevant for selecting bearing internal clearance. The operating clearance used in the advanced rating life, and the assumptions made for determining this clearance, shall be documented.

Table 5 – Temperature gradients for calculation of operating clearance

Bearing position	Temperature difference between inner and outer ring at constant operation K
High speed shaft	10 to 30
High speed intermediate shaft	5 to 20
Low speed intermediate shaft	0 to 15
Planet	–5 to +5
Low speed shaft	0 to 10

The selection of operating internal clearance shall be verified during prototype testing of the gearbox. This can for example be accomplished by temperature measurement or by analysis of the bearing running pattern.

7.3.6.3 Shaft and housing fits

Relative motion (both axial and rotational) between the bearing rings, the shaft and/or the housing may cause damage by fretting corrosion or adhesive wear, such as scoring. Further, relative motion of the bearings rings may disturb the bearing kinematics, for example if one of the rings is dislocated in the axial direction. Shaft and housing fits shall be selected to prevent or minimize this relative motion. If this is not achievable, additional means may be required to alleviate the consequences of these effects.

Heavy interference fits as specified by the bearing manufacturers for applications with varying load and elevated vibration level shall be chosen to reduce the risk of bearing ring creep and fretting corrosion. Effects to consider when selecting the appropriate fits and tolerances include:

- condition of rotation;
- magnitude of load;
- bearing type;
- bearing internal clearance;
- temperature conditions;
- required running accuracy;
- design and material of shaft and housing;
- displacement of non-locating bearing;
- allowable hoop stresses;
- ease of mounting and dismounting.

Tight bearing fits are not always sufficient to prevent the relative motion of the bearing rings, and in some situations tight fits are not desirable (e.g., assembly issues, required axial movement of the non-locating bearing, outer rings of FPCBB bearings). In these cases, the design shall contain appropriate additional means to either reduce the relative motion, or alleviate the damage associated with this, for example by means of pins, axial clamping, adhesives or appropriate surface coatings.

When the load is stationary relative to the ring, a locating pin may be used, but positioned away from the load zone. If a pin is used, appropriate consideration shall be given to the stress concentration in the bearing ring. Design and location of the pins and corresponding slots shall be approved by the bearing manufacturer. Assumptions for coefficient of friction, creep load, pin dimension, loads on the pin and strength of the connection shall be documented.

Bores of the planet wheels should have a minimum surface hardness of 55 HRC, to avoid severe wear when there is unavoidable ring creep. Special consideration should be given to the risk of axial creep of planet bearing outer rings. Influence factors include reaction forces from helix angle, gear rim distortion, and directional textures in the surface finish.

Adhesives shall not be used as the primary means to avoid relative motion of bearing rings. In case adhesives are used as complementary means, appropriate processes for cleaning and application of the adhesives, assembly of the bearing with the adhesive and curing procedures shall be specified at the design stage. In addition, compatibility of the adhesive used with the gearbox lubricant shall be verified.

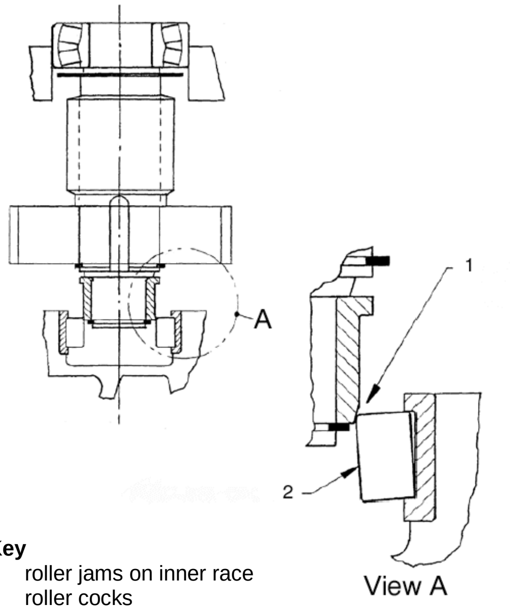
Planet bearing outer rings should be fitted to planet bores with a tolerance of at least R6. Other fits may be required depending on operating conditions and planet wheel design. Planet bearing inner rings should be clamped to avoid rotation, except when spherical roller bearings are used.

If relative bearing ring rotation is unavoidable for planet bearing inner rings, mating surfaces including support shoulders, snap rings and spacers should have minimum surface hardness of 55 HRC.

7.3.6.4 Assembly

Bearings can be easily damaged during installation. Bearings shall be installed using appropriate tools and techniques that minimize the risk of damage. General rules and recommendations provided by the bearing manufacturers for the assembly of bearings should be applied.

For example, Figure 6 shows how blind installation of cylindrical roller bearings can be especially risky. Blind assembly should be avoided.



Key
1 roller jams on inner race
2 roller cocks

IEC 2210/12

Figure 6 – Blind bearing assembly

7.3.7 Bearing lubrication

7.3.7.1 Lubricant cleanliness

Oil cleanliness is an important influence factor to reliable bearing performance. State-of-the-art filtration systems can achieve low particle count levels that are beneficial to bearing life. This is reflected by cleanliness factor e_C in the bearing calculations according ISO/TS 16281.

An oil cleanliness level of –/17/14 based on ISO 4406 shall be used when calculating bearing life for filtered systems unless operating cleanliness is demonstrated to be better (see Table 11).

It shall be verified that actual oil cleanliness in the field is one class better than the value used in bearing calculations, and that the system can maintain this oil cleanliness level, see 7.6.9.

7.3.7.2 Lubricant temperature

The viscosity ratio, κ, shall be calculated at the bearing lubricant temperature (see 7.6.6.4 and Table 6). The temperatures may need to be adjusted according to actually measured data (see 8.4.3). For the bearing rating life, viscosity ratio should be determined in accordance with ISO 281. Clause C.4 proposes more detailed methods for comparing film thickness between different lubricants.

The bulk oil temperature is the steady-state operating temperature of the oil as defined in 7.6.6.2. The oil inlet temperature is the steady-state temperature of the oil as defined in 7.6.6.3. If the bulk oil or oil inlet temperature is unknown, a temperature of 5 K below the shutdown temperature specified by the wind turbine manufacturer should be used.

Table 6 – Bearing lubricant temperature for calculation of viscosity ratio, κ

Bearing position	Lubricant temperature for splash lubricated bearings K	Lubricant temperature for pressure lubricated bearings K
High speed shaft	Bulk oil +15	Oil inlet +5
High speed intermediate shaft	Bulk oil +10	Oil inlet +5
Low speed intermediate shaft	Bulk oil +5	Oil inlet
Planet	Bulk oil +5	Oil inlet
Low speed shaft	Bulk oil	Oil inlet

The oil inlet port and oil bulk temperature shall be verified with on-site testing carried out by the wind turbine manufacturer. The results of temperature verification shall be discussed in the design validation process (see Clause 8).

The gearbox manufacturer, the wind turbine manufacturer and the bearing manufacturer shall agree on which bearings will have temperature monitoring in the turbine. They shall also agree on shutdown limits, as well as the control system actions in case these limits are exceeded.

7.3.7.3 Lubricant parameters

Lubricant parameters shall be provided by the lubricant manufacturer. Values may be approximated in accordance with ANSI/AGMA 925-A02 or GFT worksheet 3.

The EHL film thickness between the races and rolling elements, which determines the kappa value is influenced by the lubricant properties.

Two properties influencing this are as follows.

- Oil viscosity is a function of the operating temperature. The lubricant viscosity properties are usually specified by the oil supplier at 40 °C and 100 °C. Alternatively, they are specified at 40 °C with a viscosity index (*VI*) which is a measure of the change of viscosity with temperature. The oil viscosity index is typically higher for PAG than that for PAO oils, and the *VI* for PAO oils are higher than that for mineral oils.
- Oil viscosity increases exponentially with pressure. The pressure-viscosity coefficient varies with the lubricant type and viscosity and decreases with increasing temperature. The pressure-viscosity coefficient for mineral oils is typically higher than that for PAO and PAG lubricants.

With respect to viscosity behaviour, there is little difference between PAO and mineral oils for the typical wind turbine gear oil temperature range of 65 °C to 90 °C. In practice the increased reduced effect of the temperature on the viscosity (higher *VI*) of synthetic oils is compensated by the disadvantage of the pressure-viscosity coefficient (lower alpha). Thus the standard viscosity graphs of mineral oils (*VI* = 95) should be used as input for kappa computation. This, in general, is safe for many different type of oils (mineral and synthetic).

7.3.8 Rating calculations

7.3.8.1 Bearing life rating

Bearings shall be rated according to methods stated in 7.3.8.2 and 7.3.8.3. These sections provide empirical guidelines derived from field experience. Calculated bearing lives are valid for comparison of different bearing options, and may not reflect actual bearing lives under the actual service conditions.

Annex G provides an example for the documentation of the life calculation process and results.

7.3.8.2 Static rating

The static safety factor of any bearing shall be at least 2,0 at the specified extreme design load (see 6.3.4).

The static safety factor shall be calculated using the actual internal load distribution from a detailed model such as ISO/TS 16281. The ratio S_0 according to ISO 76 provides an approximation. The calculation of the internal load distribution for the planet bearings shall include a gear load offset specified by the gearbox manufacturer. The offset is the distance deviation of the axial mid-position of the gear to the resulting load impact.

7.3.8.3 Rating life

Modified reference rating life L_{nmr} shall be calculated in accordance with ISO/TS 16281. The permissible failure probability n shall be specified by the wind turbine manufacturer, but shall be equal to or less than n = 10. The calculated modified reference rating life L_{nmr} shall meet or exceed the specified design lifetime for the gearbox.

If the combined modified reference rating life L_{nmr} is greater than 10 times the combined basic reference rating life L_{nr} according to ISO/TS 16281, then the combined modified reference rating life L_{nmr} shall be set to 10 times L_{nr}.

The calculations shall be performed bin-by-bin using the load spectrum specified by the wind turbine manufacturer. Combined modified reference rating life L_{nmr} shall be calculated using Equation (3):

$$L_{nmr} = a_1 \frac{\sum q_i}{\sum \frac{q_i}{L_{10mr,i}}} \tag{3}$$

where

L_{nmr} is combined modified reference rating life in hours at (100 – n) % reliability;

n is the reliability expressed as a percentage of failures;

a_1 is the life adjustment factor for reliability per ISO 281;

$L_{10mr,i}$ is the modified reference rating life for the i^{th} load level at 90 % reliability per ISO/TS 16281;

q_i is either the time (if at constant speed) or revolutions at the i^{th} load level.

It may be required to reduce the number of bins in the load spectrum, for example to facilitate data processing when considering flexibility of the bearing environment. The method of load bin reduction shall be agreed upon by the bearing manufacturer, the gearbox manufacturer and the wind turbine manufacturer, and shall be documented together with the presentation of the life calculation results. C.2 proposes methods for reducing the number of bins in a given load spectrum.

The modified reference rating life L_{nmr} shall be adjusted for the required reliability using the life adjustment factor a_1 and include the effects of:

- radial, axial and moment loads;
- internal design of bearings;
- operating internal clearance;
- elasticity of carriers, bearings and shafts;
- load sharing between rolling elements;
- load distribution along rolling element length, considering:
  - actual roller and raceway profiles;
  - truncation of ball/raceway contact;
- lubricant viscosity at operating temperature (see 7.3.7.3);
- the effectiveness of the additive system at low κ in accordance with ISO 281;
- operating lubricant cleanliness (see 7.3.7.1);
- gear load offset;
- mesh load factor K_γ (see 7.2.3.3).

The modified reference rating life calculation may further account for elasticity of mounting. In addition, advanced calculations may be used to study the sensitivity of load sharing and load distribution against misalignment and manufacturing variation. Statistical tolerance analysis should be used to determine the misalignment. Special care shall be taken to avoid excessive stress risers at the roller ends and contact truncations.

Rating life calculation for the planet bearings shall be performed with one constant, representative gear load offset as specified by the gearbox manufacturer. The offset is the distance deviation of the axial mid position of the gear to the resulting load impact. It shall be verified by calculation that the assumed constant gear load offset is representative across the complete load spectrum.

7.3.8.4 Contact stress

The contact stress using the Miner's sum dynamic equivalent bearing load shall be documented and should not exceed the values listed in Table 7. Contact stress should be calculated in accordance with ISO/TS 16281 or C.3. Any differences between these methods should be resolved between the bearing manufacturer, the gearbox manufacturer and the wind turbine manufacturer. The results of an advanced contact analysis per ISO/TS 16281 are preferred.

Table 7 – Guide values for maximum contact stress at Miner's sum dynamic equivalent bearing load

Bearing position	Speed range $n \times D_{pw}$	Maximum contact stress p_{max} MPa
High speed shaft	150 000 to 430 000	1 300
High speed intermediate shaft	25 000 to 220 000	1 650
Low speed intermediate shaft	10 000 to 60 000	1 650
Intermediate sun shaft	10 000 to 60 000	1 650
Intermediate planet	20 000 to 150 000	1 500
Low-speed planet	10 000 to 60 000	1 500
Low speed shaft	5 000 to 15 000	1 650

NOTE 1 These guide values have been derived from experience with contemporary gearbox designs where the speed index $n \times D_{pw}$ falls within the specified ranges.

NOTE 2 The guide values apply for bearings manufactured from contemporary, commonly used, high quality hardened bearing steel, in accordance with good manufacturing practice and basically of conventional design as regards the shape of rolling contact surfaces.

NOTE 3 Values in this table are valid for a design lifetime of 20 years.

NOTE 4 Usually, there is no equivalent load available for the input shaft.

7.3.8.5 Proprietary bearing rating life

Although ISO/TS 16281 is the primary calculation method, proprietary calculation methods developed by bearing manufacturers may also be used for making comparisons. When proprietary calculation methods are used, the resulting bearing lives shall be compared to the values of L_{nmr} obtained with the ISO/TS 16281 methods. The values of basic rating life and contact stress obtained with the same proprietary methods should also be compared with the values in 7.3.8.3, Table 7 and C.3.

Differences observed while making these comparisons should be agreed to among the bearing manufacturer, the gearbox manufacturer, and the wind turbine manufacturer.

7.3.8.6 Skidding

With larger gearboxes and bearings, the potential for skidding damage to bearings is increased at low and transient load conditions common in wind turbines. Such damage can result in bearing failures earlier than rating life calculations predict. Therefore, consideration shall be given to such operating conditions when selecting bearings.

This issue shall be agreed between the gearbox manufacturer, bearing manufacturer, and wind turbine manufacturer.

The risk of skidding damage is affected by factors such as:

- bearing design and size including cage design, cage guidance, internal geometry and surface conditions of rolling element and raceways;
- speed;
- acceleration;

- rate of change from unloaded to loaded conditions;
- lubrication factors, such as quantity, viscosity, temperature and additives;
- operating internal clearance;
- size and location of the load zone including any pre-load;
- elasticity of mounting;
- effectiveness of coating or surface treatments (if applicable).

7.4 Shafts, keys, housing joints, splines and fasteners

7.4.1 Shafts

7.4.1.1 Strength and life rating

Shafts shall be designed using DIN 743 or ANSI/AGMA 6001. The design loads including dynamic effects shall be used in the strength calculations. Minimum safety factors for the different methods are specified in Table 8.

Table 8 – Minimum safety factors for the different methods

	DIN 743	ANSI/AGMA 6001
Fatigue	1,75	1,75
Yielding	1,30	1,30
NOTE 1 For ANSI/AGMA 6001 these values assume kc = 1,0 (P = 50 %) and Fp = 1,0.		
NOTE 2 DIN 743 values include the method uncertainty factor of 1,2.		

7.4.1.2 Materials

Materials for pinion shafts shall meet the requirements of 7.2. Materials for other shafts shall have sufficient hardenability to meet the requirements for strength, toughness and surface hardness. Quality requirements for these shafts shall be specified in accordance with the relevant part of ISO 683. All shafts shall be stress relieved after heat treatment.

7.4.1.3 Shaft lifting holes

In order to facilitate easy handling of drivetrain elements during servicing all shaft ends should have threaded holes designed to accept eye bolts for lifting.

7.4.2 Shaft-hub connections

7.4.2.1 Design principles

Loads on a gearbox are variable by nature, and will typically include reversing loads. Amplitude and frequency of occurrence depend on the control system response as well as on drivetrain dynamic properties.

To accommodate these loads without excessive relative movements, all types of shaft-hub connections shall have an adequate safety margin for transferring the reverse design torque only by interference without consideration of possible advantage from keys or other locking devices. The operating loads in normal load direction should be transferred by interference alone with an adequate safety margin.

The risk of fretting corrosion due to micro-movements needs to be addressed in addition to accommodating torque, shaft bending and hub deflection.

7.4.2.2 Keyless fits

Interference fits without keys (keyless fits) shall be designed per ANSI/AGMA 6001 or DIN 7190.

7.4.2.3 Interference fits with keys

Interference fits with keys shall be designed in accordance with ANSI/AGMA 6001, DIN 6892 or ISO/TR 13593.

Key slots shall not extend into the bearing journals. Intersection of key slots with diameter changes should be avoided. Edges of key slots shall be uniformly deburred or chamfered and be free of nicks, gouges or any sharp transitions that may act as stress risers.

All keys shall be made from steel with sufficient hardenability to obtain a microstructure that has the surface hardness, strength and toughness to meet the requirements of the application as determined from calculations in accordance with ANSI/AGMA 6001, DIN 6892 or ISO/TR 13593.

Geometry of key, shaft and hub shall be designed in accordance with DIN 6885-2. All keys shall be fitted to their shaft with a maximum interference fit.

7.4.3 Flexible splines

Flexible splines that transmit torque shall be designed to prevent fretting corrosion. Refer to the internal couplings clause of ANSI/AGMA 6123 for additional information on this topic.

In floating arrangements such as sun shafts, external spline teeth should be crowned to avoid edge stresses under motion. External and internal teeth should be surface hardened.

Lubrication should be adequate to prevent fretting corrosion. Pressure fed lubrication is preferred. The oil flow through the connection shall be directed to flush out all debris and return channels shall direct oil back into the sump.

7.4.4 Shaft seals

The high speed and low speed shafts shall have seals to retain the lubricant, and the appropriate elastomeric V–rings to exclude dust and moisture. Axial endplay, overheating, maintenance and their parameters must be carefully considered when designing the sealing system. Seal materials shall be compatible with the lubricant and environment specified.

Seals can have limited life, thus the type of seal and its expected life shall be discussed between the gearbox manufacturer and wind turbine manufacturer. Labyrinth seals are preferred over lip seals because lip seals have relatively short life and are difficult to replace in the turbine. Materials should not cause risk to humans or the environment.

7.4.5 Fasteners

7.4.5.1 General requirements

All fasteners shall be metric grade 8.8 or better. Fastener size, tightening torque and engagement should be in accordance with ANSI/AGMA 6001 or qualified by certified test. Wherever feasible, hardware shall be standardized to common sizes and finishes.

Where high strength fasteners of grade 10.9 or 12.9 are necessary to transfer the loads, the following quality requirements shall be fulfilled.

- If hardware made from high tensile strength steels (830 N/mm^2 and greater) is plated, strict quality procedures shall be followed to avoid hydrogen embrittlement. Hydrogen embrittlement results in low ductility and may cause nuts to split, washers to crack and bolt heads to break off.

- Purchasing of high strength hardware shall be source controlled.
- Substitutes or changes in the plating process shall be controlled.
- Each manufactured batch of hardware shall be sampled and tested using a stress/strain device.

7.4.5.2 Internal fasteners

Staking or peening is a procedure for low performance applications to seize parts together. Staking uses local yielding to keep two parts in position. This procedure is not permitted. Hardware such as set screws, bolts, nuts, pins, and fittings shall be secured.

7.4.5.3 Housing joints

All sealing compounds shall be applied in accordance with the compound manufacturer's instructions. If the housing has a split plane for gear removal, it shall be maintained oil tight. O-rings and sealing compounds shall be compatible with the lubricant. Split plane housings shall have positive locating devices such as dowel pins. Bearings shall not be used for centering split housings.

Bolted housing joints between the annulus and mating housings in epicyclic gearboxes require special design considerations to avoid motion and fretting between the members. The joint shall be capable of carrying the maximum operating load by friction under the design bolt tension with an adequate safety margin. If friction is not sufficient, the joint shall be pinned with sufficient solid pins to carry the extreme load without over stressing the housing material in compression at the pin surface. The contribution of joint friction to pin capacity shall not be counted in this calculation.

When designing a housing joint the following parameters should be considered: flatness, bolting positions, distortion, joint sealing, location of halves (repeatability of assembly), spot facing (for bolt), counter bores, use of cover for bearing retention.

7.4.6 Circlips (snap rings)

Circlips may be used as a retaining element to prevent axial displacement of the shaft assembly elements. Circlips should not be used to support operating axial loads.

External and internal circlips shall be designed in accordance with DIN 471 and DIN 472 against the extreme load. Shaft strength at the groove shall comply with 7.4.1.

7.5 Structural elements

7.5.1 Introduction

This subclause defines the strength verification and deflection analyses for structural components of a wind turbine gearbox.

A structural element is a mechanical component which is part of the main structure of a wind turbine or a gearbox. It transfers variable and non-variable loads. Typically those parts are not axis symmetric (i.e. non-rotationally symmetric).

Gearing, shafts, bearings, bolts, pins, couplings, circlips (snap rings) and keys are not considered in this subclause.

The strength verification, as described below, shall be conducted for the following components:

- torque arm;
- planet carrier;
- any other structural component transferring major loads.

Explanations for general principles on reliability for structures may be taken from ISO 2394.

For strength verification of the structural components, loads as well as interface and environment data are required. The interface loads shall be specified as described in 6.1.2 and 6.3.

The interface data are needed to quantify the transferring loads and to select appropriate boundary conditions for the stress calculations.

The interface should include:

- sketch of the drivetrain with all relevant interface dimensions;
- detailed interface geometry;
- deflection and spring rates (i.e. stiffness) of the interface;
- movements of the interfaces generated by deflections and clearances, both supplied in relation to the design loads.

7.5.2 Reliability considerations

The minimum requirements for reliability and safety shall be in accordance with IEC 61400-1. Partial safety factors are detailed and defined in 7.5.5 and 7.5.6.

7.5.3 Deflection analysis

Gears and bearings are sensitive to misalignment, and excessive deflection may cause improper function and/or interference with other components. Hence the consideration of deflection and stiffness is important in the design process of a wind turbine gearbox. See Annex D for more discussion on this.

7.5.4 Strength verification

7.5.4.1 General

Strength verification is the comparison of component stress against the allowed local stress limits. Strength verification shall be performed for both ultimate and fatigue loads.

The mechanical load on the component should be calculated using a transformation and a superposition of several load data sources (e.g. design loads, internal gear loads, static pre-loads).

The partial safety factors γ_n (consequence of failure) and γ_m (materials) shall be used according to IEC 61400-1.

The application of recognized material codes shall be in accordance with IEC 61400-1.

7.5.4.2 Determination of stress and strain

Component stress is affected by design and internal loads and by relative displacements of the interfaces. The stress level depends on loads, shape, material properties and boundary conditions.

The stress and strain shall be calculated by using analytic mathematical models or numerical analysis tools, such as FEA.

Simplifications (including those for boundary conditions) and sub-modelling may be used. However, it should be proven that the accuracy of strength assessment is not reduced by using these methods. The local stresses shall be calculated under the assumptions of linear elastic behaviour that is according to Hooke's Law. This is valid only as long as the yield stress for the material is not exceeded.

In some cases changes in the boundary conditions due to loading lead to non-linear behaviour. These effects can be linearized in a consistent manner where appropriate. The generated stress field should be proportional to the applied load level.

The stresses are calculated with Equation (4):

$$\sigma = c \cdot L \tag{4}$$

where

σ is stress;

L is the applied load.

Further, it is permitted to assume a stress field, ${}^{e}\sigma_{ij}$, that arises out of a linear superposition of stress fields generated by the individual load cases (e.g. external loads, internal loads, pre-tension).

$${}^{e}\sigma_{ij} = \sum_{k} c_{ij,k} \cdot L_k \tag{5}$$

7.5.4.3 Finite element analysis

Finite Element Analysis (FEA) is based on models consisting of discrete elements with boundary conditions such as loads, supports and constraints. Constraints are geometrical boundary conditions which define displacements and rotations. The applied constraints may have a significant influence on the accuracy of the results. To achieve a sufficient accuracy it may be necessary to expand the system to include the neighbouring components instead of defining conditions at the boundary of the component under study.

The FEA model shall be accurate enough to cover physical effects such as peak stresses or strains due to notches and high stiffness gradients. Mesh sensitivity analyses should be performed.

7.5.4.4 Loads and displacements

External forces, moments and displacements determine the predominant stress on a structural component. Additional influences should be taken into account if they increase the stress in the component significantly (e.g. shrink fit or bolt pre-load).

7.5.4.5 Material properties

Material data should be taken from internationally recognized standards or data from material tests according to 7.5.7. Typical properties of materials commonly used in FEA calculations of structural components are listed in Annex D.

7.5.5 Static strength assessment

7.5.5.1 Ultimate and yield strength

The static strength assessment shall be in accordance to this standard or other internationally recognized codes. Static strength assessment should be performed for those load combinations which result in the highest local stress. Further information about the limit state function is given in IEC 61400-1.

The safety of the structural component is defined by the load reserve factor for ultimate strength (or yield strength where applicable), LRF_u, which is calculated by Equation (6):

$$LRF_u = \frac{\sigma_{\text{lim}}}{\sigma_{\text{max}} \cdot \gamma_n} \geq 1 \tag{6}$$

where

σ_{max} refers to the maximum linear elastic stress

σ_{lim} is defined in 7.5.5.4.

Partial safety factors are used to calculate the limiting stress levels. The partial safety factor for material (γ_m), shall be used as listed in Table 9. The partial safety factor for the consequence of failure (γ_n), is defined in IEC 61400-1.

Table 9 – Partial safety factors for materials

Characteristic strength	Partial material factor
R_p	$\gamma_{m\_Rp} = 1{,}1$
R_m	$\gamma_{m\_Rm} = 1{,}3$

7.5.5.2 Stress hypothesis for ultimate or yield strength

The dimensioning of a structure or component depends on the type of possible failure. The static strength assessment should be carried out using a stress hypothesis appropriate to the type of failure expected.

- For brittle materials the behaviour of the material is described by the maximum principal stress hypothesis. Both maximum and minimum principal stress shall be considered in the strength verification.
- For ductile materials, the maximum shear strain energy hypothesis (e.g. von Mises) or the maximum shear stress hypothesis (e.g. Tresca) describes the failure mechanism.

Alternative hypotheses may be applied if they are proven by component tests.

7.5.5.3 Influence of size

Yield and tensile strength are size dependent. When the size effect is not experimentally quantified, the influence of size on the material strength shall be taken from standards or determined by state-of-the-art codes, for example Gudehus and Zenner (1999) or Wegerdt, Hanel, Hänel, Wirthgen (2003).

7.5.5.4 Limiting stress level

The limiting stress σ_{lim} is the maximum local stress that the component can withstand. The characteristic material values shall be taken from standards or determined by measurements carried out according to ISO standards.

Standard method:

For several materials, the compressive strength is significantly higher than the tensile strength. These elevated strength values ($R_{m,\text{compressive}}$ and $R_{p,\text{compressive}}$) can be applied when applicability is justified.

For the standard method, the limiting stress level shall be calculated according to Equation (7) for structural steel and spheroidal graphite cast iron (nodular) and Equation (8) is to be used for grey (lamellar) cast iron.

$$\sigma_{\text{lim}} = \text{Min}\left[\frac{R_p}{\gamma_{m\_Rp}};\frac{R_m}{\gamma_{m\_Rm}}\right] \tag{7}$$

$$\sigma_{\text{lim}} = \frac{R_m}{\gamma_{m\_Rm}} \tag{8}$$

where:

R_p is the yield point.

Advanced method:

If the yield stress is exceeded locally, the notch stress and strain will have a non-linear relation to the load level due to local yielding. The local strain has to be considered for assessment of the static strength on the component. Here it must be taken into account that the local strain distribution depends on the material, the component shape (i.e. notch) and on the load level. This effect is expressed by the section factor $n_{pl,\sigma}$, and Figure 7 shows conditions at the point of failure using linear assumptions and including plasticity.

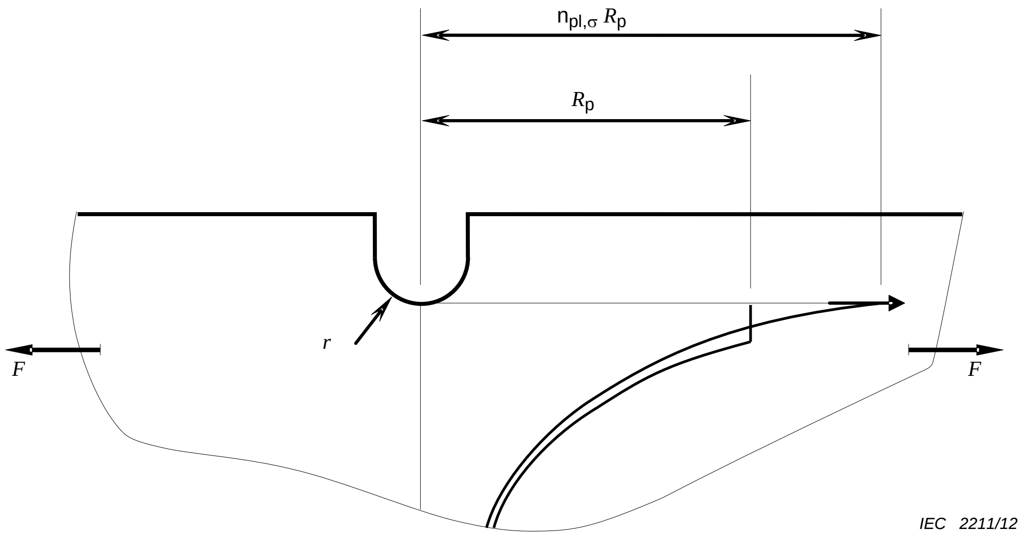


Figure 7 – Definition of section factor $n_{pl,\sigma}$ of a notched component

The section factor $n_{pl,\sigma}$ takes into account the influence of stress distribution on the component strength and allows the use of plastic reserves of the material. It shows how much the linear elastic calculated notch stress, σ_{el}, could be increased in relation to the yield strength. The factor accounts for both local and global failure, and it is defined in Equation (9).

$$n_{pl,\sigma} = \min(n_{pl,\sigma,LF}; n_{pl,\sigma,GF}) \geq 1 \tag{9}$$

The total strain limit is ε_{lim} = 1 %. For the advanced method the limiting stress level is calculated with Equation (10).

$$\sigma_{\text{lim}} = \frac{R_p \cdot n_{pl,\sigma}}{\gamma_{m\_Rp}} \tag{10}$$

Unless proof or verification is provided, this advanced method should not be used for:

- surface hardened components (by chemical-thermal or thermal methods);

- lamellar cast iron, malleable cast iron and nodular cast iron with an ultimate strain A3 or A5 $<$ 8 %.

If plastic stress is exceeded and the area is subject to fatigue loading the classical concept of linear damage accumulation may no longer be applicable.

7.5.5.5 Global and local failures

Local failure occurs when the maximum allowable strain (ε_{lim}) is exceeded at a critical location. A component does not always fail due to local stress at a notch. A global failure may occur at unnotched sections or sections that are less severely notched. Further explanation is given in Annex D.

According to Wegerdt, Hanel, Hänel, Wirthgen (2003), the section factor for local failures, $n_{pl,\sigma,LF}$, is calculated with Equation (11).

$$n_{pl,\sigma,LF} = \sqrt{\frac{\varepsilon_{lim} \cdot E}{R_p}} \tag{11}$$

where

E is the modulus of elasticity of the material.

A global failure occurs when the total plastic load limit is exceeded and permanent deformation occurs. It depends on the material, the part shape and the type of loading. The total plastic load limit L_p shall be evaluated with elastic plastic FEA by using an idealized elastic perfect plastic stress-strain curve. Figure 8 shows an example.

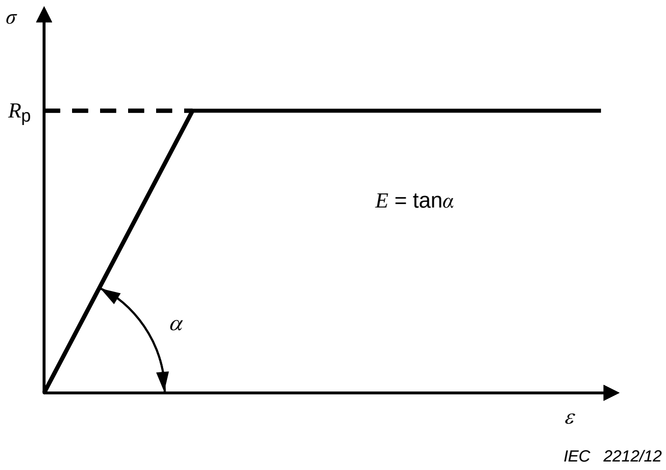


Figure 8 – Idealized elastic plastic stress-strain curve

The section factor for global failures is calculated with Equation (12).

$$n_{pl,\sigma,GF} = \frac{L_p}{L_e} \tag{12}$$

where

L_e the elastic load limit, the load at which the local stress in the notch exceeds the yield point R_p;

L_p is the total plastic load limit, load at which the entire section starts yielding.

7.5.6 Fatigue strength assessment

7.5.6.1 Fatigue strength methods

The fatigue strength analysis of structural components in wind turbine gearboxes should be performed according to this standard. Other internationally recognized analysis codes may be applied, if their usefulness for the applied material and specified load conditions is proven. Guidance for fatigue analyses using damage accumulation can be taken from Dowling (1972), Haibach (2006), Matsuishi and Endo (1969) or Wegerdt, Hanel, Hänel, Wirthgen (2003).

The analysis procedure is based on the local stress approach. Three methods (method A, B and C) for fatigue-strength analysis are described below.

In all cases, the fatigue strength should be described using the accumulated damage, D, and the load reserve factor for fatigue, LRF_f.

It shall be stated explicitly in the calculation whether amplitude (semi-range) or range is used as the measure of fatigue stress magnitude, and any non-conforming data shall be converted appropriately. The use of amplitudes is assumed here.

Method A:

In method A, the local effective uniaxial stress is calculated for each time step in the verified fatigue load time series. The stress cycles of the resulting stress history should be counted and classified into a two dimensional rainflow count matrix with mean and amplitude value. The damage of each bin in the matrix should be calculated and accumulated according to the modified Miner’s rule (see 7.5.6.7). Method A is recommended for load conditions such as

- multi-axial loading;
- where load components vary independently from each other.

Method B:

In method B, the local stress is calculated based on the two dimensional rainflow count matrix with amplitude and mean value of load. Method B can only be applied when a uniaxial load is acting on the structural component. Constant stresses (e.g. shrink fit) could also be taken into account. Method B shall not be used if the resultant stress is not approximately uniaxial. The damage of each element in the matrix should be calculated and accumulated according to the modified Miner’s rule (see 7.5.6.7). Method B may be sufficient for load conditions such as

- uniaxial loading;
- pre-loading.

Method C:

In the simplified method C, the local stress level is calculated for the equivalent load. The local stress level is compared to the allowable stress level (with an assumed mean stress) in order to obtain the load reserve factor (see 7.5.6.7). Method C can only be applied for load conditions such as:

- uniaxial load;
- no pre-loading.

7.5.6.2 Determination of local stresses

The local stress level (i.e. the component stress) will be determined as described in 7.5.4.2. All fatigue design loads shall be taken into account according to IEC 61400-1.

Determination of local stresses for method A:

The stress history for a system point s is calculated based on superposition of the stress tensor components ${}^{e}\sigma_{ij}(s,t)$, assuming that the material behaviour is linear elastic. They are a result of a linear combination of the dimensioned transfer coefficient $c_{ij,k}(s)$ and the relevant time series $L_k(t)$, as discussed in 7.5.4.2.

$$ {}^{e}\sigma_{ij}(s,t) = \sum_{k} c_{ij,k}(s) \cdot L_k(t) \tag{13} $$

The transfer coefficients $c_{ij,k}(s)$ are determined for each load case k taken from an elastic stress analyses with an appropriate load, e.g. $|L_k|$ = 1 (see Haibach (2006)).

The actual stress cycles should be calculated by means of rainflow counting of the stress history, as per Haibach (2006) or SAE (1997).

Determination of local stresses for method B:

The local stress level on the structural component is calculated for a load with a unity load level. The load levels in the rainflow count matrices are transformed to stress levels. Stresses generated by static loads (e.g. pretension) are added to the mean-stress levels.

$$ {}^{e}\sigma_{ij,a}(s) = c_{ij}(s) \cdot L_a \tag{14} $$

where

L_a is the load amplitude of each bin.

$$ {}^{e}\sigma_{ij,m}(s) = c_{ij,m}(s) \cdot L_m + \sigma_{ij,\mathrm{pre}}(s) \tag{15} $$

where

L_m is the load mean value of each bin;

$\sigma_{ij,\mathrm{pre}}(s)$ is the local pre-stress tensor.

Determination of local stresses for method C:

The local stress amplitude on the structural component is calculated for the equivalent load level.

$$ {}^{e}\sigma_{a,eq}(s) = c(s) \cdot L_{a,eq} \tag{16} $$

where

$L_{a,eq}$ is the equivalent load amplitude, at a stated reference number of cycles n_{ref}, corresponding to the appropriate inclination exponent k of the SN-curve for the material in question.

7.5.6.3 Stress hypothesis for fatigue

The selection of the stress hypothesis depends on the loading and material type of the component.

The applicable stress hypotheses for uniaxial stress conditions are:

- for brittle materials and cast iron the maximum principal stress hypothesis shall be used;

- for ductile material several different stress hypotheses are known. These methods should be used if the applicability is justified.

In case of multiaxial stress, the influence of the time depending direction of the maximum principal-stress on the damage accumulation and material type has to be taken into account. Several different stress hypotheses for multiaxial stress conditions are known, in some cases non-standard material data is needed. However, these methods should be used if the applicability is justified.

The principal stress shall be calculated by

$$\sigma_{\text{prin}} = \begin{cases} \sigma_I \text{ if } |\sigma_I| > |\sigma_{III}| \\ \sigma_{III} \text{ if } |\sigma_{III}| > |\sigma_I| \end{cases} \text{ for } \sigma_I > \sigma_{II} > \sigma_{III} \tag{17}$$

7.5.6.4 S/N curves

The S/N curves, such as Figure 9, characterize the fatigue strength of the material in terms of stress amplitude versus load cycles to failure.

In principle, statistically assured S/N curves for the component material should be used as a basis. If such S/N curves are not available synthetic S/N curves in accordance with Haibach (2006), Gudehus and Zenner (1999), Wegerdt, Hanel, Hänel, Wirthgen (2003) or similar authoritative reference should be used for fatigue analyses.

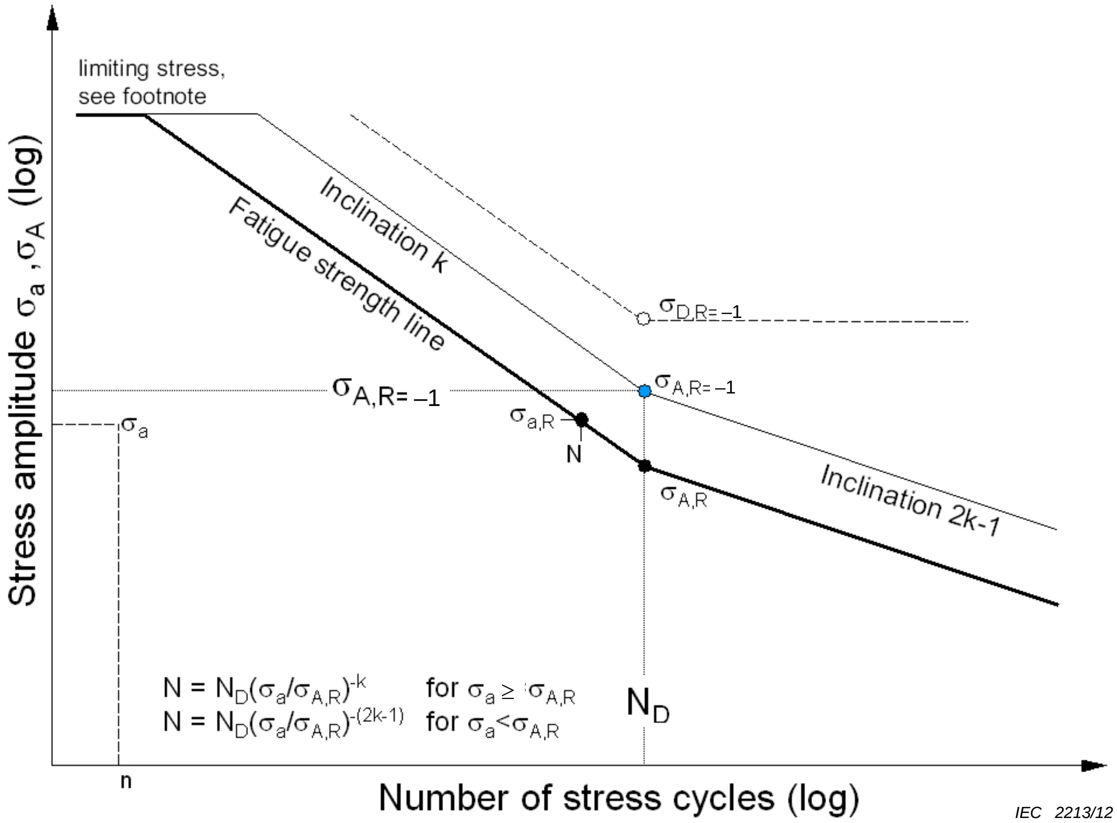


Figure 9 – Synthetic S/N curve (adapted from Haibach, 2006)

For stress cycle numbers $N_i > N_D$, the S/N curves shall be extended from $\sigma_{A,R}$ with the inclination (slope) parameter 2*k*-1, where *k* is the inclination parameter of the fatigue strength line, Haibach (2006). Here the limiting stress cycle number N_D is the number of cycles at which, under optimum test conditions (no corrosion effect, etc.), the knee point σ_D of the constant amplitude S/N curve is defined.

Appropriate limits are determined for both the occurring stress and the stress amplitude (see D.5).

7.5.6.5 Influences on fatigue strength

The following influences on the fatigue strength and reduction factors shall be considered in the calculation of stress amplitude or stress range:

- stress gradient;
- surface roughness;
- surface treatment;
- mean stress influence (an illustrative method is given in Annex D);
- size influence;
- partial factor for material;
- partial factor for the consequence of failure.

NOTE Partial factor for the consequence of failure are removed from the loads.

7.5.6.6 Partial safety factors for fatigue

7.5.6.6.1 Partial safety factors for steel

The partial safety factor γ_m for material for steel shall be taken from IEC 61400-1.

7.5.6.6.2 Partial safety factors for cast iron

The partial safety factor γ_m for cast iron in Table 10 takes account of casting flaws depending on the required local cast quality for synthetic S/N-curves.

The quality level on a component should be verified by non-destructive testing for the determined hot spots. The area to be tested shall be identified in the appropriate manufacturing drawings, either as-cast or as-machined.

Table 10 – Partial safety factors γ_m for synthetic S/N-curves of cast iron materials

Material behaviours	γ_m		
Cast quality class according to EN 12680-3	1	2	3
γ_m (P = 50 %)	1,5	1,75	2,0
NOTE These safety factors assume testing for surface and volumetric flaws			

7.5.6.6.3 Partial safety factor determined from specimens

Test data for the allowable material characteristics on specimens taken from actual components shall be statistically assured according to ISO 5725-2 or adequate methods. The S/N-curves shall be determined for a survival probability of 97,7 %. The partial material factor shall be applied with γ_m = 1,1. The test shall be carried out by accredited test laboratories according to ISO 17025.

7.5.6.6.4 Partial safety factors for consequence of failure

The partial safety factor for the consequence of failure (γ_n) is specified in IEC 61400-1.

7.5.6.7 Damage accumulation

For methods A and B (see 7.5.6.1), the damage shall be calculated according to the modified Miner's rule (Haibach 2006). The damage shall be less than or equal to unity as per Equation (18).

$$D = \sum_{i=1} D_i = \sum_{i=1} \frac{n_i}{N_i} \leq 1 \tag{18}$$

where n_i is the number of cycles in bin "i" and N_i is the number of cycles on the S-N curve corresponding to the stress amplitude of bin "i" according to Figure 9.

The load-reserve factor for fatigue LRF_f expresses the fatigue strength of the structural component, and can be approximated as per Equation (19).

$$\mathrm{LRF_f} \approx \left(\frac{1}{D}\right)^{\frac{1}{k}} \geq 1 \tag{19}$$

The limiting criterion for method C is a load reserve factor for fatigue LRF_f which shall be larger than unity:

$$\mathrm{LRF_f} = \frac{\sigma_{A,R}}{\sigma_{a,eq}} \left(\frac{N_D}{n_{\mathrm{ref}}}\right)^{\frac{1}{k}} \geq 1 \tag{20}$$

The stress amplitude $\sigma_{a,eq}$ shall be calculated by the stress hypothesis for uniaxial stress conditions as described in 7.5.6.3. For the determination of L_{eq} and σ_a the exponent *k* shall be equal to inclination *k* of the S/N curve. For method C, information about the mean stress due to external loading is not known, therefore a stress ratio of R = 0,5 shall be assumed. It should be noted that at high mean load a stress ratio of 0,5 may not be conservative. In this case the mean stress influence shall be considered in a conservative manner.

7.5.7 Material tests

Material test documents (statements of compliance) should correspond to ISO 10474 or equivalently EN 10204 and be specified in the quality plan as agreed between wind turbine manufacturer and gearbox manufacturer. These documents should contain the results of the tests laid down in the standards or additionally agreed or demanded on the basis of the requirements.

Inspection certificates in accordance with ISO 10474 should be submitted for the materials of those components that are subjected to high static or cyclic loads and that are important for the integrity of the gearbox.

The static material characteristics (e.g. yield and tensile strength) are experimentally determined according to international standards.

7.5.8 Documentation

7.5.8.1 General documentation

From the technical and calculation viewpoint, the documentation of the computational analyses shall form a unified whole together with all other documents (drawings, specifications, etc.). Here it shall be observed that the references of the adjacent components and structural areas used in the computational analyses shall be included in the documentation.

7.5.8.2 Determination of stress and strain

The presentation of the inputs for the influences on the mechanical structure model should be clear. Apart from the loads and deflections, influences include temperatures, pre-stresses and, if applicable, imperfections.

All boundary conditions (e.g. constraints, material properties and the applied loads) shall be documented in a plausible manner. All geometric simplifications; neglected drill-holes, cut-outs, radii etc. shall be stated and justified with reference to the results.

The reference (e.g. norm, code, tests) of the material data shall be documented. If FEA programs are used, the developed models shall be documented in detail.

7.5.8.3 Static strength assessment

The results of the extreme loads analysis shall be presented, with comments, preferably as a combination of the unit loads. It shall be ensured that there is correspondence between the loads and their representation in the graphics/lists.

For static strength assessments that are non-linear with respect to materials, the local strain shall generally also be determined and assessed in addition to the local elastic-plastic stress.

For the assessment, the load-reserve factors shall be described.

7.5.8.4 Fatigue strength assessment

The results shall be presented with consideration of the provisions described in 7.5.6 and with reference to the global FE model or previously selected regions (e.g. critical cross-sections, stress concentrations).

The stresses shall be compared to the permissible values. The permissible stress levels should be documented clearly (e.g. material characteristics, material quality levels) and references should be indicated. Further notes, particularly on special fatigue verifications, are also given in 7.5.6.

The classification into quality levels shall be documented consistently in the drawings, calculations and specifications.

For the assessment, the load reserve factors shall be described.

7.6 Lubrication

7.6.1 General considerations

The selection, application and monitoring of a lubricant with appropriate performance characteristics is essential for optimum service life and performance of the WTG. The specification of these minimum performance characteristics for use in the wind turbine gearbox requires a careful assessment of many different and partly opposing factors, for example:

- turbine power rating;
- gearbox design and configuration (gearing type, stages, bearing types);
- range of ambient conditions (site, temperatures);
- range of operating conditions (peak and average speeds and loads, temperatures);
- start-up conditions (low temperature issues);
- lubrication system design (oil flow, lubricant volume);
- filtration systems (pumping requirements, filter media, micron rating);
- lubrication delivering system (splash versus pressure fed);
- expected length of service;
- health, safety and environmental conditions.

From these factors, the appropriate characteristics (physical, chemical and performance) of the lubricant that will provide an optimum balance of the requirements of a given application can be determined. Key elements for consideration in the lubricant composition are:

- base oil type (mineral, synthetic, or semi-synthetic blends composed of mixtures of highly refined mineral oils and synthetic fluids);
- physical and performance enhancing additives;
- base oil type (mineral, synthetic or blends);
- thickening agents (optional); and
- performance package.

The minimum performance requirements for the lubricant shall be specified by the gearbox manufacturer. The wind turbine manufacturer should specify additional requirements for components of the oil circulation system not supplied by the gearbox manufacturer as well as for specific operation conditions and maintenance schedule. The selected brand or type of lubricant shall be agreed to between the wind turbine and gearbox manufacturer.

The minimum performance requirements shall at least cover:

- characteristics defined in DIN 51517-3, ISO 12925-1 and AGMA 9005 as a minimum;
- minimum properties and performance characteristics defined in 7.6.2 and 7.6.3;
- performance characteristics for gear contacts;
- performance characteristics for bearing contacts;
- additional requirements for components of the oil circulation system;
- environmental requirements;
- requirements for special operation conditions;
- maintenance schedules.

It shall be documented that the selected brand or type of lubricant meets these specified minimum requirements. Consistency between documented lubricant properties and assumptions in the component rating calculations as well as compatibility of gearbox components with the lubricant shall be confirmed during the design and acceptance process.

7.6.2 Type of lubricant

Wind turbine transmission gears operate at low to moderate pitch line velocity with high to very high contact loads. These conditions require the use of lubricants fortified with performance enhancing additives and of the highest practical viscosity. The base fluids of these lubricants should be chosen from highly refined mineral oils, full synthetic fluids, or semi-synthetic blends (mixtures of highly refined mineral oils and synthetic fluids). The choice of a finished lubricant depends on many factors including viscosity, viscosity index, pour point, additives, and overall lubrication costs. Site specific operating conditions, wind turbine performance and serviceability influence the selection of the most cost effective gearbox lubricant. Parameters such as cold start and operating temperature within the nacelle should be closely monitored. Lubricant type and viscosity should be appropriate for the operating conditions.

7.6.3 Lubricant characteristics

7.6.3.1 General

Most of the large modern wind turbines ≥500 kW are equipped with multistage transmissions to convert the low speed rotor input to the higher speeds required to run the output generator efficiently. Ideally, each stage of the transmission would benefit from a different viscosity fluid, but this is not practical. Additionally, the gears and bearings in each stage would benefit from different performance chemicals such as higher anti-scuff (also known as EP) levels at the output and perhaps higher anti-wear at the input stages. Oxidation stability is important because of the potential increase in the risk of deposit formations such as varnish and sludge that can plug filters, small oil passages and oil spray nozzles, as well as create deposit on critical surfaces. Therefore, some compromise in the choice of final lubricant characteristics may need to be made.

7.6.3.2 Viscosity

Viscosity is the most important physical property of a lubricant. The viscosity, at operating conditions of temperature, load, and velocity, has direct impact on gearbox performance and durability. The correct viscosity at cold start-up is important to achieve adequate lubricant flow to all critical surfaces without channelling or creating excessive drag. The correct viscosity at operating temperature is required to ensure adequate film thickness which minimizes metal to metal friction and wear without contributing to excess parasitic losses such as churning by the gear, or fluid friction drag in bearings. Maintaining proper viscosity over the entire operating temperature range of the gearbox may help to minimize the potential for foaming and air entrainment. Excessive parasitic losses elevate the operating temperature of the gearbox, which increases the oxidation rate of the lubricating oil. The useful service life of a lubricant is reduced when its oxidation rate is increased.

Selecting the correct viscosity grade of the lubricating oil for a gearbox should be based on operating, not start-up conditions. Guidelines found in ISO 12925-1 or AGMA 9005 can be used to select the correct viscosity grade based on viscosity index and bulk oil operating temperature. If the correct viscosity at operating temperature results in excessive viscosity during cold start-up, several options are available. Many lubricant suppliers can provide gear oil with enhanced low temperature properties such as very high viscosity index and low dynamic viscosity. Low temperature gear oil can be formulated from synthetics, highly refined mineral oils, or a blend of these. Another option is to incorporate a low power density heater in the gearbox sump or a space heater within the nacelle to warm the lubricating oil to achieve the proper start-up viscosity. Specific surface energy of the sump electric heater shall be limited in order to prevent degradation of the oil.

The operating conditions can be site specific. It is the responsibility of the wind turbine manufacturer to accurately inform the gearbox manufacturer(s) and lubricant manufacturer(s) of the anticipated ambient conditions for each wind turbine. The gearbox manufacturer should include the proper viscosity grade of the lubricating oil on gearbox nameplates and in installation/maintenance manuals. The lubricant manufacturer should provide a chart listing the viscosity of the selected product at various temperature ranges from the lowest anticipated cold start to the highest anticipated operating temperature.

7.6.3.3 Performance characteristics

As part of the lubricant selection process the oil should satisfy selected performance requirements which are critical to maximizing the life of the transmission. This is primarily a function of the chemical additive system used in the lubricant. Additives are essential for obtaining predicted gearbox design life. Standardized laboratory tests are used to evaluate individual and overall additive effectiveness. Unfortunately, not all of the laboratory tests are available as standardized methods.

To be considered adequate for service, the lubricant should meet minimum acceptance levels established by the wind turbine manufacturer and component suppliers for each test in the following areas:

- gear scuffing;
- bearing wear and bearing fatigue at mixed friction;
- oxidation of oil;
- corrosion protection (ferrous and non-ferrous);
- foaming and air release;
- shear stability;
- elastomer compatibility.

Acceptance of lubricant performance shall only be considered on the basis of using standardized test methods available in the public domain. The standardized test method shall be defined and sanctioned by an accredited technical organization such as ISO, CEC or ASTM and shall include repeatability and reproducibility data for the method. In-house and proprietary test methods shall not be used to establish acceptability of lubricants for wind turbine applications under this standard although they may be used to evaluate the oil properties. A list of standardized test methods and acceptance levels is shown in Annex E.

There are a number of non-standardized tests that are required by some equipment manufacturers in the industry to help define the performance of the lubricant for this application such as:

- gear wear (abrasive, adhesive, fatigue (micro- and macro));
- filterability;
- compatibility with materials of construction (ferrous and non-ferrous metals, paints, coatings, elastomers, seals and sealants, etc.);
- compatibility with auxiliary and peripheral components (filter media, desiccant used in breather vents devices, electronic sensors and connectors, etc.);

These non-standardized tests are shown in Annex E for informational purposes only. Some of these are currently being reviewed for standardization.

7.6.4 Method of lubrication

7.6.4.1 Types

Splash (or dip) and pressure fed are the two main types of lubrication systems used in wind turbine gearboxes larger than 500 kW. Except for dry sump gearboxes, these two types of lubrication system may be used together. Lubricant viscosity affects the design of the system from pump size to delivery tubing. The specifications of the lubrication pump and lubrication system(s) shall be agreed upon between the turbine and gearbox manufacturers. Depending on configuration, features such as pump location, type of tubing, hoses, fittings, type of filter element, filter change interval and quantity of oil in reservoirs may be specified. Additional means may be necessary to provide minimum oil supply while the turbine is off-line or without grid power for an extended period of time.

7.6.4.2 Splash lubrication

Splash lubrication does not use a pump to supply oil to gear meshes and bearings. It depends on the gears to direct oil to channels which lubricate the bearings. The low speed gear should dip into the oil bath for at least twice the tooth depth to provide adequate splash for gears and bearings. The oil level should be designed to minimize churning while providing adequate lubrication to all bearings and gears. The gear housing should have troughs to capture the oil flowing down the housing walls, channelling the oil to the bearings. Splash systems shall have an offline filtration system to control contamination and prevent the distribution of particles to critical gear and bearing surfaces. The offline filtration system shall be designed to maintain an oil cleanliness level one class better than the assumption made in bearing life calculations (see 7.3.8).

7.6.4.3 Pressure fed lubrication

Gearboxes rated at ≥ 500 kW shall be lubricated by an oil circulation system fitted with inline filters, offline filters or a combination of both to remove particulates and maintain cleanliness at agreed levels. Pressure fed systems can also have a heat exchanger to cool the oil. These systems should ensure adequate lubrication of all rotating elements and prolong the life of the lubricant and components. To ensure adequate lubrication and control lubricant temperature, the system should be properly designed considering viscosity, flow rate, feed pressure, and the size, number and placement of the jets. All bearings except those that submerge into the sump operating oil level shall be fed by the circulation system. The wind turbine control system can, during idling or parking, activate a pressure fed system periodically to minimize the risk of damage caused by lack of lubricant.

Pressure fed systems typically deliver oil to gears using spray nozzles or an unrestricted tube. Pressure fed bearings are typically supplied with fresh oil flow by feed lines. The pressure level in the oil distribution system and pressure drop at the orifices shall be selected as low as possible to reduce entrained air. Pressure lubrication may be required at high circumferential speeds.

Spray nozzles and manifolds should be accessible for inspection and replacement. If internal tubing has threaded components, they should be accessible for tightening. Spray nozzles should be protected from clogging by accessible inline filter screens.

Pressure fed systems can either operate with a wet or dry sump. The gears and bearings do not dip into the oil in a dry sump system.

7.6.4.4 Combined lubrication systems

Combined lubrication systems utilize both splash and pressure fed lubrication methods to ensure adequate oil is available to gears and bearings on all shafts over a wide range of operating viscosity. Oil filters and heat exchangers may be integrated in this system. Such systems allow for smaller sizes of pumps and oil lines, since they must only be dimensioned for low viscosity. Due to the reduced filtration time with such a system, it may be necessary to install a secondary filtration circuit (offline filter – see Annex E) to maintain the required cleanliness.

7.6.5 Oil quantity

Minimum quantity of oil in the lubrication system should be:

$$Q_{\mathrm{oil}} = 0{,}15 \cdot P_{\mathrm{el}} + 20 \qquad (21)$$

where

Q_{oil} is recommended oil quantity, in litres;

P_{el} is rated power of wind turbine, in kW.

This recommendation is based on experience with typical multistage gearboxes up to 2 MW where the gear housing forms the oil reservoir. Therefore, the minimum oil quantity should be reviewed if the design is not a multistage gearbox, is larger than 2 MW, or uses a separate reservoir.

7.6.6 Operating temperatures

7.6.6.1 Temperature measurement

Operating temperature is an important parameter that is used to provide a closer approximation of the actual state of the lubricant with respect to its effective viscosity and film thickness. However, one must be aware of where the measurement was taken as the temperature of the lubricant varies within the gearbox. Three different oil operating temperatures are considered: bulk oil, gear mesh, and bearings. At least one of these temperatures shall be monitored.

7.6.6.2 Bulk oil temperature

This is the temperature of the oil that is representative of the overall volume of the lubricant within the lubrication system. With splash lubricated or intermittent pressure fed lubrication system gearboxes, it is measured in a central area of the gearbox sump. This measurement should be made in a relatively large pool of oil away from stagnant areas.

With pressure fed lubrication systems, this is the temperature of the oil within the pressure line between the oil pump and filter assembly during system operation.

7.6.6.3 Oil inlet temperature

The oil inlet temperature is the steady-state temperature of the lubricant where it enters the gearbox in pressure fed or combined lubrication systems.

7.6.6.4 Bearing lubricant temperature

This is the temperature of the oil as measured in the vicinity of the rolling element bearing, typically the temperature of the oil supplied to the bearing. The accuracy of the bearing lubricant temperature measurement is increased by making the measurement as close to the rolling elements as possible. This is separate from the bearing temperature that is measured on the stationary ring of the bearing.

7.6.7 Temperature control

7.6.7.1 General

Gearbox operating temperature shall be controlled through all phases of operation. If necessary, heaters and coolers should be used to control gearbox temperature. Controls should be set as specified in 7.6.7.2 and 7.6.7.3.

7.6.7.2 Bulk oil temperature

Maximum bulk oil temperature above ambient and maximum absolute bulk oil temperature shall be specified and controlled. The specified limits shall correspond to the values used in rating calculations for gears and bearings (see 7.2 and 7.3). As an absolute maximum, controls shall be set to shutdown the turbine when the 10 min average bulk oil temperature exceeds 85 °C.

The bulk oil should be maintained at or 5 K above the temperature at which the lubricant will circulate freely prior to start-up. A heat source may be needed to achieve this. The gearbox shall be equipped with appropriate monitoring devices to ensure that the lubrication system provides adequate flow to critical components such as gears, bearings, splines, filtration devices, and external oil reservoirs during periods of cold weather. To meet these requirements, the overall gearbox and any external systems may need to be heated. This could be done by a number of methods including:

- air heating in the nacelle;
- heat exchangers in a pressurized lubrication system;

- ensure that upper bearings have sufficient lubrication during such controlled idling speed;
- wrapping external lubrication lines with resistive heating devices;
- continuous circulation of oil to maintain fluidity and utilize viscous heating of the lubricant.

7.6.7.3 Bearing temperature

Maximum bearing temperature above ambient and above the oil sump temperature as well as maximum absolute bearing temperature shall be specified.

The specified limits shall correspond to the values used in the calculation of bearing rating life (see 7.3.8). As absolute maximum, controls shall be set to shutdown the turbine when the bearing outer ring one minute average temperature exceeds 105 °C. Maximum permissible continuous bearing temperature measured at the bearing outside diameter shall not exceed 95 °C. This temperature limit may need to be reduced for some lubricants.

7.6.8 Lubricant condition monitoring

Lubricant condition monitoring intervals and tests shall be specified in the operation and maintenance manuals. The tests to verify lubricant performance characteristics shall include those recommended by the lubricant manufacturer. Tests shall at least include:

- oil cleanliness;
- viscosity;
- water content;
- wear metals;
- measure of oil oxidation;
- key metallic and non-metallic elements in the additive package.

The limits associated with the foregoing lubricant condition characteristics shall be agreed upon among the turbine manufacturer, component suppliers and lubricant supplier based on input from the wind turbine manufacturer on operating and maintenance conditions. Refer to Annex E for further guidance. The wind turbine manufacturer is responsible for preparing the operation and maintenance manual in accordance with the requirements specified by the gearbox manufacturer.

The operation and service manual shall specify the oil sampling requirements necessary to achieve representative and consistent samples for analysis (see Annex E). For example, this may include specifying deactivation of the filtration system as soon as the wind turbine has stopped prior to taking the sample. The first sample should be drawn from the gearbox within one month of commissioning. The second sample should be drawn from the gearbox not later than 1 000 operating hours after commissioning of the wind turbine. Six months has proven to be a suitable interval for subsequent samples.

7.6.9 Lubricant cleanliness

For maximum gear and bearing life, the lubricant must be as clean as possible. Some guidelines for maintaining cleanliness are:

- filter new lubricant before adding to gearboxes;
- determine oil cleanliness after factory test to determine assembly cleanliness;
- filter oil in service and maintain an oil cleanliness level as agreed between the turbine manufacturer, component suppliers and end user;
- monitor to detect contamination or other adverse changes to lubricant (see 7.6.8).

The cleanliness of the lubricant is determined according to ISO 4406. The required cleanliness level shall be agreed between the wind turbine manufacturer, the gearbox manufacturer, and the bearing manufacturer. The steady state cleanliness level for a gearbox in constant operation shall not be worse than -/17/14. Table 11 provides recommendations for the cleanliness depending on the sample source.

Table 11 – Recommended cleanliness levels

Source of oil sample	Cleanliness code according to ISO 4406
Oil added to gearbox at any location	- / 14 / 11
Bulk oil from gearbox after factory test at the gear manufacturer's facility	- / 15 / 12
Bulk oil from gearbox after having been in service within one month after commissioning of the wind turbine	- / 15 / 12
Bulk oil from gearbox sampled per the maintenance schedule	- / 16 / 13
NOTE The first digit of the ISO 4406 cleanliness code, in the case of automatic particle counting, is intentionally left blank throughout this document. The first digit describes the allowable number of particles of 6 μm and less which – according to today's knowledge – have no measurable influence on lifetime and reliability of the gearbox. Further, the measurement of these small particles in high viscous fluids is subject to large uncertainty.	

Measurements using automated particle counters can provide less than consistent results due to factors such as high viscosity of the oil, colour, air bubbles, water contamination and oil additives. Accuracy should be verified by manual count or the use of a dilution with chromatography grade solvent such as toluene. The lubricant supplier should be consulted for the proper method to be used for a specific oil. Caution should be exercised in interpreting the data. Sampling methods may influence the results, and the results of automatic particle counting methods may vary.

Additional information regarding lubricant cleanliness and filtration is provided in Annex E. The gearbox manufacturer should demonstrate by test results that the test oil used in production/acceptance testing of every gearbox meets the cleanliness shown in Table 11 or according to the cleanliness level applied as basis for the bearing calculation (see also 7.3.7.1). It should be demonstrated by test results that the oil in the gearbox at commissioning meets those cleanliness requirements. The turbine manufacturer shall ensure that, during the warranty period of the gearbox, the cleanliness of the oil is monitored and documented. The owner should continue the practice throughout the life of the gearbox.

7.6.10 Lubricant filter

Filtration devices shall remove ingested atmospheric and internally generated debris faster than they can accumulate within the gearbox. The filtration capabilities of the lubrication system shall maintain the bulk oil at the cleanliness levels outlined in 7.6.9. Responsibility for determining the appropriate components, dimensional sizes of the filter elements, the nominal sizes of filter media pores, and the efficiency ratings of the filter media should be shared by the wind turbine manufacturer, gearbox manufacturer, and filter manufacturer. The wind turbine manufacturer shall provide the gearbox manufacturer with environmental conditions expected at the turbine installation site (see clauses on operating environment). Refer to Annex E for further information on filtration.

The filter shall have a bypass function to permit adequate oil flow during cold start and when the filter media becomes contaminated. The status of the filter bypass valve should be monitored. The filter shall also have an electric clogging indicator providing a fail-safe signal when the filter element must be changed. Additionally, it should have an optical indication to allow assessment of the filter status by the service personnel.

7.6.11 Ports

7.6.11.1 General

Gearbox designers use a wide variety of ports for inspection, oil sampling, drain, fill, lubrication system, sensors, and the breather. Position, dimension and form of these ports shall be specified with respect to functionality of the gearbox, the functionality and interaction of the attached components, and serviceability. The following items are recommended depending on configuration.

7.6.11.2 Drain and fill plugs

Drain and fill plugs should be placed where lines are easily attached. The drain should be at the bottom of a sloped sump so that the oil can be completely drained. The drain opening should be an appropriate diameter to allow the oil to drain in a reasonable time. A valve shall be provided for draining the lubricant in the field. A plug should be provided to seal the drain valve.

7.6.11.3 Pressurized ports

Pressurised ports shall use parallel thread fittings with the seal achieved between two machined flat faces. A suitable sealing element must be included between the faces. Drawings and manufacturer's manuals shall clearly identify the connection points for the oil circulation and offline filter system.

7.6.11.4 Non-pressurized ports

Ports exposed to only fluid head pressure may be specified as tapered thread type (NPT).

7.6.11.5 Oil sampling ports

Gearboxes that use systems combining splash and pressure fed lubrication should be provided with ports for sampling for both sump and pressurized oil flow cleanliness. Ports for sump sampling should be located at the mid-height of the sump. The port should include a sampling valve and the inlet should extend into the sump where oil circulates freely. Avoid sampling stagnant areas. Pressure fed lubrication systems should incorporate two oil sampling ports. One should be between the pump and the filter, and the other directly after the filter in the flow direction. The sampling procedure should be specified as this can have a significant influence on consistent test results.

7.6.12 Oil level indicator

The gearbox shall be equipped with a device for field inspection of the oil level. The device shall be placed where it will give an accurate reading and not be damaged during routine maintenance. Metal dipsticks shall have a positive seal and shall be designed so that false readings due to oil being scraped off or picked up during removal and replacement is not possible. Quantity lines shall be permanently marked on the dipstick.

Sight gauges or glasses shall be made from appropriate materials that do not become clouded or opaque when in contact with the oil. The sight gauge and its connections shall be large enough to avoid sedimentation.

If an oil level sensor is installed, the sensor shall give a fail safe level indicator signal to the condition operation system.

7.6.13 Magnetic plugs

The use of magnetic plugs or dipsticks with magnets to detect wear metal particles is recommended. Magnetic plugs shall be placed in an area that is easily accessible in a circulating region of oil. The plugs or dipsticks should be regularly inspected for debris, and they should be cleaned or exchanged at each oil change. An oil filter housing equipped with a removable magnetic element in the inlet port is preferred for pressure fed systems. Magnetic plugs are not recommended for use in systems that rely on inline particle counters to monitor the health of the gearbox.

7.6.14 Breather

The requirements for a breather port with a filter element shall be specified. The breather should be of the disposable or serviceable spin-on type with a threaded inlet/mounting port, properly sized for unobstructed airflow during thermally and mechanically induced changes in the oil level within the gear housing. It should be designed and located to prevent discharge of oil to the atmosphere and to prevent entrance of environmental dust and water, brake dust, and other foreign material. The breather shall be located such that ingress of contaminants is not directed to gears and bearings. Furthermore, the location shall be chosen such that the breather is not contaminated by oil spray from gears and bearings. For maximum gear and bearing life, the breather should have a filtration rating equal to or less than 5 μm with low flow resistance. Depending on the environment and service conditions, a desiccant to minimize condensate formation in the gearbox may be required.

7.6.15 Flow sensor

The oil flow shall be monitored by a pressure sensor or oil flow indicator incorporated into the lubrication delivery system. Pressure sensors should be able to detect both excessively low and high pressures indicating inadequate or blocked flow. The sensor or indicator shall give a fail-safe signal to the control system.

7.6.16 Serviceability

When designing lubrication systems for wind turbines, serviceability and ease of maintenance should be considered. The following are examples that should be taken into consideration.

- Providing sample ports in the lubrication circuit both before and after the filter in easy to access locations.
- Ensuring that no loose items, such as washers, bolts, spacers or seals, are misplaced or lost during filter element servicing.
- Placement of the filter housing for easy access to remove and replace the filter element including a drip pan to capture any lubricant that may spill during an exchange.
- Incorporation of a bulk head mounted strainer of metallic construction with a bypass indicator.
- Ensuring that the lubrication system is self-priming in all modes of operation including initial fill and subsequent maintenance refilling.

8 Design verification

8.1 General

This clause defines minimum requirements for verification testing of new gearbox designs. Planning of the test shall, at the least, involve the turbine manufacturer, the gearbox manufacturer, the bearing manufacturer, the lubricant supplier and the designer of the lubrication system. The plans should also be reviewed with the certification body.

8.2 Test planning

8.2.1 Identifying test criteria

Based on the demands in the gearbox specification, test criteria for the verification of the gearbox design shall be established and agreed between the turbine manufacturer, gearbox manufacturer, bearing manufacturer and other relevant sub-suppliers. To develop these criteria a detailed evaluation of all elements in the design should be performed including (but not limited to) the assumptions made in:

- the design and rating factors for gears and gearbox elements;
- the selection, sizing and rating of the bearings;
- the details of the lubricant and lubrication system;
- structural analysis of deflections, mounting reactions and system dynamics.

Loads and load assumptions as well as other specifications delivered by the wind turbine manufacturer should be taken as a starting point for the evaluation. Special attention should be paid to assumptions and uncertainties related to the interfaces between the gearbox and the wind turbine such as torque arm supports, connections, couplings, and alignment.

8.2.2 New designs or substantive changes

Significant changes of gearbox and drivetrain design shall be evaluated for impacts on the system and may require repetition of relevant parts of the prototype test program. The following would require such evaluation:

- change in number of teeth on any gear/ pinion;
- change in rated speed of any stage;
- change in bearing type within the gearbox;
- change in bearing size within the gearbox;
- change to gearbox input or output nominal torque (see 6.3.3.3);
- change to gearbox mounting arrangement.

8.2.3 Overall test plan

A general test plan shall be formulated from the identified criteria. The test plan shall include prototype tests of the gearbox as a unit, some tests of the gearbox as an integrated part of the wind turbine and requirements for acceptance testing in serial production.

Testing should be performed as early as possible in the design process as described in 5.2 under repeatable and reproducible conditions. As many verifications as possible should be included in the gearbox testing in a test bench environment. It may also be beneficial to test single components or sub-systems of the gearbox externally. The content of gearbox verification testing in a wind turbine should be limited to turbine level requirements that cannot be accurately simulated on a test bench, such as noise emissions, transient effects and full system dynamics. When evaluating results from testing in a wind turbine, the stochastic nature of the environmental conditions in the field needs to be considered.

The general plan shall describe how and where (workshop, field test, etc.) the various objectives are met including what parameters to use to assess the verification criteria.

8.2.4 Specific test plans

For each separate test a specific test plan shall be written. The following items should be considered in these test plans:

- a description of the purpose and objectives including which parameters are being evaluated;
- description of acceptance/ rejection criteria to be used;
- methodology and procedures to be used for each test including required safety measures;
- the environmental conditions and system configurations to be tested including (as appropriate) operational settings for the control system;

- a list of the physical quantities to be measured;
- requirements for test instrumentation and data acquisition system including accuracy, calibration and traceability;
- special calibration procedures or requirements;
- repeatability and reproducibility of test results;
- the data analysis required to achieve the test goals;
- reporting and documentation requirements.

Where appropriate (e.g., field testing), IEC/TS 61400-13 can be used for guidance in planning, executing and documenting the test campaigns.

8.3 Workshop prototype testing

8.3.1 General

It is acceptable to prototype test single components, sub-assemblies, or entire gearboxes in a workshop or bench test to achieve the desired results.

8.3.2 Component testing

Some criteria can be most efficiently evaluated in sub-assembly testing (e.g., cooling capacity of the lubrication system). In this case objectives shall be described and a detailed test plan (see 8.2.3) shall be developed with clear definition of the component operating conditions and a rationale of how this is, and is not, similar to the actual wind turbine environment.

8.3.3 Workshop testing of a prototype gearbox

The gearbox shall be subjected to prototype tests at a suitable test bench. These tests serve to check the assumptions made in the design of the gearbox and obtain important parameters for the execution of acceptance tests during the production of wind turbine gearboxes.

At least one prototype test of a complete gearbox shall be performed. A test plan shall be developed before testing, including items as listed in 8.2.4 and definitions of all interface conditions. The test shall include the following, as a minimum.

- Low initial torque to be applied until the oil meets cleanliness limit stated in Table 11.
- Test torque application in a minimum of 4 steps up to nominal torque as defined in the gearbox specification. The torque steps shall be applied at the same rotational speed (preferably at nominal) or along the power-speed curve of the turbine.
- Continued testing at each load level until the oil meets the cleanliness limits stated in Table 11.
- Gear mesh contact patterns visually evaluated and documented after each of the torque steps. For inaccessible meshes other methods shall be applied to evaluate the related face load distribution factor, such as tooth root strain gauges.
- Measurement of actual gear face load distribution at each load step using tooth root strain gauges for the low-speed, high-torque parallel gear stage(s), typically the first two stages. The results shall be used to evaluate the design in relation to the resulting contact pattern and applied load factors (K-factors in 7.2.3) for the gear rating. Annex H provides literature references regarding methods for making such assessments.
- Measurement of actual load sharing for planetary and other split path gear meshes at each load step, for example using tooth root strain gauges. The results shall be used to evaluate the design in relation to the applied load factors (K-factors in 7.2.3) for the gear rating.
- Test duration at the nominal torque until bulk oil and bearing temperatures are stable ($\pm$ 2 K in 15 min) with normal cooling, or 6 h minimum.

- Record of temperatures, vibrations, noise and contact pattern at each torque step.
- Selection of bearing operating internal clearance shall be verified during prototype testing of the gearbox. This can for example be accomplished by temperature measurement or by analysis of the bearing running pattern.
- Temperature of bulk oil, oil inlet and outlet temperatures and temperatures of all stationary bearings (e.g. on the non-rotating ring) in the load path shall be measured and verified on the prototype gearbox.
- Appropriate instrumentation (such as displacement, strain or deflection measurements) to verify the structural analysis model of structural components such as gear housing or planet carriers and the torsional stiffness.
- Test gearbox shall be fully disassembled after the test. All gearbox elements shall be visually inspected, at least including all gears, bearings, bearing and gear journals, tooth couplings and splines, shafts and keys. The condition of these parts shall be documented in a detailed test report.

If the test results do not meet the agreed criteria then recalculations, redesign or both shall be performed.

8.3.4 Lubrication system testing

The gearbox shall be tested with the same oil type as used in the wind turbine and a lubricant delivery/cooling system of similar capacity to the planned/designed turbine system to verify that all the requirements for the cooling system are met, including bulk oil and bearing temperatures as described in 7.3.7.2, 7.6.6 and 7.6.7.

8.4 Field test

8.4.1 General

After the prototype workshop test (and resultant design review) a field test in a wind turbine shall take place. The test plan (see 8.2.3) and test specification for the field test shall include inputs from the gear and bearing manufacturers, but it is recommended that the field test only include items relevant for the gearbox as part of a wind turbine.

Gearbox specific measures such as bearing temperature measurements or load distribution patterns should not necessarily be a part of the field test. When possible these tests should be included in the workshop prototype test. Temperatures and mesh load factors can be part of the field test if there are uncertainties of the influence of the complete wind turbine assembly in the gearbox behaviour.

8.4.2 Validation of loads

8.4.2.1 Validation of gearbox design loads

Each wind turbine type will be subject to load measurements as part of the certification process according to IEC 61400-22:2010, Annex C. The purpose of load measurements for certification is to verify design calculations and to directly determine loads under specific conditions. The minimum extent of required load measurements is detailed in IEC 61400-22.

The loads validation from the type certification process shall be reviewed with the gearbox manufacturer and the bearing manufacturer. These will be used to verify that the as-designed unit is sufficiently conservative. It is not required to repeat this testing for new gearboxes unless there is a change in the drivetrain system that would substantively increase the wind turbine drivetrain design loads and response as noted in 8.2.2.

8.4.2.2 Validating wind turbine design models

Models used to simulate wind turbine response to prescribed design load cases (from IEC 61400-1 or IEC 61400-3) cannot normally be verified in all situations with field tests. This is generally true in extreme wind events and odd control responses. However, the uncertainty of the use of such models can be reduced with the following approach:

- in the wind turbine simulation codes, adjust turbine characteristics in order to reproduce as-measured response accurately using data from field tests (assure that atmospheric conditions used in simulations are also as measured in the field test);
- reproduce the simulations for design load cases not experienced in the field test (as performed in original load determinations);
- verify that loads used in the design are sufficiently conservative.

8.4.2.3 Gearbox specific field test requirements

Some design assumptions may have to be evaluated with specific testing. This could include torsional vibration, combined structural response and reaction at the gearbox supports.

These specific tests shall be performed for any new gearbox design (see substantive change criteria in 8.2.2) and the data be shared with the gearbox manufacturer with appropriate annotation describing events, configuration and conditions. The specific measurements shall be agreed upon between the gearbox manufacturer and the wind turbine manufacturer and shall as a minimum include:

- time series during selected events, such as
  - run-up through all operating speed ranges,
  - cut-in at transition winds and high winds,
  - shutdown at low and high winds,
  - brake application,
  - emergency stops at high and low winds,
  - idling and backwind idling,
  - electrical events, e.g. low voltage ride through;
- measured Campbell diagram (plot of system forcing and response frequencies) through the complete operating speed range to evaluate resonance risk.

The following signals should be measured with instrumentation that will observe all relevant frequencies and amplitudes of the mechanical vibrations:

- high speed shaft torque;
- low speed shaft torque, if applicable;
- shaft speed.

Sampling rate shall be selected in cooperation with the gearbox manufacturer. Typical sample rates will be in the range of 3 to 5 times of the relevant vibration frequency. Additional signals such as non-torque forces and moments may be required to evaluate gearbox interface loads and design assumptions.

For any time series provided, the conditions, configuration and control actions associated with the event should be described.

8.4.3 Type test of gearbox in wind turbine

The purpose of the type test is to demonstrate that the gearbox operates as anticipated when incorporated in the specific wind turbine. At least the following shall be demonstrated.

- Drivetrain resonances are avoided or minimized. Vibration levels at representative locations are at least corresponding to work shop test locations (see VDI 3834-1).

- Effectiveness of the lubricant delivery/ cooling system measured to verify that all the requirements for the cooling system are met, at least including oil inlet, oil outlet and oil sump temperatures.

8.5 Production testing

8.5.1 Acceptance testing

All gearboxes produced shall be subjected to a regimen of run-in and acceptance bench tests to assure proper performance of the gearbox. This test shall include, as a minimum, 3 load steps including 100 % nominal torque (see 6.3.3.3). All elements of the acceptance testing shall be specified in the acceptance test plan. This plan shall include all the required measurements, conditions and the acceptable performance criteria. The actual tests and the associated acceptance criteria shall be agreed upon between the wind turbine manufacturer and the gearbox manufacturer.

8.5.2 Sound emission testing

Acceptance tests for verification of the sound emissions from the gearbox should be performed on all units. ISO 8579-1 should be used in developing the acceptance requirements and methodology.

8.5.3 Vibration testing

Acceptance tests for verification of the gearbox vibrations shall be performed on all units. ISO 8579-2 is a possible method for measurement.

8.5.4 Lubrication system considerations

Cleanliness of the lubricant shall be proven to meet requirements in 7.6.9 (see Table 11) before loading the gearbox and at every load step. External filters are recommended to guarantee appropriate running-in conditions. The filter element shall be changed before shipping of the gearbox.

8.5.5 System temperatures

The acceptance test at the gearbox manufacturer shall at least include temperature measurements on those bearings and other measurement points on the gearbox which will be monitored in the wind turbine. Bearing temperature monitoring during the acceptance test may also be done as a means of quality control, for example for checking the clearance setting of paired tapered roller bearings.

The gearbox manufacturer, the wind turbine manufacturer and the bearing manufacturer shall agree on which bearings' temperatures shall be monitored during production acceptance test. They shall also agree on acceptance limits.

8.6 Robustness test

A test at elevated load levels above nominal torque shall be conducted to assess robustness and identify weak links. The specific test objectives, procedure and acceptance criteria shall be agreed between wind turbine manufacturer, gearbox manufacturer and bearing manufacturer. After such testing, the gearbox should be completely disassembled and all components inspected for wear or other distress.

8.7 Field lubricant temperature and cleanliness

Operating lubricant temperature and cleanliness are strong determinants of bearing rating life. It is therefore important to verify that the actual ranges of temperature and cleanliness of a representative number of turbines in the field meet the design assumptions. These results should be assessed with a band of uncertainty that is inherent in the use of ISO 4406 measurements on high viscosity fluids.

The gearbox manufacturer, the wind turbine manufacturer and the bearing manufacturer shall agree on which bearings will have temperature monitoring in the turbine. They shall also agree on shutdown limits, and the control system actions in case these limits are exceeded.

8.8 Bearing specific validation

8.8.1 Design reviews

The bearing selection shall be reviewed in a final design review between gearbox manufacturer, bearing manufacturer and wind turbine manufacturer. The design review should cover all bearing related requirements described in 7.3, including:

– bearing choice (bearing design);
– load assumptions;
– elasticity of mounting;
– review of assumptions and input data for bearing analysis;
– fulfilment of design lifetime requirements;
– contact stress;
– minimum load;
– operating temperature;
– oil flow;
– bearing clearance;
– bearing fitting practice including maintaining initial assembled clearance.

The result of the design review should be documented.

8.8.2 Prototype verification/validation

After the prototype test the bearings shall be visually inspected by the gearbox manufacturer, the wind turbine manufacturer, and the bearing manufacturer, and checked for the following:

– running patterns on raceway and ribs;
– condition of rolling elements;
– compliance of running patterns with the expected size and position of the load zone;
– evidence of edge load;
– evidence of excessive housing deflection;
– wear;
– size, count and location of significant particle indentations;
– polishing;
– micropitting on running surfaces;
– indications of scuffing or adhesive wear (e.g. slippage);
– discoloration or indication of heating;
– fretting corrosion;
– evidence of relative motion of rings (e.g. bearing ring positions marked prior to the prototype test);
– performance of coatings;
– condition of rolling element retainer (cage);
– condition of other bearing components (e.g., guide rings or seals).

These findings and the results of temperature measurements shall be documented and compared with the calculation results of the design process. The bearing manufacturer shall be informed in time about disassembly date and should be present when the prototype gearbox is dismantled.

8.9 Test documentation

All phases of the verification testing shall be comprehensively documented and evaluated by means of test reports, annotated measurement data files and photographs, oil analyses, and inspection or assembly reports. As an important part of the evaluation, an appropriate plan shall be defined for the test run of the production gearboxes. The documentation and evaluation shall be submitted together with the plan for the acceptance of production gearboxes to the certification agency for assessment.

Where appropriate – especially for the field tests – IEC/TS 61400-13 can be used for guidance for the reporting and documentation.

Uncertainty of measurements shall be considered. Annex B of IEC/TS 61400-13:2001 includes a methodology for assessing uncertainties in load related measurements.

9 Operation, service and maintenance requirements

9.1 Service and maintenance requirements

This clause defines the requirements for operation and maintenance (O&M), general inspection, and the service needed to obtain the expected life of the gearbox. Scheduled maintenance requirements should be agreed upon between wind turbine manufacturer and gearbox manufacturers. The wind turbine manufacturer shall provide documentation (manuals) for the installation, service and maintenance of the gearbox.

The manuals should include the inspections needed and service intervals required for oil filters, seals and other components. It should also define the bearing end play and service limits for bearing, shaft and gear elements.

In the design, if the decision is made to have components that need replacement during the design life, the turbine manufacturer shall be responsible for providing work instructions, and tooling to perform the replacement of the component. An agreement should be made to have bearings, gears and other consumable parts available during the time frame required for the change out of the components.

9.2 Inspection requirements

Documentation should describe methods and schedule for inspecting the gearbox and quantifying the inspection results, including templates for recording measurements. The objective is to provide information to the wind turbine owner to determine gearbox condition. These inspection documents should define the recommended repair or removal criteria including the following:

- method to inspect bearing end play with appropriate limits;
- methods to inspect the gearbox for debris and how to interpret results;
- methods to take lubricant samples, set limits for contamination, and what action is needed when limits are exceeded;
- condemning limits for lubricant viscosity and chemical parameters (moisture, additives content, etc.) with actions needed when limits are exceeded.

9.3 Commissioning and run-in

New gearboxes for replacement shall meet the criteria established in 8.5.1 for run-in and testing under load by the gearbox manufacturer. If the gearbox is not tested under load, instructions for the run-in of the gearbox shall be provided.

9.4 Transport, handling and storage

To prevent wear and damage to gearbox internal and external parts, instructions and specifications shall be provided for proper storage, transport, and handling during installation, turbine construction, and replacement. These instructions shall cover all possible ways of transport, for example by sea, rail, road or air. Protective means to prevent damage from environmental hazards such as ingress of moisture, debris or salt and loading from transportation (e.g. vibration, dynamic loading from rails, roads, etc.) shall be defined. Acceptance criteria for any relevant influence factors should be specified. The appropriate information for fitting of shipping fixtures and packing/unpacking instructions for replacement gearboxes shall be defined.

9.5 Repair

Considerations on design for repairability should be given. Different levels should be identified:

- how to change consumables, such as filters, sensors, etc., for scheduled or unscheduled maintenance;
- in-field accessible parts (scheduled or corrective actions) including HSS bearings, gear pumps and other activities that require skilled technicians;
- major repairs and rebuild activities including reversible parts, dismantling procedures, updated designs, etc.

9.6 Installation and exchange

Methods and tooling for the installation and exchange of the gearbox shall be defined. Special care shall be taken into account when exchanging a gearbox, so that the replacement is perfectly exchangeable with the original. At minimum, the replacing unit shall include any type of adapting part with the original interfaces.

Clear working, lifting and safety instructions shall be included in the manuals so that the change can be performed without risks and with minimum impact to availability (in this order).

9.7 Condition monitoring

If an agreement is made on a condition monitoring system with specified strategy and boundaries, the best options for monitoring locations and parameters should be identified during the design and verification process. Typical input parameters include:

- operating temperature of bearing(s);
- gear mesh frequencies;
- rolling element pass frequency;
- allowable oil pressure values.

This should drive the design of the condition monitoring system, so that gearbox response can provide information about its condition.

9.8 Lubrication

9.8.1 Oil type requirements

See 7.6 for lubrication system detailed requirements including selection and performance. The requirements to maintain this system shall be included in the manual.

9.8.2 Lubrication system

The wind turbine manufacturer shall include in the manuals:

- lubrication system servicing requirements and instructions;
- hydraulic system schematic diagram;
- expected cooling capacity of the system, and expected input and output oil temperatures.

9.8.3 Oil test and analysis

Limit values for viscosity, moisture and metal contents should be identified in the manuals as per 7.6.8. An oil analysis every six months is recommended.

The oil sampling procedure shall be specified. Guidance on developing such a procedure is found in E.4.3.

9.9 Operations and maintenance documentation

The O&M documentation shall include at a minimum an outline dimension drawing of the gearbox including the overall dimensions and the centre of gravity location, sufficient information to calculate gear mesh frequencies for condition monitoring, cross section drawing of the gearbox and a parts list for all components. The parts list shall include a description of the bearings and gears sufficient to order replacement parts including the bearing part numbers. Further, the list shall include components for the lubrication, cooling, and heating systems. This includes coolers, heaters, oil filter elements, filter housings, lubrication pumps, hoses, pipes, and all switches, thermometers or other monitoring devices.

Annex A
(informative)

Examples of drivetrain interfaces and loads specifications

A.1 General

This annex provides examples of specifying interfaces and loads for gearboxes in common wind turbine architectures. Further, it provides examples of how the interface definitions required in 6.1.2 can be detailed for these architectures.

This annex provides examples of how loads across these interfaces may be documented in accordance with the requirements in Clause 6.

As can be observed here, the specifications, loads and their description and documentation change with the drivetrain architecture.

A.2 Common wind turbine drivetrain architecture

A.2.1 Modular drivetrain with two main bearings

The modular configuration shown in Figure A.1 is a typical drivetrain consisting of two main bearings that support a separate main shaft that is normally not included in the gearbox assembly. The gearbox is mounted to the main shaft as either shaft or foot mounted. In this arrangement, the rotor bending moments are not transferred through the gearbox.

This configuration may cause a statically undetermined support of the drivetrain, such that reaction forces cannot be determined. Deflections and tolerances may cause additional reaction loads being transferred to the gearbox.

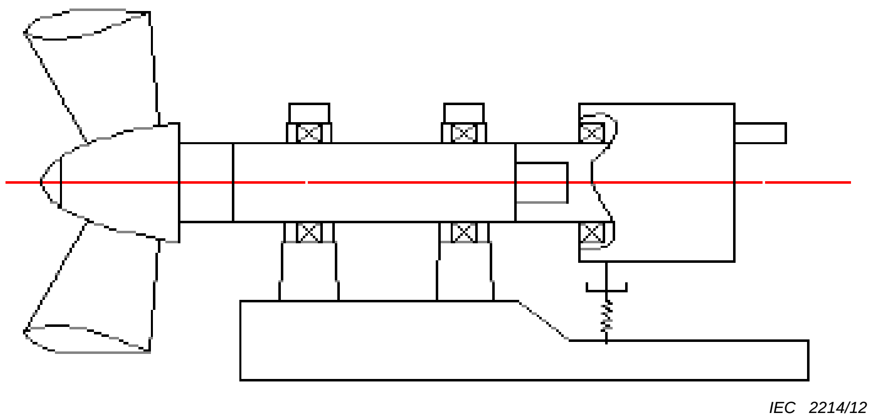


Figure A.1 – Modular drivetrain

A.2.2 Modular drivetrain with 3-point suspension

Another common wind turbine architecture is the 3-point suspension shown in Figure A.2 . In this configuration, a single main bearing is used separate from the gearbox, and this main bearing provides reactions to the rotor thrust and some part of the rotor bending moments. One or more bearings on the input shaft of the gearbox support the remaining part of the rotor reactions.

The rotor moments and forces that are reacted through the gearbox, as well as the torque, must be safely transmitted through the gearbox structure and mounting system.

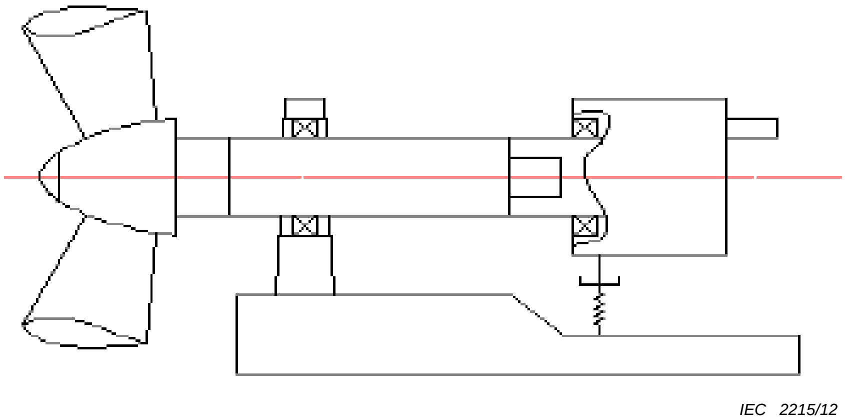


Figure A.2 – Modular drivetrain with 3-point suspension

A.2.3 Integrated drivetrain

Another layout opted for in wind turbine designs are integrated drivetrain solutions. In this arrangement the rotor bearings and much of the main shaft are integrated into the gearbox structure. However, since all rotor moments and forces are transmitted by the gearbox structure, the full panoply of rotor loads need to be provided at the gearbox LSS interface.

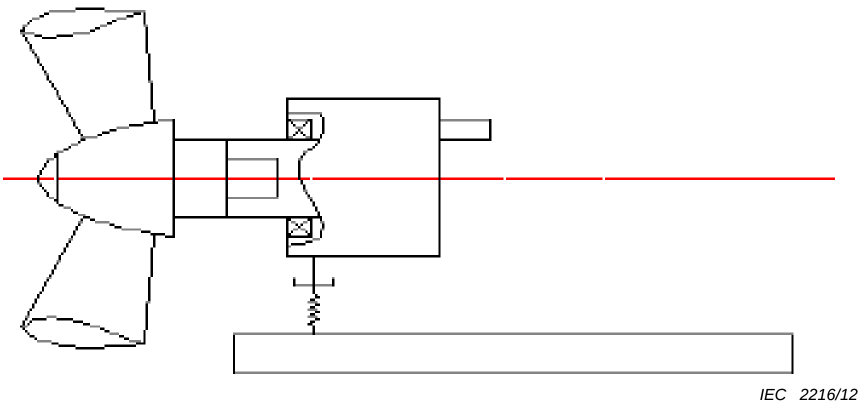


Figure A.3 – Integrated drivetrain

A.3 Interface definitions

A.3.1 Interfaces

In advance of providing the loads, the interfaces must be clearly defined for the specific drivetrain configuration. The information required for the relevant interfaces can then be further described specific to the interface.

A.3.2 Coordinate system

A specific referencing system for all loads should be described that includes the origin of all load references, the primary coordinates (x, y, z) and any small angle references such as main shaft tilt that may be needed to clarify orientation. This may be different for rotating and non-rotating coordinate systems.

Standards, authorities or the major wind turbine aero-elastic load simulation codes use different coordinate systems. The wind turbine manufacturers may have their own, company-specific definitions. These coordinate systems are not harmonized.

Therefore, the coordinate system shall be explicitly defined at each interface or globally with the use of figures, explanations and annotated as needed. This coordinate system shall be consistently referenced in all loads descriptions and presentations.

A.3.3 Interface descriptions

A.3.3.1 General

A thoroughly annotated drawing should be presented with the loads and specifications to inclusively describe the drivetrain interfaces and referenced elements or components. All reference points for applied loads and reactions should be detailed.

Figure A.4, Figure A.6 and Figure A.7 provide examples of such reference drawings for the drivetrain configurations typified in A.2, using the abbreviations and coordinate systems defined in Table A.1.

Table A.1 – Drivetrain elements and local coordinate systems

Drivetrain element	Index	Non-rotating coordinate system [a]	Rotating coordinate system [a]
Rotor	R		
Rotor hub	H	HCN	
(Main) rotor shaft	RS		
(1st) main bearing	MB	MBN	MBR
2nd main shaft bearing	SB	SBN	
Torsion support right	TSR	TSRN	
Torsion support left	TSL	TSLN	
Damper element	DE		
Brake disc	B	BCN	
Coupling	C	CCN	
Generator	G	GEN	
[a] Coordinate systems are located in the centre of the respective drivetrain element.			

In addition to the drawings specifying the interfaces, accurate physical dimensions between interfaces and pertinent structural elements should be specified in tabular forms for each interface, at least including the dimensions in Table A.2 and the respective tolerances.

Table A.2 – Drivetrain element interface dimensions

Dimension	Unit	Distance
l_1	mm	Rotor centre to centre of first main bearing
l_2	mm	Centre to centre of two main bearings
l_3	mm	Main bearing centre to torsion support centreline
l_4	mm	Main bearing centre to brake disc
l_5	mm	Main bearing centre to coupling
l_6	mm	Main bearing centre to generator
l_7	mm	Centerline distance between right and left torsion supports
l_8	mm	Straightline distance between brake disc centre and main shaft axis
l_9	mm	Height distance from main shaft axis to torsion support axis
a	deg	Angle of l_7 to horizontal

A.3.3.2 Modular drivetrain with two main bearings

A modular drivetrain configuration with two separate main bearings is shown in Figure A.4 and Figure A.5 with the proposed interfaces, dimensions, and referenced elements.

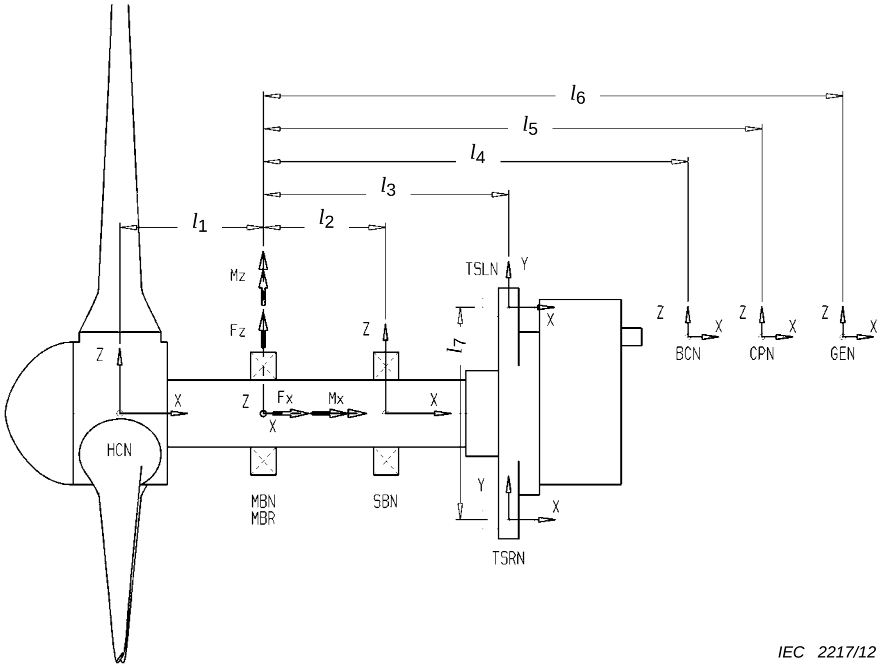


Figure A.4 – Reference system for modular drivetrain

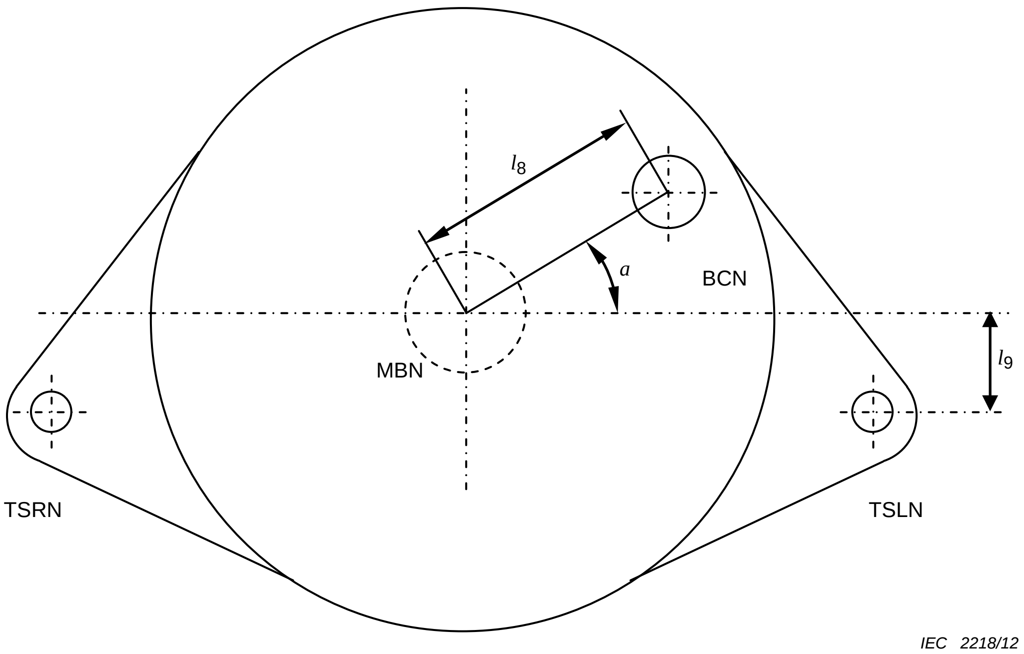


Figure A.5 – Rear view of drivetrain

For analysis of such a drivetrain, information about the mass, stiffness, and damping for relevant drivetrain elements may be required. An example of such an interface specification is given in Table A.3.

Table A.3 – Interface requirements for modular drivetrain

	Interfacing drivetrain components					
Drivetrain element	**R**	**RS**	**TSR/TSL**	**B**	**C**	**G**
Coordinate system	**HCN**	**MBN**	**TSRN/ TSLN**	**BCN**	**CCN**	**GEN**
Mass, m	x	x		x	x	x
Inertia, I	x	x		x	x	x
Stiffness, c_x			x			x
Stiffness, c_y			x			x
Stiffness, c_z			x			x
Damping, d			x		x	
Coupling stiffness					x	

A.3.3.3 Modular drivetrain with 3-point suspension

A modular drivetrain configuration with one separate main bearing is shown in Figure A.6 with the proposed interfaces, dimensions, and referenced elements. Table A.4 provides an example of an interface specification including information that may be required for analysis of such a drivetrain.

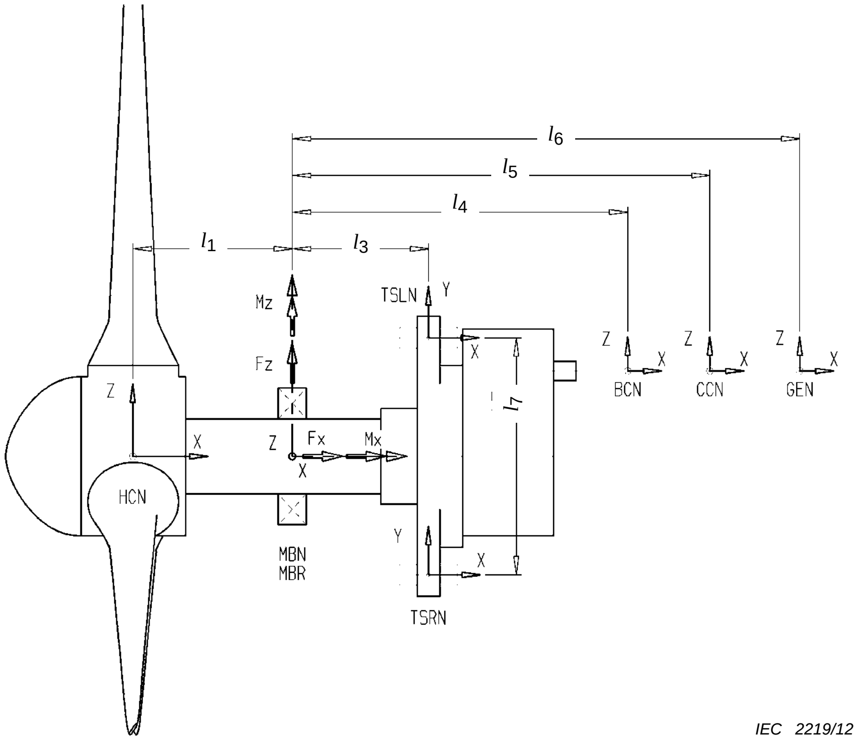


Figure A.6 – Reference system for modular drivetrain with 3-point suspension

Table A.4 – Interface requirements for modular drivetrain with 3-point suspension

	Interfacing drivetrain components					
Drivetrain element	**R**	**RS**	**TSR/TSL**	**B**	**C**	**G**
Coordinate system	**HCN**	**MBN**	**TSRN/ TSLN**	**BCN**	**CCN**	**GEN**
Mass, m	x	x		x	x	x
Inertia, I	x	x		x	x	x
Stiffness, c_x			x			x
Stiffness, c_y			x			x
Stiffness, c_z			x			x
Damping, d			x		x	
Coupling stiffness					x	

A.3.3.4 Integrated drivetrain

Figure A.7 shows an integrated drivetrain configuration including one momentum-type main bearing with the proposed interfaces, dimensions, and referenced elements.

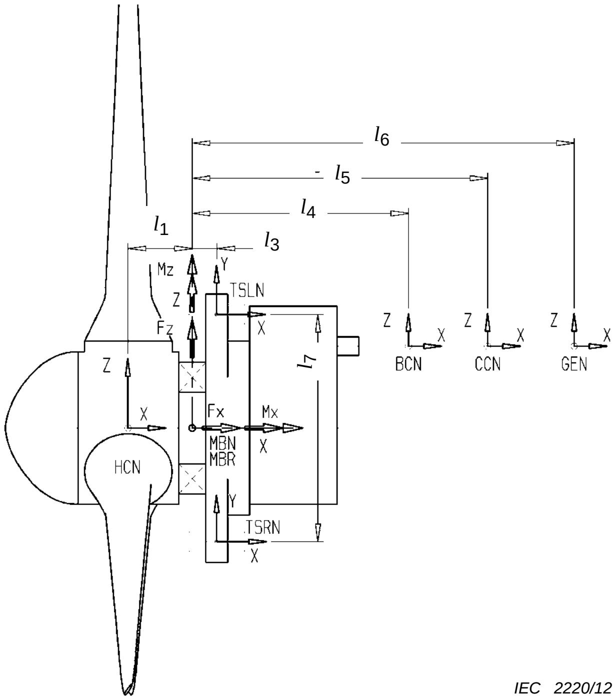


Figure A.7 – Reference system for integrated drivetrain

Table A.5 summarizes the information about mass, stiffness, and damping for relevant drivetrain elements that may be required for analysis of such integrated drivetrains. The table should be supplemented with a figure locating all referenced drivetrain elements. The panoply of loads and the complete gamut of stiffness information should be made available.

Table A.5 – Interface requirements for integrated drivetrain

	Interfacing drivetrain components					
Drivetrain element	**R**	**RS**	**TSR/TSL**	**B**	**C**	**G**
Coordinate system	**HCN**	**MBN**	**TSRN/ TSLN**	**BCN**	**CCN**	**GEN**
Mass, m	x			x	x	x
Inertia, I	x			x	x	x
Stiffness, c_x						x
Stiffness, c_y						x
Stiffness, c_z						x
Damping, d					x	
Coupling stiffness						

A.4 Required engineering data at the interface

A.4.1 Engineering data

The engineering data specified and referenced at each interface will take several forms depending on the situation and configuration. Commonly specified information is listed in Table A.6. Some of this information is required for dynamic analysis and may be exchanged only if the vendor is doing this analysis.

Table A.6 – Engineering data and required design load descriptions

load information	Unit	Engineering data	RFC	LDD	EXT [a]	Time series
Moment, [b] M_{xyz}	Nm	X	X		X	X
Torque, [b] T_R	Nm		X	X	X	X
Force, F_{xyz}	N	X	X		X	X
Deflection, δ_{xyz}	mm		X			X
Rotation, α_{xyz}	deg		X			X
Rotor speed, n	min^{-1}		X			X
Moment of inertia, I	$kg{\cdot}m^2$	X				
Torsional stiffness	Nm/rad	X				
Axial stiffness, c	N/mm	X				
Damping, d	% critical	X				
Mass, m	kg	X				

[a] Extreme load matrices, see A.4.5.

[b] Index of rotor torque depends on simulation code used.

Dynamic stiffness and damping properties of complex structures and systems is frequency dependent. Data such as stiffness and inertias in the interface definition (whether structural or control-related) should be associated with a frequency range for which it is valid.

A.4.2 Required wind turbine load descriptions

Table A.6 summarizes wind turbine load information that may be required for designing the gearbox. The interface point depends on the configuration as described previously. In case of difference between loads in the rotating and the non-rotating coordinate system both shall be provided. Deflections and rotations shall be provided in the non-rotating system.

A table should be provided that uniquely indexes (including data file names, etc.) statistical summaries, load matrixes and time series that meet the interface requirements specified in Table A.6. This would provide an index to the pertinent data files provided to all drive train sub-suppliers.

A.4.3 Rainflow matrices

One common presentation of forces and moments for fatigue analysis of structures are rainflow count matrices as shown in Table A.7. A rainflow matrix shows mean loads L_i and range loads (R_j, peak to peak differential loads in a given cycle) and the amount of cycles. The rainflow counting method is used to derive these loads and cycles data from a load time series (see e.g. Downing and Socie (1982) or Matsuishi and Endo (1968) or ASTM E1049).

Table A.7 – Rainflow matrix example

	Load range			
Mean load	$\mathbf{R_1}$	$\mathbf{R_2}$	$\mathbf{R_{...}}$	$\mathbf{R_j}$
L_1	n_{11}	n_{12}		n_{1j}
L_2	n_{21}	n_{22}		n_{2j}
$L_{...}$				
L_i	n_{i1}	n_{i2}		n_{ij}

NOTE The rainflow count matrix is a helpful means for calculating the fatigue strength of structural components of the gearbox such as torque arm or planet carrier, because the mean value of each range of cycles has a strong influence due to the steep S/N-curve. As a result, it can be used to get an accurate calculation of the fatigue damage during the component lifetime.

This is not the case for gear life calculation as per ISO 6336-6 (calculation of service life under variable load). In that case the data needed does not include the mean value of the stress because the proposed SN curves are the minimum possible for every material so that for any mean stress value the result of the calculation is conservative.

The rainflow count matrix shown in Table A.7 can be transformed into a vector containing the sum of the cycles obtained for every range. Figure A.8 shows an example of a rainflow count without mean values represented in a plot that also displays the contribution of the different DLC to the total load spectrum.

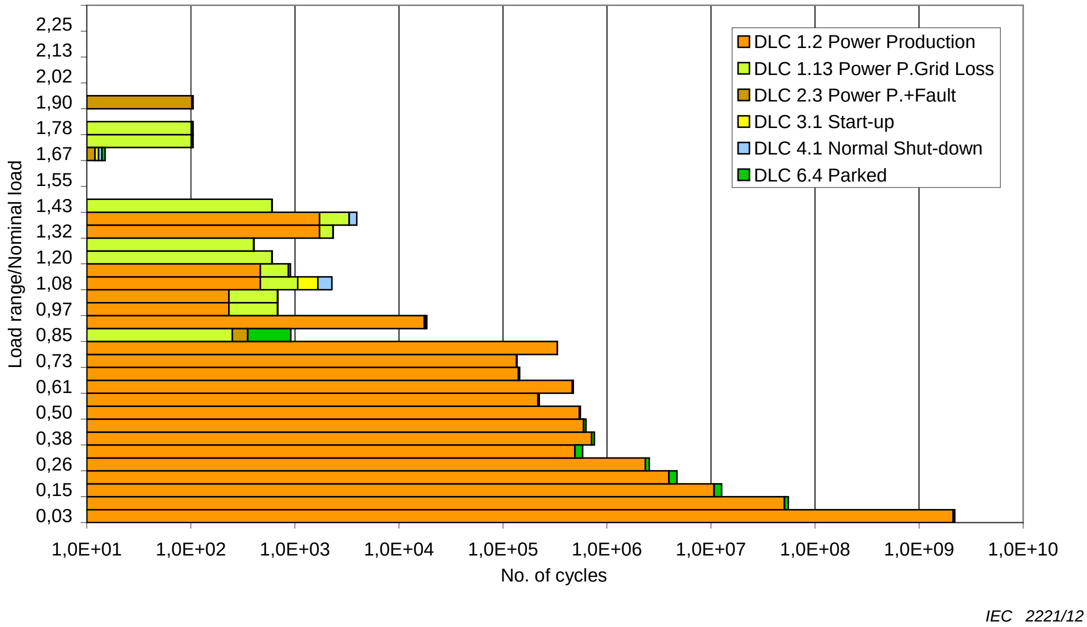


Figure A.8 – Example of rainflow counting per DLC

A.4.4 Load duration distribution

Due to the nature of the gear and bearing systems the load does not only depend on the magnitude of the driven load but also on the speed at which every geared shaft and bearing is turning. For a bearing the equivalent load is determined by the number of cycles at a particular load range. The load duration, or revolution distribution, is a convenient way to describe load data for the design of gearboxes.

A load duration matrix is shown in Table A.8. The load described shall be clearly defined according to the coordinate system and the accepted referencing methodology.

The load duration matrix shown in Table A.8 can be transformed into a vector Σn_j containing the accumulated time or sum of the cycles at each load level L_i. Figure A.9 shows an example of a load revolution distribution specified in cycles-at-level.

The load described shall be clearly defined according to the coordinate system and the accepted referencing methodology.

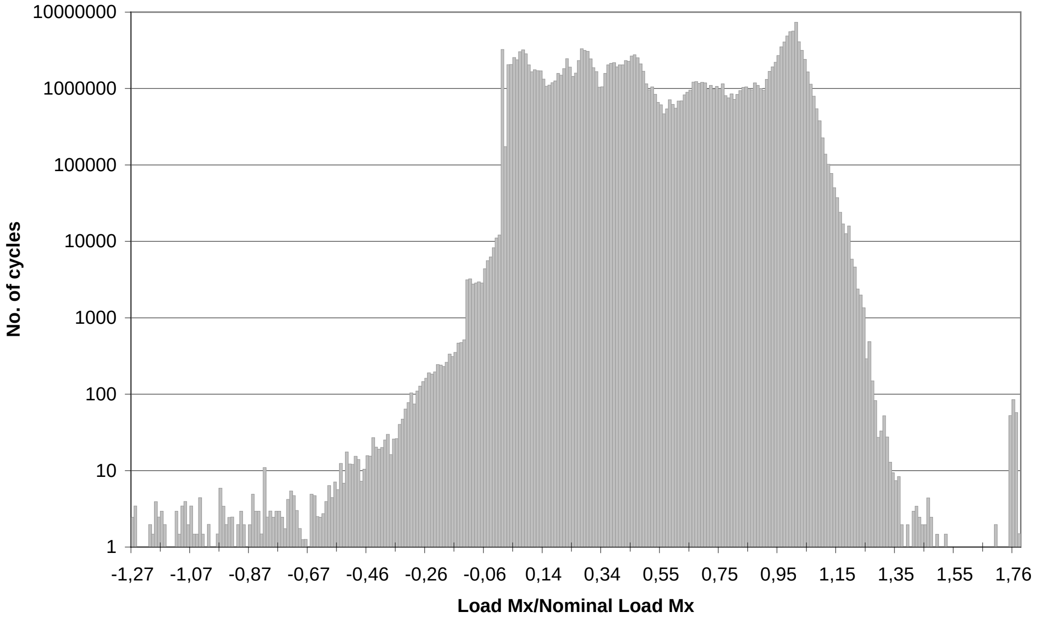


Figure A.9 – Example of load revolution distribution (LRD)

Table A.8 – Example of load duration distribution (LDD)

Torque ratio (to nominal)	**Time at load** (hours)
1,67	0,04
1,59	0,03
1,51	0,02
1,44	0,69
1,36	3,16
1,29	26,60
1,21	316,00
1,14	2 730,00
1,06	9 480,00
0,98	12 000,00
0,91	9 300,00
0,83	8 400,00
0,76	7 490,00
0,68	6 540,00
0,61	6 420,00
0,53	8 090,00
0,45	7 730,00
0,38	10 200,00
0,30	13 100,00
0,23	14 500,00
0,15	15 600,00
0,08	13 700,00
0,00	15 100,00
- 0,08	8,07
- 0,15	3,84
- 0,23	5,22
- 0,30	1,37
- 0,38	0,29
- 0,45	0,44
- 0,53	0,89
- 0,61	0,67

A.4.5 Extreme load descriptions

Extreme loads shall be presented in matrices that reveal the magnitude of one load component, and the simultaneous level and implied phase of the other load components at that same time t when that extreme load occurs. An example with the 10 worst load cases for one load component (M_x in this example) is given in Table A.9.

Table A.9 – Extreme load matrix example

Rank	$M_{x,max}$	M_y	M_z	F_x	F_y	F_z	n_R	DLC	t	γ_f
	kNm	kNm	kNm	kN	kN	kN	min^{-1}	-	s	-
1										
2										
…										
4										

As the other load components reach their respective maximum at a different time, similar tables should be provided for each load component.

Annotated time series may be supplied as appropriate to further describe the extreme load events, and the implied phase relations between the load components.

Annex B
(informative)

Gearbox design and manufacturing considerations

B.1 General

This annex covers some important, but non-normative, aspects of gear design and manufacturing.

B.2 Gearbox design

B.2.1 Gear arrangements

Requirements for the arrangement of the gears inside the housing may be specified.

B.2.2 Lifting holes

All large gears should have some means for lifting, such as holes or threaded holes designed to accept shackles or eye bolts for lifting, when there is sufficient material between the root diameter and the bore of the gear.

B.2.3 Aspect ratio

The aspect ratio, defined as the ratio of the common face width b divided by the reference diameter of the pinion d_1, is an indicator of how sensitive a gear set is to misalignment. To achieve good load distribution on spur and single helical gears, the aspect ratio should be less than 1,25. For double helical gears the aspect ratio should be less than 2,0.

B.2.4 Profile shift

Profile shift is used to:

- prevent undercut;
- balance specific sliding;
- balance flash temperature;
- balance bending fatigue life;
- avoid narrow top lands.

The profile shift should be large enough to avoid undercut and small enough to avoid narrow top lands. For wind turbine gears, which are speed increasers, it is usually best to design the profile shift for balanced specific sliding.

B.2.5 Planet gear rim thickness

Planet gear rim thickness should equal at least 3 modules. Rim strength, tooth strength, and risk of movement of the bearing outer ring should be considered.

B.2.6 Gear tooth surface roughness

Gear tooth surface roughness is one of the most important factors influencing the risk of micropitting. Table B.1 provides recommended maximum values for the average as-manufactured surface roughness R_a that – by experience – have proven to reduce the risk of micropitting.

Table B.1 – Recommended gear tooth surface roughness

Gear	R_a µm
High speed pinion and gear	≤ 0,7
Intermediate pinion and gear	≤ 0,7
Low speed pinion and gear	≤ 0,6
Low speed sun and planet	≤ 0,5

B.3 Cleanliness in gearbox assembly

B.3.1 Cleanliness

Gearbox assembly facilities have to be clean in order to minimize initial damage to bearing and gear surfaces during first operation that can occur due to particulate contaminants that accumulate within the gearbox throughout the assembly process.

The following recommendations apply to facilities where the gearbox is assembled, and to storage and handling of the following:

- housing;
- gears and pinions;
- bearings and seals;
- shafts;
- lubrication system and heating/cooling system components such as pipes, tubes, hoses, fittings, heat exchangers or other fluid conditioning parts, valves, spray jets, manifolds, and bearing oil trays.

All methods and procedures should result in minimizing potential sources of solid contaminants.

B.3.2 Dedicated cleaning area

All components should be cleaned prior to assembly in an area separate from the assembly area. Bearings should be taken directly from manufacturer's packing.

B.3.3 Storage and handling

Components should be delivered to the assembly area in a clean condition. This includes the thorough flushing of internal surfaces and passages of all lubrication system components with clean filtered oil.

At a minimum, those components listed in the scope should be cleaned, packaged, wrapped, and stored such that dust, dirt and particulates are not present on or in these components. In the case of the lubrication system the single parts such as pipes; tubes, or hoses should be covered or plugged.

B.3.4 Dedicated assembly area

The gearbox assembly area(s) should be installed in a manner that contamination from other production areas is prevented. These are production areas for single parts and cleaning areas.

Examples of potential measures to prevent contaminations are:

- avoid distribution of contamination;
- regular cleaning of floors, cranes, handling systems and other tools;
- suitable surface finish of facility areas.

B.3.5 Covering work in-process

It should be assured that contamination is avoided during significant stops in the assembly process of the gearbox. This should be prevented by wrapping or covering the gearbox and/or gearbox components.

B.3.6 Verification of cleanliness

To assure cleanliness in the assembly area, a specified process should be in place. The gearbox manufacturer's quality plan should fully describe the cleanliness monitoring, inspection and action plan.

Annex C
(informative)

Bearing design considerations

C.1 Preliminary bearing selection

Basic rating life L_{h10} in accordance with ISO 281 or the bearing manufacturers' catalogue may be used for a preliminary selection of bearings in the design process of the gearbox. This standardized calculation method for dynamically loaded rolling bearings is based on equivalent load (P), speed (n), bearing dynamic load rating (C) and simplified load distribution assumptions. Table C.1 lists recommended minimum values for the basic rating life, L_{h10}, for this preliminary bearing selection process. In accordance with ISO 281, L_{h10} is calculated from Equation (C.1).

$$L_{h10} = \frac{10^6}{60 \cdot n} \cdot \left(\frac{C}{P}\right)^p \tag{C.1}$$

Miner's rule should be used to combine loads and speeds given in the load spectrum supplied by the wind turbine manufacturer. The life exponent, p, should be 3,0 for ball bearings and 10/3 for roller bearings.

Table C.1 lists recommended minimum values for the basic rating life for preliminary bearing selection. Note that the values in this table are valid for a design life of 20 years, and they should be adjusted for designs with different design lifetime.

Table C.1 – Guide values for basic rating life L_{h10} for preliminary bearing selection

Bearing position	Speed range $n \times D_{pw}$	Recommended basic rating life L_{h10} in hours
High speed shaft	150 000 to 430 000	30 000
High speed intermediate shaft	25 000 to 220 000	40 000
Low speed intermediate shaft	10 000 to 60 000	80 000
Intermediate sun shaft	10 000 to 60 000	80 000
Intermediate planet	20 000 to 150 000	80 000
Low-speed planet	10 000 to 60 000	100 000
Low speed shaft	5 000 to 15 000	100 000

NOTE 1 These guide values have been derived from experience with contemporary gearbox designs where the speed index $n \times D_{pw}$ falls within the specified ranges.

NOTE 2 The guide values apply for bearings manufactured from contemporary, commonly used, high quality hardened bearing steel, in accordance with good manufacturing practice and basically of conventional design as regards the shape of rolling contact surfaces.

NOTE 3 Values in this table are valid for a design lifetime of 20 years.

NOTE 4 Usually, there is no equivalent load available for the input shaft.

C.2 Method for load bin reduction

C.2.1 Purpose

This clause proposes different methods for reducing the number of bins in a given load spectrum. The need for reducing the number of bins, and the method applied, should be agreed upon by the bearing manufacturer, the gearbox manufacturer and the wind turbine manufacturer. The reduction of bins should be supported with an uncertainty analysis, to ensure that the method assesses bearing life in a conservative manner.

The methods presented here are only applicable for bearings that are predominantly loaded by forces resulting from rotor torque. This is the case with most gearbox bearings, but the notable exception are wind turbine designs where the input shaft carries other moments and forces from the main shaft and rotor. Further, the methods presented here neglect other effects (such as vibration or component weight) for simplification.

C.2.2 Lumping neighbouring load bins

The simplest way of reducing the number of bins is to lump a number of neighbouring bins as illustrated in Figure C.1. If this method is applied, the load for the reduced bin should be the maximum load of the original load bins. It is not recommended to use less than 20 load bins when applying this method. The size of the load steps which define a bin need not be held constant. The number of bins that are lumped together does not need to be constant either.

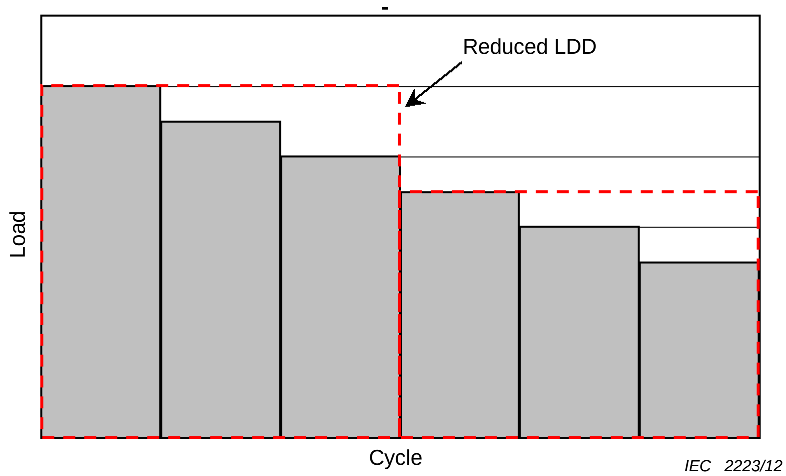


Figure C.1 – Load bin reduction by lumping neighbouring load bins

The number of cycles n_j and the load level P_j of the j^{th} bin of the reduced load spectrum are calculated by Equations (C.2) and (C.3):

$$n_j = \sum_{i=n}^{m} n_i \tag{C.2}$$

$$P_j = \max_{i=n}^{m} (P_i) \tag{C.3}$$

where

$i = n..m$ are the bins of the original spectrum combined into the j^{th} bin of the reduced load spectrum;

P_j is the load level of the j^{th} bin of the reduced load spectrum;

P_i is the load level of i^{th} bin of the original load spectrum;

n_j is the number of cycles in the j^{th} bin of the reduced load spectrum;

n_i is the number of cycles in the i^{th} bin of the original load spectrum.

C.2.3 Weighted load averaging

This method for reducing the number of bins uses Miner's rule to determine a weighted average load for each bin of the reduced spectrum. The same life exponent is used as applied in the calculation of the reference rating life L_{nmr}, according to ISO/TS 16281.

The number of bins of the reduced spectrum should not be less than 20. The number of bins that are combined does not need to be constant.

The number of cycles n_j is determined according to Equation (C.2), and the load level P_j of the j^{th} bin of the reduced load spectrum is calculated by the following equation:

$$P_j = \left(\frac{\sum_{i=n}^{m} \left(P_i^{\,p} \times n_i \right)}{n_j} \right)^{\frac{1}{p}} \tag{C.4}$$

where

$i = n..m$ are the bins of the original spectrum combined into the j^{th} bin of the reduced load spectrum;

P_j is the load level of the j^{th} bin of the reduced load spectrum;

P_i is the load level of i^{th} bin of the original load spectrum;

n_j is the number of cycles in the j^{th} bin of the reduced load spectrum;

n_i is the number of cycles in the i^{th} bin of the original load spectrum;

p is the life exponent used in the bearing life calculation.

C.2.4 Weighted life averaging

The methods presented in the previous two clauses assume a fixed exponential relation between the load level and the life consumption of a particular bin. However, the modified reference rating life L_{nmr} per ISO/TS 16281 is a function of the local pressure in combination with contamination- and lubrication parameters e_C and κ. This means that bins with higher loads consume proportionally more life than the methods suggest. This may result in an overestimation of the rating life compared to a detailed bin-by-bin analysis.

The weighted life averaging method for spectrum reduction takes this non-linearity into account by applying a fine resolution to the reduced spectrum at loads where most of the life is consumed, and a coarse resolution at low loads where only small part of the life is consumed. The life consumption of a particular bin of a load spectrum is expressed as

$$CLI_i = \left(\frac{P_i^{\,p} \times n_i}{\sum_{i=1}^{m} \left(P_i^{\,p} \times n_i \right)} \right) \times 100 \tag{C.5}$$

where

CLI_i is the consumed life index for the i^{th} bin of a load spectrum, in %;

P_i is the load level of i^{th} bin of a load spectrum;

m is the number of bins in a load spectrum.

The method described below is based on a number of assumptions that need to be verified in each particular application:

- operation around nominal load consumes most of the bearing life;
- the time share of and the consumed life at negative load levels is small;
- the original load spectrum does not contain extreme loads;
- the original load spectrum contains corresponding speed information for each load bin, such that the influence of e_C and κ can be considered.

When properly applied in the following steps, this method should achieve a high accuracy (compared to using the original load spectrum bin-by-bin) already at approximately m = 10 bins. This is desirable when calculation time is significant, for example when a sophisticated model for considering flexible bearing environment is used.

– The first bin j = 1 of the reduced load spectrum cumulates the cycles of all bins with reverse loads from the original load spectrum. The load level of this bin should be set to the maximum reverse load. Bin j = 1 should be maintained in the reduced spectrum, even in case the time share or consumed life index of that bin is almost zero.

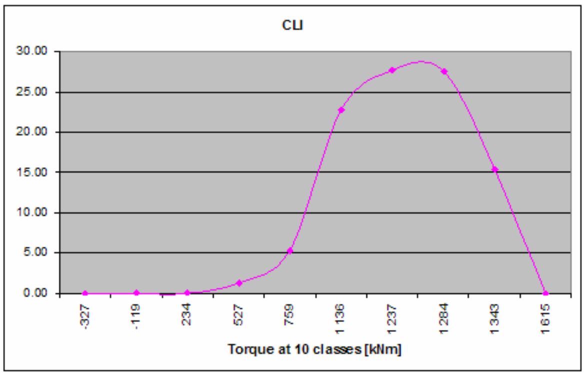
– The last bin j = m of the reduced spectrum includes the maximum operating torque, i.e. the highest bin of the original LDD or LRD. The load level of this bin should be set to the maximum operating torque. Bin j = m should be maintained in the reduced spectrum, even in case the time share or consumed life index of that bin is almost zero.

– The consumed life index CLI is calculated for each bin of the original load spectrum. Then bins are cumulated such that the CLI for each bin of the reduced load spectrum is lower than 25 %. If any individual bin of the original load spectrum already exceeds this limit, this bin should be directly transferred into the reduced spectrum. In this case, a finer resolution of the original load spectrum should be considered.

– One bin of the reduced spectrum should include the nominal mechanical load. The consumed life index CLI should reach its maximum for this particular bin.

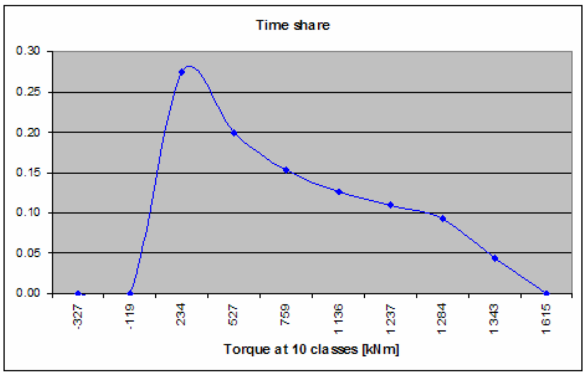
– Bins of higher and lower loads are combined such that the consumed life index decreases steadily to both sides as shown in Figure C.2.

– The time share should steadily decrease from zero load to the maximum operating load, as shown in Figure C.3. The two criteria regarding consumed life index CLI above should have higher importance than the distribution of time shares.



IEC 2224/12

Figure C.2 – Consumed life index (CLI)



IEC 2225/12

Figure C.3 – Time share distribution

The number of cycles n_j is determined according to Equation (C.2), and the load level P_j of the j^{th} bin of the reduced load spectrum is calculated according to the following equation (variables defined above):

$$P_j = \left(\frac{\sum_{i=n}^{m} \left(P_i^{\,p} \times n_i \right)}{n_j} \right)^{\frac{1}{p}} \qquad (C.6)$$

The equivalent speed $n_{eq,j}$ for the j^{th} load bin should be the average speed of each original load bin weighted with the respective time share of that bin in the original LDD or LRD.

If the estimated consumed life index CLI deviates significantly from the life consumptions calculated in the final life calculation per ISO/TS 16281, the load spectrum reduction should be revised until a better convergence is achieved.

C.3 Simplified bearing contact stress calculation

C.3.1 Purpose

This annex provides a simplified method to calculate the maximum contact pressure in spherical, cylindrical and tapered radial roller bearings (SRB, CRB, TRB) operating without preload. For toroidal bearings (TORB), the methodology for SRB should be used. This method is not intended to replace more advanced bearing life calculations, but rather to create transparency for the advanced methods that may be proprietary to the bearing manufacturer. Precise methods for calculation of contact stress are described by Reusner (1977), Hartnett (1980) or DeMul et al. (1986).

C.3.2 Influence factors

The method determines three factors describing the load distribution in the bearing: k, K_m and K_{lc}. It is assumed that, due to convex-convex contact, the highest contact pressure occurs at roller contact with the inner ring. This method is valid only for inner ring to roller contact and bearings with steel rollers and rings.

The method provides a first approximation for the key factors and resulting contact pressure. In practice, additional influence factors include micro-geometry of the bearing, support system stiffness, and bearing type. This method is limited to applications where roller and raceway modifications are adequate to prevent high edge contact stresses.

C.3.3 Procedure

C.3.3.1 Equivalent bearing load

Based on the applied radial load F_r and the applied axial load F_a an equivalent load P_0 is determined. Static factors X_0 and Y_0 are used, since they describe a relation between the applied load and maximum stress, without adding dynamic factors:

$$\begin{aligned} &P_0 = X_0 \cdot F_r + Y_0 \cdot F_a \\ &P_0 = F_r \quad \textit{if} \quad X_0 \cdot F_r + Y_0 \cdot F_a \le F_r \end{aligned} \qquad (C.7)$$

where

F_r is the maximum radial load, in N;

F_a is the maximum axial load, in N;

X_0 is the static radial load factor according to Table C.2;

Y_0 is the static axial load factor according to Table C.2.

Table C.2 – Static load factors for radial bearings

Type of radial bearing	Single row bearing		Double row bearing	
	X_0	Y_0	X_0	Y_0
Spherical roller bearing	0,5	$0,22 \cdot \cot \alpha_0$	1,0	$0,44 \cdot \cot \alpha_0$
Cylindrical roller bearing	1,0	0	1,0	0
Tapered roller bearing	0,5	$0,22 \cdot \cot \alpha_0$	1,0	$0,44 \cdot \cot \alpha_0$

C.3.3.2 Maximum rolling element load

The combined external force P_0 is distributed over a number of rolling elements creating a statically indeterminate support system. Based on methods by Brandlein et al. (1999), the load distribution may be determined from a mutual displacement of the rings and calculating the associated deformation of the rolling elements. The single roller maximum load for a clearance-free bearing, Q, becomes hence:

$$Q = \frac{P_0}{Z \cdot \cos \alpha_0} \cdot \mathrm{k} \qquad \text{(C.8)}$$

where:

k is the load sharing factor for the maximum loaded roller;

Z is the number of rolling elements in a bearing row;

α_0 bearing nominal contact angle.

This yields the ratio:

$$\mathrm{k} = \frac{Q \cdot Z \cdot \cos \alpha_0}{F_r} \qquad \text{(C.9)}$$

and k = 4,06 at zero internal clearance and a 180° load zone. Figure C.4 shows k for various levels of the assembled internal radial clearance, G_r (in mm). k varies by the ratio:

$$\frac{F_r}{C_{\delta L} \cdot \left(\dfrac{G_r}{2}\right)^{1,08} \cdot Z} \qquad \text{(C.10)}$$

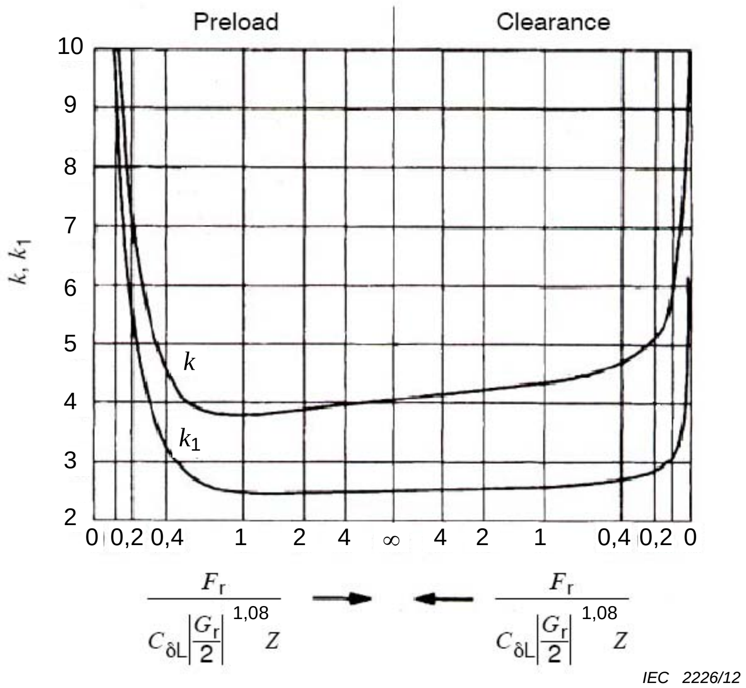


Figure C.4 – Effects of clearance and preload on pressure distribution in radial roller bearings (from Brandlein et al, 1999)

The elastic constant $C_{\delta L}$ is calculated as

$$C_{\delta L} = 26\,200 \cdot \left(L_{we} \right)^{0,92} \tag{C.11}$$

where

L_{we} is the effective roller length, in mm.

The variable G_r must be greater than zero for preventing a numerical instability. A practical lower boundary is G_r = 0,0005.

Equation (C.12) provides a curve fit for k for radial bearings operating with clearance:

$$k = 4,05 + 0,3209 \cdot \left[\frac{F_r}{C_{\delta L} \cdot \left(\frac{G_r}{2} \right)^{1,08} \cdot Z} \right]^{-0,7911} \tag{C.12}$$

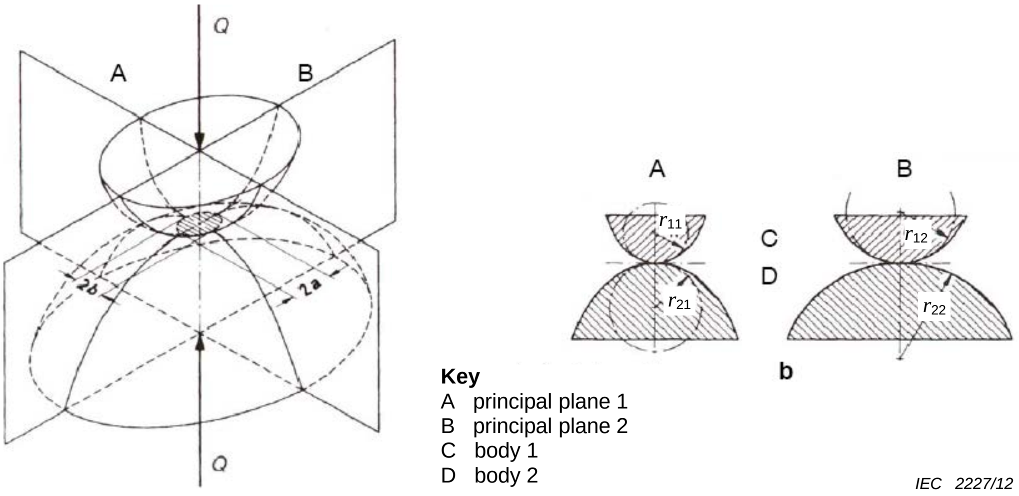
Figure C.4 is based on radial bearings with contact angle α_0 equal to zero. Introducing α_0 into the equation and replacing the radial load F_r with the combined equivalent load P_0 provides the general case:

$$k = 4,05 + 0,3209 \cdot \left[\frac{P_0}{C_{\delta L} \cdot \left(\frac{G_r}{2} \right)^{1,08} \cdot Z \cdot \cos \alpha_0} \right]^{-0,7911} \tag{C.13}$$

With the elastic constant $C_{\delta L}$ and factor k known, Equation (C.13) allows calculating the load on the maximum loaded rolling element. This converges back to the form of Equation (C.8):

$$Q = \frac{P_0}{Z \cdot \cos \alpha_0} \cdot \mathrm{k} \qquad \text{(C.14)}$$

C.3.3.3 Contact pressure factors



Key
A principal plane 1
B principal plane 2
C body 1
D body 2

Figure C.5 – Nomenclature for bearing curvature

Figure C.5 depicts the planes under consideration and defines the nomenclature for the equation subscripts. Note that convex curvatures are positive, and concave curvatures are negative. The osculation S is the ratio of the radius of curvature of the inner ring to the radius of curvature of the roller, both in the principal plane 2 through the axis of the rolling element, see Equation (C.15):

$$\mathrm{S} = \frac{r_{22}}{r_{12}} \geq 1{,}0 \qquad \text{(C.15)}$$

The value of S by definition is greater than or equal to unity. ISO/TS 16281 provides standard values for the reference geometries for common bearing designs, resulting in values for S between 1,03 and 1,08. Actual bearing design may deviate from these standard value down to S = 1,01. To prevent numerical instability, the minimum value of S is set equal to 1,001.

With this definition, Equations (C.16) through (C.17) allow calculation of the curvature factors for point and line contact:

$$\rho_{11} = \frac{2}{D_w} \qquad \text{(C.16)}$$

for CRB and TRB (C.17)

$$\rho_{12} = 0$$

for SRB

$$\rho_{12} = -\rho_{22} \cdot \mathrm{S} \qquad \text{(C.18)}$$

$$\rho_{21} = \frac{2}{\dfrac{D_{pw}}{\cos \alpha_0} - D_w} \tag{C.19}$$

for CRB and TRB (C.20)

$$\rho_{22} = 0$$

for SRB (C.21)

$$\rho_{22} = \frac{-2}{\dfrac{D_{pw}}{\cos \alpha_0} + D_w}$$

$$\Sigma\rho_{\text{point}} = \rho_{11} + \rho_{12} + \rho_{21} + \rho_{22} \tag{C.22}$$

$$\Sigma\rho_{\text{line}} = \rho_{11} + \rho_{21} \tag{C.23}$$

$$\cos \tau = \frac{(\rho_{11} - \rho_{12} + \rho_{21} - \rho_{22})}{\Sigma\rho_{\text{point}}} \tag{C.24}$$

where

S is the osculation of the inner ring contact;

D_{pw} is the pitch diameter of the ball- or roller set in mm;

D_w is the ball or roller diameter in mm;

α_0 is the nominal contact angle of the bearing, in °.

With the curvature factors and curvature sums $\Sigma\rho_{\text{point}}$ and $\Sigma\rho_{\text{line}}$, the Hertzian coefficients required to compute stress can be calculated. Brandlein et al. (1999) table the values for μ and ν as function of $\cos \tau$. Equations (C.25) through (C.28) describe curve fits for these values.

if $\cos \tau > 0{,}87$ $$\mu = 1{,}396748 \cdot \cos \tau^{\,0{,}665242} \cdot (1 - \cos \tau)^{-0{,}37399} \tag{C.25}$$

else (C.26)

$$\mu = 5{,}6864 \cdot \cos \tau^4 - 6{,}0607 \cdot \cos \tau^3 + 2{,}7985 \cdot \cos \tau^2 + 0{,}35289 \cdot \cos \tau + 1{,}005$$

if $\cos \tau > 0{,}87$ (C.27)

$$\nu = 0{,}683241 \cdot \cos \tau^{\,0{,}4} \cdot (1 - \cos \tau)^{-0{,}189343}$$

else (C.28)

$$\nu = -0{,}30365 \cdot \cos \tau^3 + 0{,}373719 \cdot \cos \tau^2 - 0{,}67694 \cdot \cos \tau + 1{,}0014$$

With μ and ν, the dimensions of the Hertzian contact ellipse represented in Figure C.6 can be computed:

$$2 \cdot a \ = \ 0{,}0472 \cdot \mu \cdot \sqrt[3]{\frac{Q}{\Sigma\rho_{\text{point}}}} \tag{C.29}$$

$$2 \cdot b \ = \ 0{,}0472 \cdot \nu \cdot \sqrt[3]{\frac{Q}{\Sigma\rho_{\text{point}}}} \tag{C.30}$$

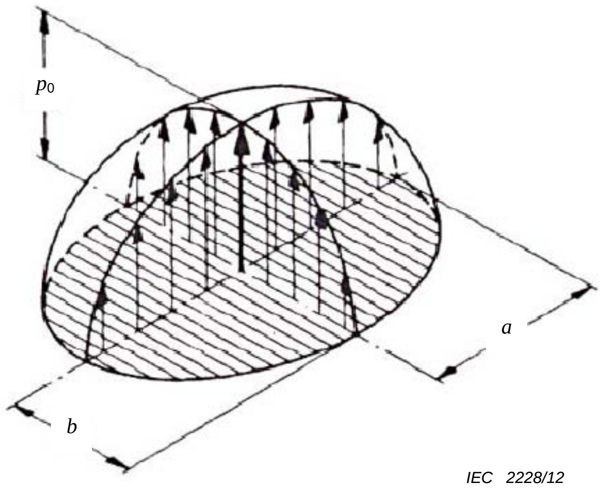


Figure C.6 – Stress distribution over the elliptical contact area

C.3.3.4 Contact pressure

Maximum contact pressure for SRB is calculated for elliptical contact considering osculation. Maximum contact pressure for CRB and TRB is calculated for line contact modified by an empirical factor that accounts for axial crowning of rollers and inner ring raceway.

Equations (C.31) and (C.32) describe the unadjusted contact pressure (Brandlein et al., 1999):

$$p_{\text{line}} \ = \ 270 \cdot \sqrt{\frac{1}{2} \cdot \frac{Q}{L_{we}} \cdot \Sigma\rho_{line}} \tag{C.31}$$

$$p_0 \ = \ \frac{858}{\mu \cdot \nu} \cdot \sqrt{Q \cdot (\ \Sigma\rho_{\text{point}}\ )^2} \tag{C.32}$$

For ball bearings, the maximum contact stress can be approximated from the static safety factor S_0 by

$$p_0 = 4200\ \text{N/mm}^2 \cdot S_0^{\frac{-1}{3}} \tag{C.33}$$

C.3.3.5 Misalignment factor, K_m

For cylindrical and taper roller bearings, the contact pressure is influenced by misalignment of the raceways, represented by shaft slope angle θ_L. K_m is set to unity for spherical roller bearings that freely accommodate misalignment:

$$\text{if } L_{we} / D_w < 1{,}3 \tag{C.34}$$

$$K_m = 0{,}00105 \cdot \theta_L^{\ 2} + 0{,}00406 \cdot \theta_L + 0{,}9976$$

$$\text{else} \tag{C.35}$$

$$K_m = 0{,}0042 \cdot \theta_L^{\ 2} - 0{,}0092 \cdot \theta_L + 1{,}013$$

where:

Θ_L is the misalignment slope of the shaft in arc minutes and

K_m is set to unity for spherical roller bearings, and values to larger than or equal to unity for other bearing types.

C.3.3.6 Truncation factor, C_T

If the calculated length of the contact 2a exceeds the effective roller length L_{we}, the contact is truncated. This is considered in the calculation of the maximum stress by the truncation factor C_T. Equation (C.36) provides an approximation of this factor:

$$C_T = 1 + \frac{4 \cdot \left(\frac{b}{a^3}\right) \cdot \sqrt{a^2 - \left(\frac{L_{we}}{2}\right)^2} \cdot \left[\frac{64}{105} \cdot a \cdot \left(a - \frac{L_{we}}{2}\right)^2 - \frac{40}{189} \cdot \left(a - \frac{L_{we}}{2}\right)^3\right]}{\pi ab - \frac{8}{3} \cdot (a - L_{we}) \cdot \frac{b}{a} \cdot \sqrt{a^2 - \left(\frac{L_{we}}{2}\right)^2}} \tag{C.36}$$

If the length of the contact ellipse 2a is less than the effective roller length L_{we}, C_T is set to unity.

C.3.3.7 Ratio of maximum to nominal line contact pressure

The ratio of the maximum to nominal line contact pressure for CRB and TRB have empirically been established by test calculations in accordance with ISO/TS 16281 for bearing inner ring diameters between 80 mm and 500 mm, and contact stresses between 800 MPa and 2 500 MPa. Equation (C.37) summarizes the curve fit of these test calculations:

$$K_{lc} = 1 + 3185 \cdot (p_{line})^{-1{,}3633} \tag{C.37}$$

The maximum contact stress for SRB can be calculated using the formula for elliptic contacts. In this case, K_{lc} is defined by Equation (C.38):

$$K_{lc} = \frac{p_T}{p_{line}} = \frac{c_T \cdot p_0}{p_{line}} \tag{C.38}$$

C.3.3.8 Calculating the contact stress

The maximum contact stress is the previously calculated contact pressure for line contact p_{line}, adjusted with factors K_m and K_{lc} described above:

$$p_{max} = K_{lc} \cdot K_m \cdot p_{line} \tag{C.39}$$

where

p_{max} is the maximum contact stress in MPa;

K_{lc} is the ratio of maximum contact pressure to contact pressure for line contact without misalignment;

K_m is the ratio of maximum contact pressure with misalignment to maximum contact pressure without misalignment.

C.4 Calculation of lubricant film thickness

Lubricant film thickness has a decisive impact on the performance and reliability of rolling element bearings. For life calculations, ISO 281 contains an empiric approximation suitable for most commercial lubricants.

If alternative lubricants are used, the following methods may be helpful to approximate actual film thickness with these lubricants since experimental verification of real film thickness is difficult.

There are many methods that have been developed for calculating the film thickness for rolling element bearings. These depend on the type of contact, either line or point, and the location of contact, either minimum or average thickness. There is no consistent set of equations that are used by all bearing manufacturers. Most modern wind turbines use line contact bearings (i.e. TRB, SRB, CRB) and some point contact bearings (i.e. DGBB). The typical contact patch of a DGBB has an axis ratio that is close to line contact. Therefore only line contact is considered here.

The purpose of this clause is to explain how to evaluate and compare bearing viscosity ratio and film thickness differences between synthetic and mineral oils.

The calculation of κ per ISO 281 assumes typical values for the pressure viscosity coefficient of mineral oils. Many wind turbine gearboxes today utilize synthetic oils, which have different pressure viscosity values. This difference may affect the film thickness calculations.

According to ISO 281, it is generally not necessary to account for the difference between mineral and synthetic oils since the larger viscosity index of synthetic oils is compensated for by a larger pressure-viscosity coefficient for mineral oils. Therefore, the same oil film is formed at different operating temperatures if both oil types have the same viscosity at 40 °C.

In ISO 281 κ is defined as the ratio of the actual kinematic viscosity ν to the reference kinematic viscosity ν_1:

$$\kappa = \frac{\nu}{\nu_1} \tag{C.40}$$

Lambda Λ is the ratio of the film thickness h to the composite surface finish:

$$\Lambda = \frac{h}{\sqrt{s_R^2 + s_{RE}^2}} \tag{C.41}$$

where

s_R is the rms roughness of the raceway;

s_{RE} is the rms roughness of the rolling element.

ISO 281 proposes following relationship between Λ and κ:

$$\kappa \approx \Lambda^{1,3} \tag{C.42}$$

This relationship is derived from the Dowson & Higginson equation for line contact

$$\overline{H}_0 = \frac{h_{\min}}{R} = 2{,}65\overline{U}^{0{,}7} G^{0{,}54} \overline{Q}^{-0.13} = 2{,}65\left(\frac{\eta_0 U}{2E'R}\right)^{0{,}7} (aE')^{0{,}54} \left(\frac{Q}{lE'R}\right)^{-0{,}13} \tag{C.43}$$

where

$$E' = \frac{E}{1-\xi^2} \tag{C.44}$$

and therefore

$$h_{\min} = 2{,}65R\left(\frac{\eta_0 U}{2E'R}\right)^{0{,}7} (aE')^{0{,}54} \left(\frac{Q}{lE'R}\right)^{-0{,}13} \tag{C.45}$$

and for inner raceway film thickness calculations

$$R = \frac{D_w}{2}(1-\gamma) \tag{C.46}$$

$$\begin{aligned} U_i &= \frac{D_{pw}}{2}\left[(1-\gamma)(\omega_i - \omega_m) + \gamma\omega_R\right] = \\ &\frac{D_{pw}}{2}\left[(1-\gamma)\left(\omega_i - \frac{1}{2}\left[\omega_i(1-\gamma) + \omega_o(1+\gamma)\right]\right) + \frac{\gamma D_{pw}}{2D_w}(1-\gamma)(1+\gamma)(\omega_0 - \omega_i)\right] \end{aligned} \tag{C.47}$$

$$\gamma = \frac{D_w \cos\alpha}{D_{pw}} \tag{C.48}$$

In order to relate κ for synthetic and mineral oils, the following equations can be used:

$$\kappa_{\text{syn}} = \kappa\left(\frac{\Lambda_{\text{syn}}}{\Lambda_{\text{mineral}}}\right)^{(x)(y)} \tag{C.49}$$

where

x is the exponent of G in the film thickness equation;

y is the exponent of Λ in the κ – Λ equation.

By combining the Dowson & Higginson equation for line contact with Equation (C.42) and

$$\kappa_{\text{syn}} = \kappa\left(\frac{\Lambda_{\text{syn}}}{\Lambda_{\text{mineral}}}\right)^{(0{,}54)(1{,}3)} \tag{C.50}$$

the film thickness ratio κ for synthetic oils can be approximated as:

$$\kappa_{\text{syn}} = \kappa\left(\frac{\Lambda_{\text{syn}}}{\Lambda_{\text{mineral}}}\right)^{(0{,}70)} \tag{C.51}$$

C.5 Documentation of bearing rating

Bearing rating as described in 7.3.8 produces a wide variety of data that may be difficult to evaluate. Annex G suggests a standardized reporting format for these calculations.

C.6 Bearing types

Guidance for selecting appropriate bearing types and bearing arrangements for a wind turbine gearbox is provided in Table C.3 through Table C.6. These guidelines should not replace a detailed analysis of each bearing and bearing arrangement during the design phase. Special care should be taken before using bearing types or bearing arrangements not shown or described as having limited experience.

Bearing types shown in Table C.3 can support radial and double direction axial loads, i.e. combined loads, acting simultaneously.

Bearing types shown in Table C.4 can support radial and single direction axial loads acting simultaneously. They may be used as a locating bearing in combination with an additional bearing to support the axial load in two directions or as cross-locating bearing.

Bearing types shown in Table C.5 can support radial loads only. They may be used as a non-locating bearing or in combination with an additional bearing to support the axial load. Outer and inner rings should be axially fixed in their journals.

Bearing types shown in Table C.6 support axial loads. They should be used only in combination with an additional bearing to support the radial load. The additional bearing should be a pure radial bearing to avoid axial load sharing. It should be ensured that the axial bearing itself does not share any radial load, for example, by providing clearance in the radial direction at the outer ring or appropriate internal clearance of the bearing. Appropriate means should be taken to prevent the outer ring from spinning, for example, by a locking pin.

Table C.3 – Bearing types for combined loads with axial loads in double directions

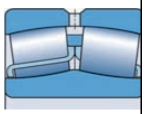
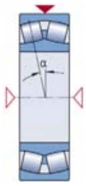
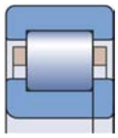
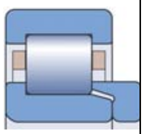
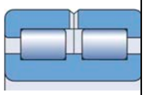
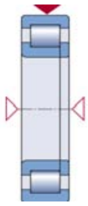
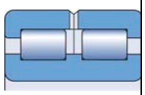
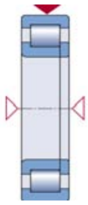
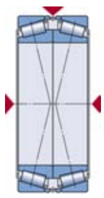
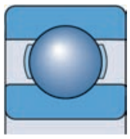
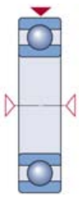
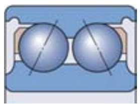
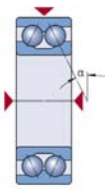
Type	Abbreviation	General comments	Symbol	Load
Spherical roller bearing	SRB	The high load carrying capacity and misalignment capability of the SRB can be advantageous in wind turbine gearboxes. Internal load distribution and movements under the different and fast changing operating conditions should carefully be analysed.		
Cylindrical roller bearing Double row cylindrical roller bearing	CRB NUP and NJ+HJ design DRCRB	This bearing type can accommodate light to moderate axial loads only. Furthermore the ratio F_a/F_r should be small. Rapid load reversals should be avoided. NUP and NJ+HJ type may be used as locating bearings, if the necessary precautions are taken to prevent looseness between the bearing and the HJ ring. If this bearing is chosen for supporting axial load, flange strength of the inner and outer rings should be carefully evaluated in respect to fatigue bending and shock loads, as well as heat dissipation. Proper support of the flange by a well-designed abutment is mandatory. Where the required capacity allows, caged CRB should be preferred to FCCRB.	  	
Double row full complement cylindrical roller bearing	DRFCCRB	Wear due to roller-roller contact may limit the life of full complement bearings, and hence they should only be used with special care. Full complement bearings should only be selected where centrifugal forces do not contribute significantly to contact forces between rollers. If significant contact forces occur, special considerations need to be taken. If this bearing is chosen for supporting axial load, flange strength of the inner and outer rings should be carefully evaluated in respect to fatigue bending and shock loads, as well as heat dissipation. Proper support of the flange by a well-designed abutment is mandatory.		
Double row taper roller bearing	DRTRB	For double row or paired single row taper roller bearings appropriate mounted clearance or preload setting is essential for proper performance. The mounted clearance or preload setting can be made by various methods or by using a preset assembly of paired bearings with spacers. The operating conditions throughout the load spectrum should be considered when determining the optimum clearance or preload setting.		
Deep groove ball bearing	DGBB	The suitability of this type may be restricted by its limits on load carrying capacity.		
Double row angular contact ball bearing	DRACBB	The suitability of this type may be restricted by its limits on load carrying capacity.		

Table C.4 – Bearing types for combined loads with axial loads in single direction

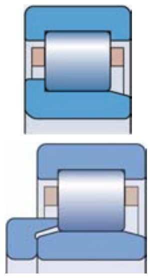
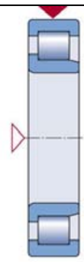
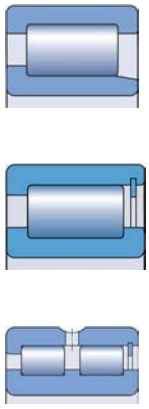
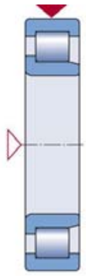
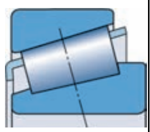
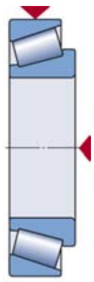
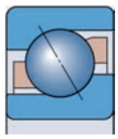
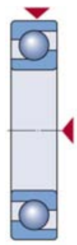
Type	Abbreviation	General comments	Symbol	Load
Cylindrical roller bearing	CRB NJ and NU+HJ design	This bearing type can accommodate light to moderate axial loads only. Furthermore the ratio F_a/F_r should be small. Rapid load reversals should be avoided. If this bearing is chosen for supporting axial load, flange strength of the inner and outer rings should be carefully evaluated in respect to fatigue bending and shock loads, as well as heat dissipation. Proper support of the flange by a well-designed abutment is mandatory. Where the required capacity allows, caged CRB should be preferred to FCCRB.		
Full complement cylindrical roller bearing Double row full complement cylindrical roller bearing	FCCRB NJG and NCF design DRFCCRB	Full complement bearings are used in single and double row execution. Wear in the roller-roller contact may limit the life of full complement bearings, and hence they should only be used with special care. Full complement bearings should only be selected where centrifugal forces do not contribute significantly to contact forces between rollers. If significant contact forces occur, special considerations need to be taken. This bearing type can accommodate light to moderate axial loads only. Furthermore the ratio F_a/F_r should be small. If this bearing is chosen for supporting axial load, flange strength of the inner and outer rings should be carefully evaluated in respect to fatigue bending and shock loads, as well as heat dissipation. Proper support of the flange by a well-designed abutment is mandatory. Where the required capacity allows, caged CRB should be preferred to FCCRB.		
Taper roller bearing	TRB	For taper roller bearings appropriate mounted clearance or preload setting is essential for proper performance. The operating conditions throughout the load spectrum should be considered when determining the optimum clearance or preload setting. Single row TRB can be used in cross-locating arrangements only when the internal clearance or preload is maintained within acceptable limits during all operating conditions. A careful evaluation should be performed to determine the acceptable clearance range to avoid roller skidding, overheating, and other operational problems. Single row tapered roller bearings are preferably used as paired bearings for combined loads and/or to axially locate the shaft (see Table C.3).		
Angular contact ball bearing	ACBB	The suitability of this type may be restricted by its limits on load carrying capacity.		

Table C.5 – Bearing types for pure radial load

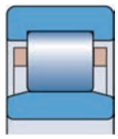
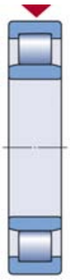
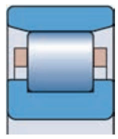
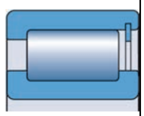
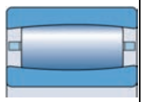
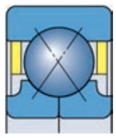
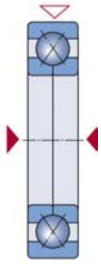
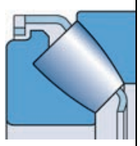
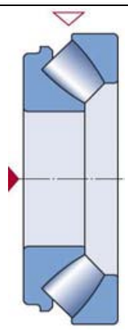
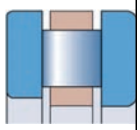
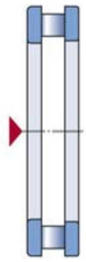
Type	Abbreviation	General comments	Symbol	Load
Cylindrical roller bearing	CRB NU design	This type has an outer ring with two flanges and inner ring without flanges. The outer ring can retain a small amount of oil to avoid dry start-up.		
Cylindrical roller bearing	CRB N design	N-designs may be considered on rotating shaft or rotating outer ring to minimize rollers skidding at the inner ring contact under low load running conditions.		
Full complement cylindrical roller bearing	FCCRB	Wear due to roller-roller contact may limit the life of full complement bearings, and hence they should only be used with special care. Full complement bearings should only be selected where centrifugal forces do not contribute significantly to contact forces between rollers. If significant contact forces occur, special considerations should be taken. Where the required capacity allows, caged CRB should be preferred to FCCRB.		
Torroidal roller bearing	TORB	Axial displacement of the inner ring relative to the outer ring affects the radial clearance of the bearing. This effect should be considered when determining the gear mesh alignment.		

Table C.6 – Bearing types for axial load

Type	Abbreviation	General comments	Symbol	Load
Four-point contact ball bearing	FPCBB	Alternating axial loads may cause oscillating contact points and excessive cage wear. Application of an FPCBB should therefore be avoided in applications with loads alternating at high frequencies.		
Spherical roller thrust bearing	SRTB	This bearing should have sufficient axial load under all operating conditions – refer to bearing manufacturer's recommendations.		
Cylindrical roller thrust bearing	CRTB	This bearing should have sufficient axial load under all operating conditions – refer to bearing manufacturer's recommendations. CRTB have unfavourable sliding and requires special properties of the lubricant. The bearing selected should have a small mean diameter and a small sectional height to minimize sliding.		

C.7 Bearing arrangements

C.7.1 General

Pinions should be mounted between bearings. The only exception to this is the sun pinion, where overhung pinions are used. Sun pinions should be designed without bearings to achieve load sharing between planet gears.

A bearing arrangement generally requires two bearing positions, one on each side of the pinion, to locate the shaft radially and axially. The arrangement may consist of locating and non-locating bearing arrangements or cross-locating bearing arrangements.

Examples of the two principle concepts are shown in Figure C.7 and Figure C.10.

C.7.2 Locating and non-locating bearing arrangements

In a locating and non-locating bearing arrangement the locating bearing(s) provides radial support and locates the shaft axially in two directions. The locating bearing should, therefore, be fixed in position both on the shaft and in the housing. The non-locating bearing at the other end of the shaft provides radial support only. It must also enable axial displacements to accommodate shaft length thermal variations.

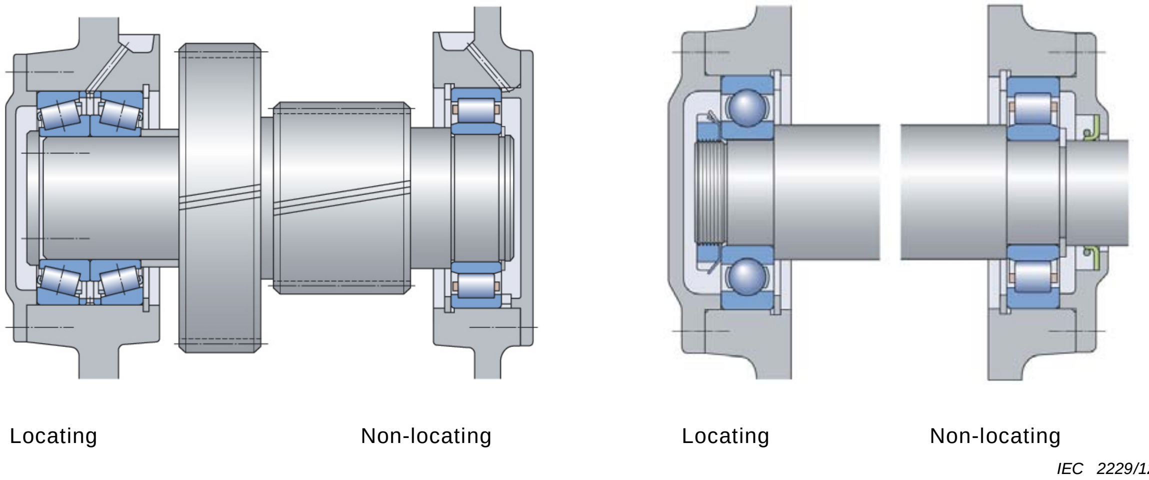


Figure C.7 – Examples of locating and non-locating bearing arrangements

The locating function can be achieved by one bearing suitable for combined loads (Table C.3) or by paired bearings that can accommodate axial loads in one direction (Table C.4) in a face-to-face or back-to-back arrangement or by combining a pure radial bearing (Table C.5) and a thrust bearing (Table C.6), see Figure C.8.

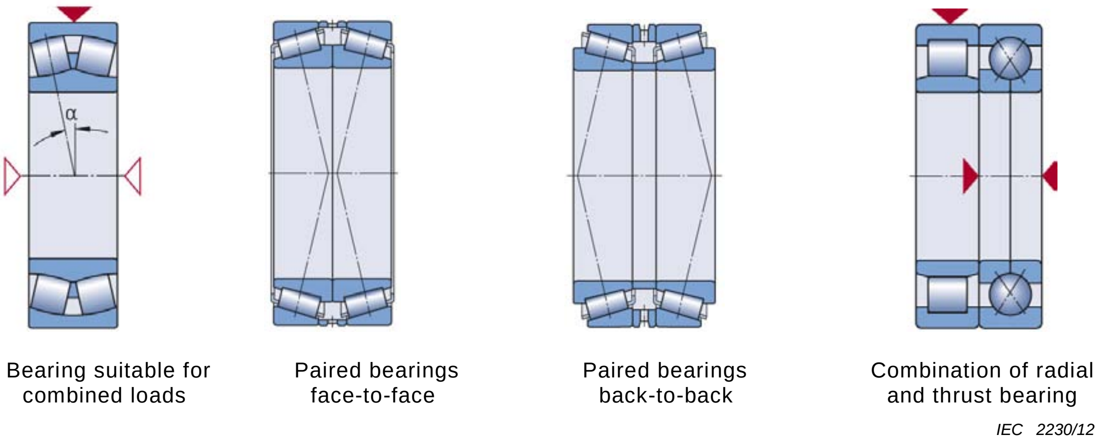


Figure C.8 – Examples of locating bearing arrangements

The non-locating function can be achieved by a bearing suitable for pure radial loads (Table C.5) or a bearing suitable for combined loads (Table C.3 and Table C.4) with one ring floating on its shaft or in its housing bore, see Figure C.9. Bearings where the axial displacement takes place within the bearing at very low friction (like CRB (NU and N designs) or TORB) are preferred.

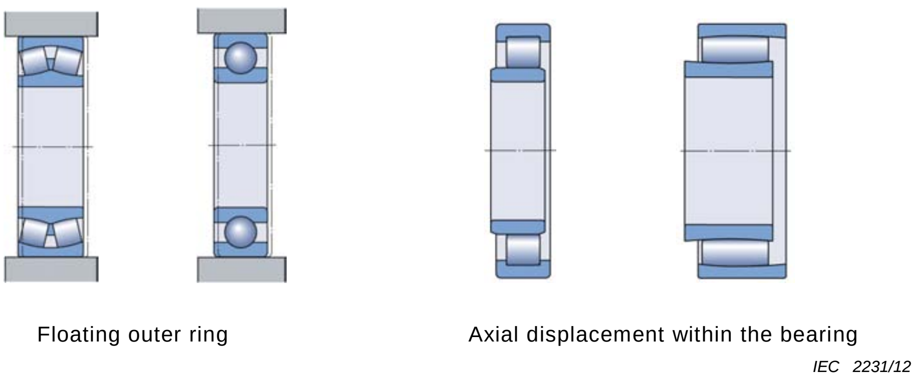


Figure C.9 – Examples of accommodation of axial displacements

C.7.3 Cross-locating bearing arrangements

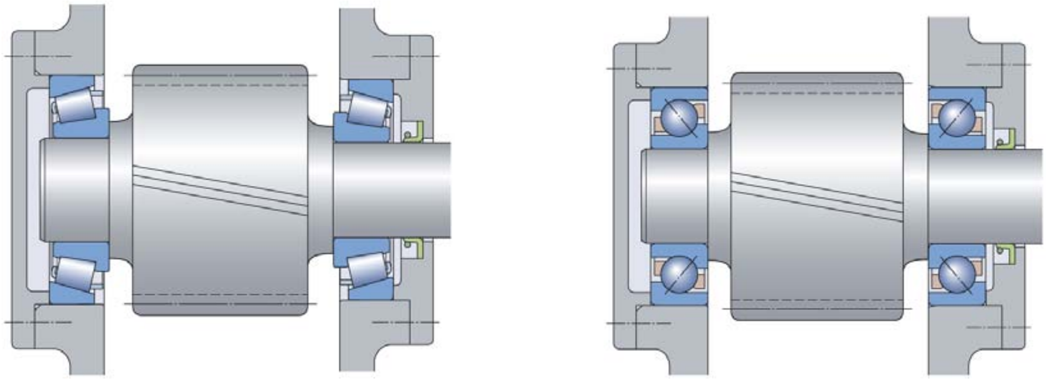
In cross-locating bearing arrangements the shaft is axially located in one direction by one bearing position and in the opposite direction by the other bearing position, see Figure C.10.

The cross-locating function can be achieved by bearings that can accommodate axial loads in one direction (Table C.4) or bearings suitable for combined loads (Table C.3).

The axial clearance should be sufficient to accommodate the axial displacements from shaft length changes due to thermal variations.

The housing deformation in operation might influence the axial bearing setting and should therefore be considered in the bearing rating life and the design of the gear tooth profile.

Cross-locating bearing arrangement is generally used for short shafts.



Cross-locating Cross-locating Cross-locating Cross-locating

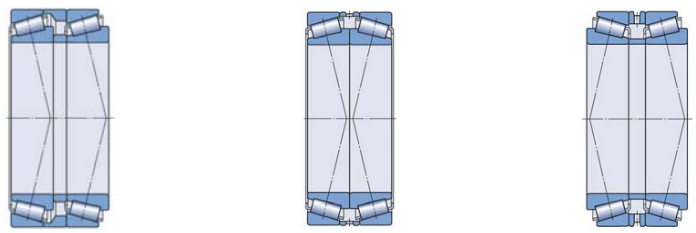
IEC 2232/12

Figure C.10 – Examples of cross-locating bearing arrangements

C.7.4 Paired mounting

Paired mounting is used when the load carrying capacity of a single bearing is inadequate (tandem arrangement) or when combined or axial loads act in both directions (back-to-back or face-to-face arrangements), see Figure C.11.

Bearings mounted back-to-back provide a relatively stiff bearing arrangement that can also accommodate tilting moments. Bearings mounted face-to-face are a less stiff arrangement and is less suitable for the accommodation of tilting moments.



Tandem Face-to-face Back-to-back

IEC 2233/12

Figure C.11 – Examples of bearing arrangements with paired mounting

C.7.5 Bearing arrangement selection matrixes

Each shaft should be supported at two bearing positions. Suitable bearings for locating and non-locating bearing arrangements and cross-locating bearing arrangements respectively are shown in the selection matrixes Table C.8 through Table C.12 for the different gearbox positions. For interpretation of the matrixes refer to Table C.7.

Depending on the speed range, Table C.9 (LSIS) or Table C.10 (HSIS) may apply for selection of bearings for the medium speed intermediate shaft (MSIS). Table C.11 (HSS) applies for the selection of bearings on the high speed shaft. Gearboxes with more than one planet stage are generally not covered by these recommendations.

Planet bearings (Table C.12) where the outer ring is integrated in the gear wheel can provide more space for the rolling elements. Thus higher load carrying capacity can be achieved. It is also beneficial, if outer ring movement cannot be prevented.

If an integrated bearing arrangement is selected the following items should be agreed upon by gear and bearing manufacturers:

- material selection;
- hardness and micro-structure;
- raceway dimensions and tolerances;
- surface roughness;
- manufacturing processes.

The following rules apply to Table C.8 through Table C.12.

- The symbol for a NJ-CRB is used for all caged CRB. However, NU- or N-design bearings should be used in non-locating positions.
- If an axial bearing (Table C.6) is chosen as the locating bearing, this implicitly means that this axial bearing should be combined with a radial bearing. The radial bearing should be selected from the “non-locating” row, preferably one of the pure radial bearings (Table C.5).
- Where TRB or ACBB are recommended as locating bearings, this always refers to paired bearings arranged face-to-face or back-to-back or to double row bearings.

Table C.7 – Bearing selection: Legend

Legend	Description	General comments
●	Suitable	This bearing type has proven suitable.
◐	Suitable with restrictions	This bearing type may be suitable if certain conditions and additional precautions are complied with.
⊖	Unsuitable	This bearing type cannot fulfil the requirements in this specific bearing position.
ⓘ?	Limited experience	This bearing type has insufficient, if any, field experience to validate acceptable performance in this position. Use of this bearing type should be preceded by detailed theoretical analysis, acknowledged by the bearing manufacturer, and appropriate testing for the specific application.

Key for Table C.8 to Table C.12.

1) When this bearing type is used as non-locating bearing or in a cross-locating arrangement, the outer ring fit should allow accommodation of thermal expansion by movement of the ring in the housing bore. The outer ring should be prevented from spinning.

2) Single-row TRB or ACBB in cross-locating arrangement should only be used where endplay can be controlled within acceptable limits under typical operating conditions.

3) Performance of paired double-row TRB or ACBB depends on appropriate internal clearance or preload.

4) The NJ inner ring should be mounted offset to enable axial movement or clearance. The outer ring should be prevented from spinning.

5) This is intended for bearings that can take thrust in both directions.

6) Influences of variation of loads and shaft movement (amplitude and frequency) within the bearing internal clearance should be carefully analyzed.

7) Sliding motion occurs in roller contacts. This bearing should be spring loaded.

8) This design is successful when used with short shafts that do not have large thermal expansions and housings with small elastic deformations.

9) Review cooling requirements when using this bearing type.

10) Special attention to alternating axial loads, for example from couplings, is required when using this bearing type.

11) Two SRB may be used when appropriate fit allows sufficient axial movement of the inner ring for load sharing. The risk of wear on the inner rings and their shaft should be evaluated. When closely spaced, total capacity may be less than the two bearing single capacities (see ISO 281 for details).

12) Full-complement bearings should be used only for low speed stages where centrifugal forces do not contribute to contact forces between rollers.

13) The helix angle generates a tilt moment on the planet that causes uneven load sharing between the bearing rows. The elastic deflection of the planet wheel along the width can also affect load sharing. These effects should be considered in detail.

14) Cross-locating TRB in planet wheels should be assembled with zero clearance or light pre-load. This enables a stiff support for the mesh.

15) Rotating outer ring in combination with misaligned axis of inner ring relative to the outer ring causes dynamic misalignment with axial sliding of rollers against the outer ring.

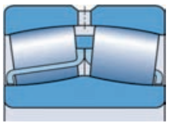
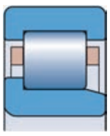
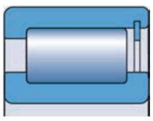
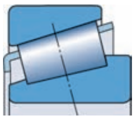
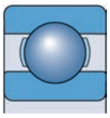
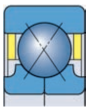
16) Optimise series and cages to cope with low load conditions to avoid smearing damage.

17) This bearing type has a perceived increased failure rate in some applications under some conditions.

18) Limited experience, verify robustness against high load.

19) The influence of deflections and misalignment of surrounding parts needs to be evaluated with respect to the contact conditions of FCCRB. Where required capacity allows, caged CRB should be preferred.

Table C.8 – Bearing selection: Low speed shaft (LSS) / planet carrier

Bearing type		**Locating**	**Non-locating**	**Cross-locating**
	SRB	◐ 6	◐ 1	◐ 1,6
	CRB	? 5	● 4	●
	FCCRB	? 5	◐ 4	●
	TRB	◐ 3	⊖	◐ 2
	DGBB	?	◐ 1	◐ 1
	FPCBB	?	⊖	⊖

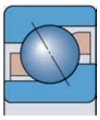
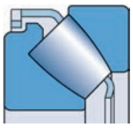
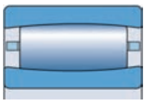
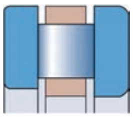
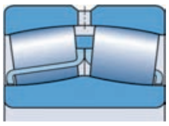
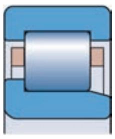
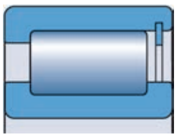
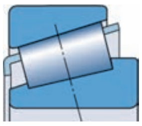
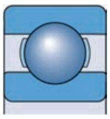
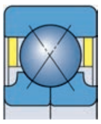
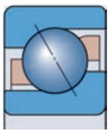
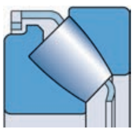
Bearing type		Locating	Non-locating	Cross-locating
	ACBB	?	–	? 2
	SRTB	?	–	–
	TORB	–	?	–
	CRTB	?	–	–

Table C.9 – Bearing selection: Low speed intermediate shaft (LSIS)

Bearing type		Locating	Non-locating	Cross-locating
	SRB	◐ 6,17	◐ 1	◐ 1,6
	CRB	? 5	◐ 4	◐ 4
	FCCRB	? 5, 19	◐ 4, 19	◐ 4, 19
	TRB	◐ 3	–	◐ 2
	DGBB	?	◐ 1	◐ 1
	FPCBB	◐ 10	–	–
	ACBB	? 3	–	? 2
	SRTB	●	–	–

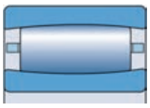
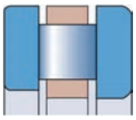
Bearing type		Locating	Non-locating	Cross-locating
	TORB	⊖	●	⊖
	CRTB	◐ 7	⊖	⊖

Table C.10 – Bearing selection: High speed intermediate shaft (HSIS)

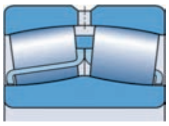
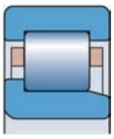
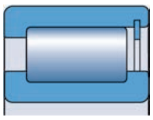
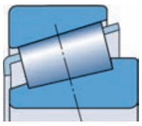
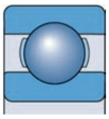
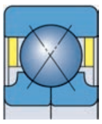
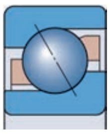
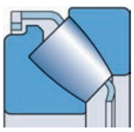
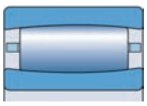
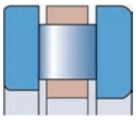
Bearing type		Locating	Non-locating	Cross-locating
	SRB	◐ 6,17	◐ 1,17	◐ 1,6,17
	CRB	◐ 8,9	● 4	◐ 9
	FCCRB	⊖	⊖	⊖
	TRB	◐ 3	?	◐ 2
	DGBB	?	?	?
	FPCBB	◐ 10	⊖	⊖
	ACBB	?	⊖	⊖
	SRTB	?	⊖	⊖
	TORB	⊖	●	⊖
	CRTB	?	⊖	⊖

Table C.11 – Bearing selection: High speed shaft (HSS)

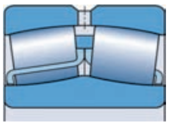
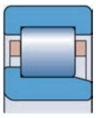
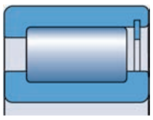
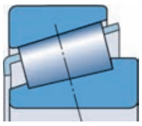
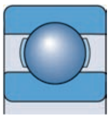
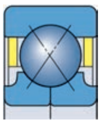
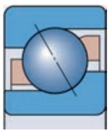
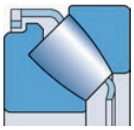
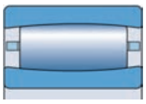
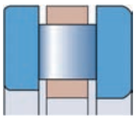
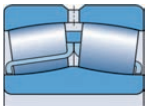
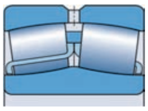
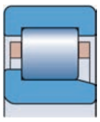
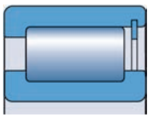
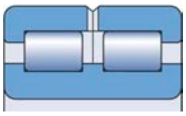
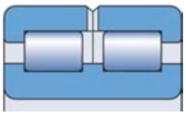
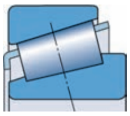
Bearing type		Locating	Non-locating	Cross-locating
	SRB	◐ 6,17	◐ 1,17	⊖
	CRB	(?)	◐ 16	⊖
	FCCRB	⊖	⊖	⊖
	TRB	◐ 3	(?)	◐ 2,8
	DGBB	◐ 18	◐ 1, 18	⊖
	FPCBB	◐ 10, 18	⊖	⊖
	ACBB	◐ 18	⊖	⊖
	SRTB	(?)	⊖	⊖
	TORB	⊖	●	⊖
	CRTB	⊖	⊖	⊖

Table C.12 – Bearing selection: Planet bearing

Bearing type		Arrangement	Spur gear	Single helical gear
	SRB	Single	◐ 15,17	⊖
	SRB	Two bearings	◐ 11,17	◐ 11,17
	CRB	Two bearings cross-locating	●	●
	FCCRB	Two bearings cross-locating	◐ 12	◐ 12
	DRFCCRB	Single	◐ 12	◐ 12,15
	DRFCCRB	Two bearings	◐ 12	◐ 12,13
	TRB	Two bearings cross-locating	◐ 14	◐ 14

Annex D
(informative)

Considerations for gearbox structural elements

D.1 General

This annex provides information in support of the analysis requirements for structural elements in 7.5.

D.2 Deflection analysis

Since gears and bearings are very sensitive to misalignment, deflection and stiffness requirements are important in a successful wind turbine gearbox design. Component and housing stiffness should also be sufficient to avoid resonant frequencies, which could contribute to excessive stress, noise and vibration.

Experience has shown that a detailed deflection analysis of the housing and bearing bores is needed to fully understand their shape when loaded. Bearing bores can distort, tilt and move due to housing deflection, and this may affect the internal stresses in the bearings.

All forces, moments and displacements, both across the interfaces and internally generated, should be included in the deflection analysis. The influence of torque arm bushings should also be included.

Ring gears in planetary designs usually are part of the housing and transmit a significant amount of torque. They distort as a result of both internal forces from the planet gears and all external forces on the housing.

Planet carriers in planetary designs also require detailed deflection analysis to determine their torsional and bending deflections during operation. All forces, moments and displacements generated by the rotor should be included if the carrier is rigidly attached to the rotor shaft.

Temperature gradients within the gearbox may cause thermal distortion sufficient to adversely affect alignment and fits of bearings, gears and shafts.

The accuracy of these calculations is strongly dependent on the boundary condition assumptions. Care should be taken in analysing results since deflections that are very small in relation to the global model may be significant.

Table D.1 – Typical material properties

Material	E MPa	ν	Unit mass kg/m^3	Ultimate strain	Material behaviour
Structural steel	$2,1 \times 10^5$	0,3	7 850		Ductile
Nodular cast iron	$1,7 \times 10^5$	0,275	7 200	$\geq$ 12,5 %	Ductile
Nodular cast iron	$1,8 \times 10^5$	0,275	7 200	< 12,5 %	Brittle
Lamellar cast iron	$0,88 – 1,13 \times 10^5$	0,26	7 200		Brittle

D.3 Material properties

Typical properties of materials commonly used in FE calculations of structural components of gearboxes are listed in Table D.1, as an example.

D.4 Global and local failures

A component does not always fail due to local stress at a notch. Global failure can occur at un-notched sections or sections that are less severely notched, e.g. section B in Figure D.1. Figure D.2.a) shows component failure due to notch stress, while Figure D.2.b) shows component failure due to section stress.

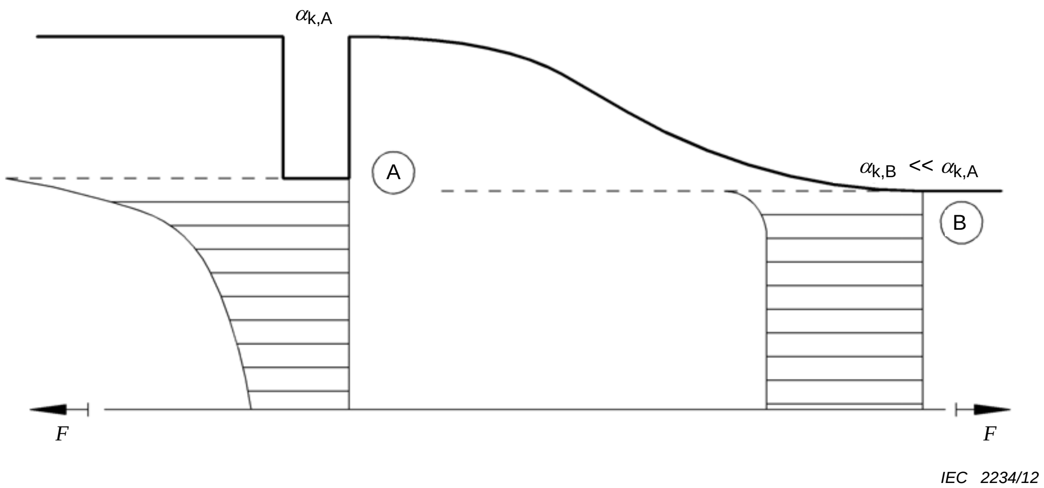


Figure D.1 – Locations of failure for local (A) and global (B) failure

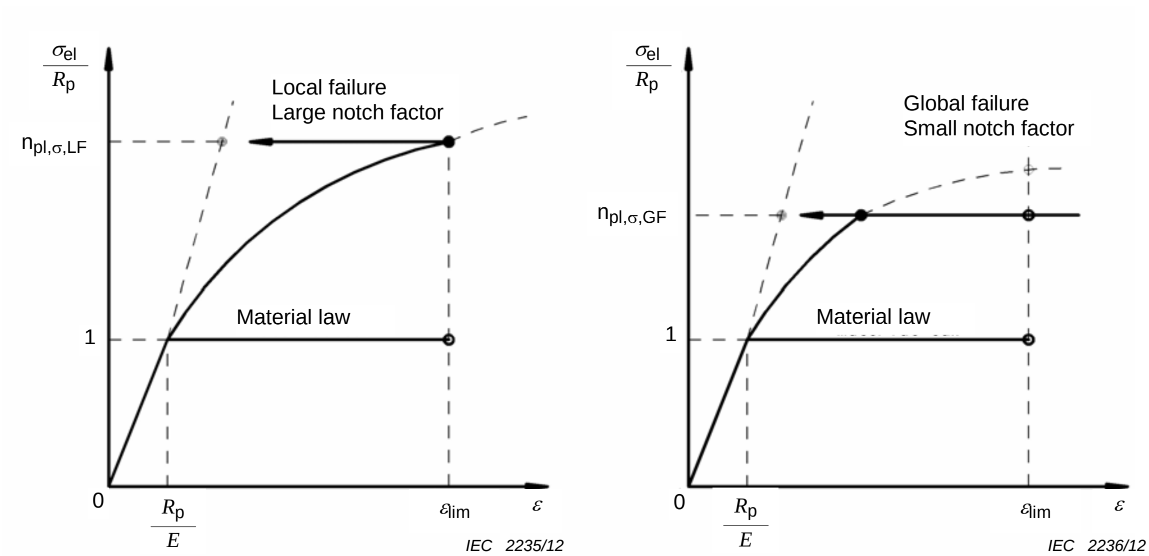


Figure D.2.a) – Limiting strain ε_{lim} is reached before the plastic load limit

Figure D.2.b) – Plastic load limit is reached before the strain limit ε_{lim}

Figure D.2 – Local and global failure for two different notch radii

D.5 Mean stress influence

In general the fatigue strength is sensitive to mean stress, which may be considered by use of a Haigh-diagram (Figure D.3). In the Haigh-Diagram the mean stress dependent curve of the fatigue strength $\sigma_{A,R}$ is given.

For the definition of characteristic values for fatigue testing, like stress amplitude σ_A, stress ratio R etc. refer to ISO 12107.

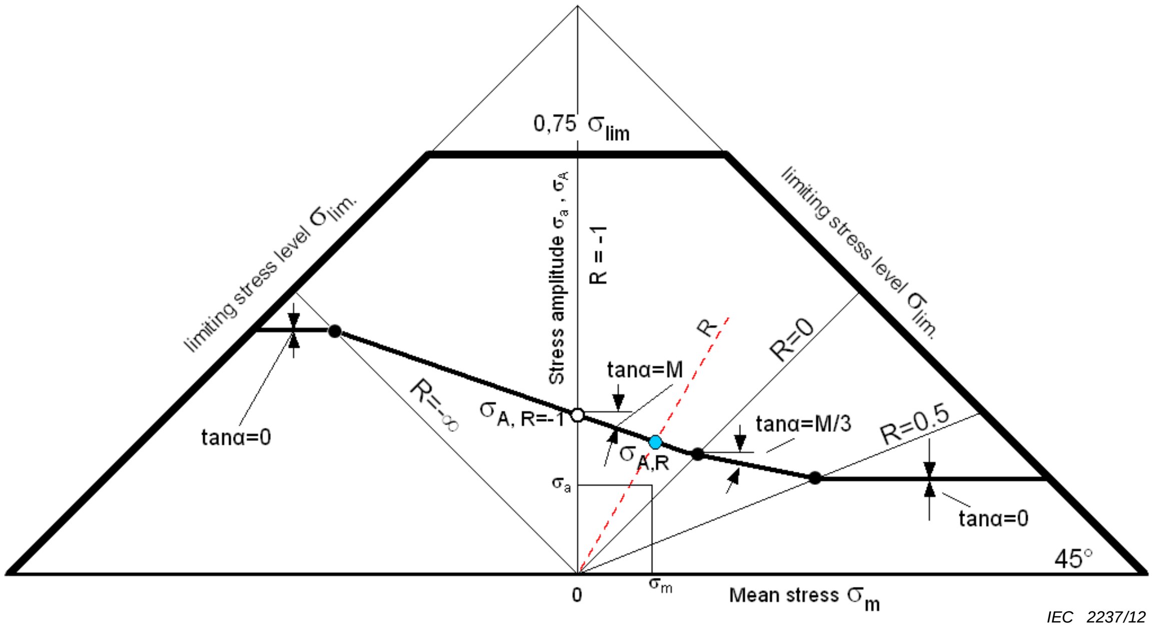


Figure D.3 – Haigh-diagram for evaluation of mean stress influence (Haibach, 2006)

The occurring stresses are restricted by the limiting stress level, σ_{lim}. The stress amplitude σ_a is limited by 0,75*σ_{lim} (Neuber et al., 2003). For this purpose, σ_{lim} is reduced by the partial factor for consequence of failure, γ_n.

The various slopes of the curve are defined by the mean stress sensitivity M, which is defined as follows:

$$M = \frac{\sigma_D(R=-1)}{\sigma_D(R=0)} - 1 \tag{D.1}$$

The fatigue strength amplitude adjusted for the stress ratio is:

for $-\infty < R \le 0$

$$\sigma_{A,R} = \sigma_{A,R=-1} \cdot [1]/[1 + M \cdot (1+R)/(1-R)] \tag{D.2}$$

for $0 < R \le 0,5$

$$\sigma_{A,R} = \sigma_{A,R=-1} \cdot [(1+M/3)/(1+M)]/[(1+M/3)\cdot(1+R)/(1-R)] \tag{D.3}$$

for $0,5 < R \le 1$

$$\sigma_{A,R} = \sigma_{A,R=-1} \cdot [(1+M/3)/(1+M)^2] \tag{D.4}$$

for $1 < R \le +\infty$

$$\sigma_{A,R} = \sigma_{A,R=-1} \cdot [1/(1-M)] \tag{D.5}$$

Annex E
(informative)

Recommendations for lubricant performance in wind turbine gearboxes

E.1 Purpose

E.1.1 Lubricant considerations

Wind turbine gearboxes can be quite varied in their design and configuration. As such, lubrication requirements may vary among gearboxes and also by their operating environment, i.e., ambient conditions, duty cycle, etc. The information contained in this annex is designed to provide users (end users, owners, turbine manufacturers, and component suppliers) with a guideline for a minimum level of lubricant performance for this application. Additionally, guidelines for condemning limits of selected lubricant parameters are offered based on experience in the industry. This information is not meant to replace any equipment builder or component supplier specification requirements. A more detailed discussion of lubricant types, additives, filtration methods, condition monitoring and oil analysis can be found in Annex F of ANSI/AGMA/AWEA 6006-A03.

Additional guidance is provided in this clause on maintaining lubricant cleanliness.

E.1.2 Lubricant selection guidelines

The key functions of the lubricant are to minimize the friction and wear between surfaces in relative motion and remove heat generated by the mechanical action of the system. To accomplish these tasks the lubricant must have sufficient viscosity in order to separate the mating surfaces as much as possible, and the appropriate chemical additive system to minimize thermal and oxidative degradation and promote anti-wear performance.

The choice of the appropriate lubricant depends in part on matching its properties to the particular application. A detailed elasto-hydrodynamic (EHD) analysis of the gearbox is the most desirable and thorough assessment of the gear lubrication requirements, but this is not always practical due to the amount of information required. For more information about this approach, the reader should review the information provided in ANSI/AGMA 925-A02. In the absence of detailed information about gear geometry, loading, etc., it is recommended that the user follow the guidelines in this annex.

However, the user should still ascertain certain performance attributes for the gearbox to make a reasonable lubricant selection. The user should be prepared to:

- determine the type of gearing used in the transmission;
- determine selected operating conditions, such as:
  - ambient temperature;
  - operating temperature;
  - operating speed range;
- determine any critical special circumstances, such as:
  - low temperature start-up;
  - ambient temperatures above 50 °C;
  - high, transient loads.

Applying the above information to Figure E.1, one can estimate the appropriate viscosity for the particular application based on the effective operating temperature the gears will see in service. Since wind turbine applications usually involved enclosed gear drives that operate under a range of loads, Figure E.1 and Table E.2 through Table E.4 offer a minimum performance level for enclosed gear systems requiring anti-scuff protection.

E.2 Lubricant physical characteristics

E.2.1 Viscosity grade

The viscosity grade should be chosen on the basis of the gearbox bulk oil operating temperature, the viscosity index of the fluid, and the pitch line velocity of the gears. In multistage gearboxes commonly used in wind turbine applications there is usually a large discrepancy between the input and output shaft speeds. In these cases the viscosity grade should be based on the low speed input gear to ensure an adequate film is developed. An estimate of the proper viscosity can be made using Equation (E.1). A graphical representation of the response is shown in Figure E.1. Further, Table E.1 through Table E.4 provide the ISO viscosity grades for various operating temperature, speed, and viscosity index fluids. These are conservative estimates based on the viscosity – velocity function shown in Equation (E.1) and assuming Newtonian behaviour of the fluid, i.e., no loss of viscosity due to shear. There are several alternative methods to calculate the appropriate viscosity grade such as DIN 51509-1, ANSI/AGMA 925-A02or in-house methods from lubricant suppliers.

$$\nu_{40} = 500 \cdot \nu_t^{-0,5} \qquad \text{(E.1)}$$

where

ν_t is the pitch line velocity of the gear set, in m/s;

ν_{40} is the kinematic viscosity of the lubricant at 40 °C, in mm²/s.

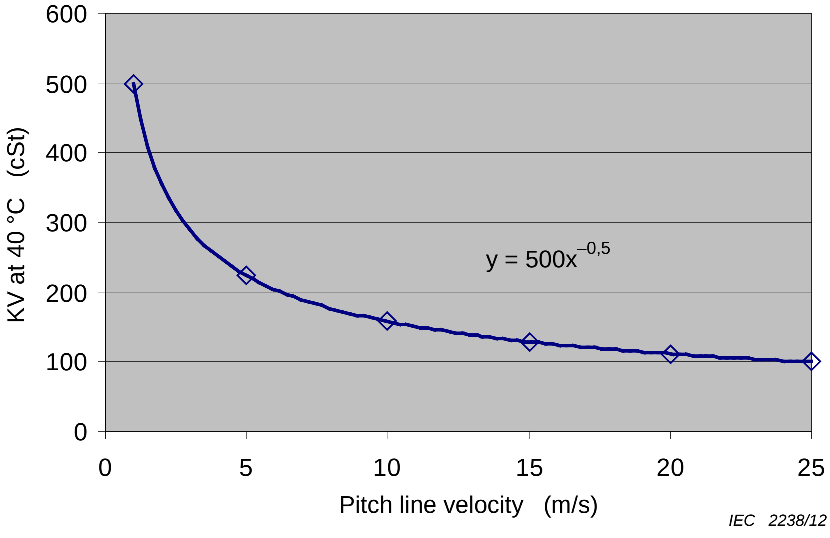


Figure E.1 – Viscosity requirements versus pitch line velocity

E.2.2 Low temperature characteristics

Just as the viscosity of the fluid at operating temperature is important to the film formed in the gear and bearing contact, consideration should also be given to the start-up conditions the turbine and transmission may encounter. It is important to ensure that the lubricant is capable of flowing sufficiently to all gear and bearing contacts at the lowest start-up condition, otherwise starvation could lead to premature damage. There are no published low temperature requirements for ISO viscosity grades as there are with many automotive applications. However, a reasonable estimate is to have a fluid with a maximum dynamic viscosity of 20 000 mPa·s (cP) at the start-up condition. This value can be obtained by several methods, the most common being the Brookfield viscometer method (ASTM D2983).

Table E.1 – Viscosity grade at operating temperature for oils with *VI* = 90

Temp. °C	Pitch line velocity, m/s[c]							
	1,0 – 2,5	**2,5**	**5,0**	**10,0**	**15,0**	**20,0**	**25,0**	**30,0**
10	32							
15	46	32						
20	68	46	32					
25	68	46	32					
30	100	68	46	32				
35	100	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	32
50	320	220	150	100	46	46	46	32
55	460	220	150	100	68	68	68	46
60	460	320	220	150	68	68	68	46
65	680	460	320	220	150	100	100	68
70	1 000	680	320	220	150	100	100	68
75	1 500	680	460	320	220	150	150	100
80	2 200	1 000	680	460	220	220	220	150
85	3 200	1 500	1 000	460	320	220	220	150
90	3 200	2 200	1 000	680	460	320	320	220
95		3 200	1 500	1 000	460	460	320	220
100		3 200	2 200	1 000	680	460	460	320

a
- (hatched) Consult gearbox, bearing and lubricant manufacturers if a viscosity grade of less than 32 or greater than 3 200 is indicated.
- (grey) Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
- (white) Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
- (double-ruled) Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

b This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

c Determine pitch line velocity of all gear sets. Select viscosity grade for critical gear set taking into account cold startup conditions.

Table E.2 – Viscosity grade at operating temperature for oils with *VI* = 120

Temp. °C	Pitch line velocity, m/s[c]							
	1,0 – 2,5	**2,5**	**5,0**	**10,0**	**15,0**	**20,0**	**25,0**	**30,0**
10	32							
15	46	32						
20	68	46	32					
25	68	46	32	32				
30	100	68	46	32				
35	150	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	32
50	320	220	100	100	68	46	46	46
55	320	220	150	100	68	68	46	46
60	460	320	220	150	68	68	68	46
65	680	460	320	150	100	100	100	68
70	1 000	460	320	220	150	150	100	68
75	1 000	680	460	220	150	150	150	100
80	1 500	1 000	460	320	220	220	150	100
85	2 200	1 000	680	460	220	220	220	100
90	2 200	1 500	1 000	460	320	320	220	150
95	3 200	2 200	1 000	680	320	320	320	220
100		2 200	1 500	680	460	460	320	220

a Consult gearbox, bearing and lubricant manufacturers if a viscosity grade of less than 32 or greater than 3 200 is indicated.

Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.

Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.

Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

b This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

c Determine pitch line velocity of all gear sets. Select viscosity grade for critical gear set taking into account cold startup conditions.

Table E.3 – Viscosity grade at operating temperature for oils with *VI* = 160

Temp. °C	Pitch line velocity, m/s[c]							
	1,0 – 2,5	2,5	5,0	10,0	15,0	20,0	25,0	30,0
10	32	32						
15	46	32	32					
20	68	46	32					
25	68	46	32	32				
30	100	68	46	32				
35	150	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	
50	220	150	100	68	46	46	46	32
55	320	220	150	100	68	68	46	32
60	460	220	150	100	68	68	68	46
65	460	320	220	150	100	100	68	46
70	680	460	220	150	100	100	100	68
75	680	460	320	220	150	150	100	68
80	1 000	680	320	220	150	150	150	100
85	1 500	680	460	320	220	220	150	100
90	1 500	1 000	680	320	220	220	220	150
95	2 200	1 500	680	460	320	220	220	150
100	3 200	1 500	1 000	460	320	320	220	150

a (hatched) Consult gearbox, bearing and lubricant manufacturers if a viscosity grade of less than 32 or greater than 3 200 is indicated.

(grey) Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.

(white) Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.

(double-ruled) Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

b This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

c Determine pitch line velocity of all gear sets. Select viscosity grade for critical gear set taking into account cold startup conditions.

Table E.4 – Viscosity grade at operating temperature for oils with *VI* = 240

Temp. °C	Pitch line velocity, m/s[c]							
	1,0 – 2,5	**2,5**	**5,0**	**10,0**	**15,0**	**20,0**	**25,0**	**30,0**
10	46	46						
15	68	46	32					
20	68	68	32	32				
25	100	68	32	32				
30	100	68	32	32	32			
35	150	68	68	46	32	32		
40	150	100	68	46	32	32	32	
45	220	100	100	68	46	32	32	
50	220	100	100	68	46	46	46	32
55	320	150	150	68	68	46	46	32
60	320	150	150	100	68	68	46	46
65	460	220	150	100	100	68	68	46
70	460	320	220	150	100	68	68	46
75	680	320	220	150	100	100	68	68
80	680	460	220	150	100	100	100	68
85	1 000	460	320	220	150	100	100	68
90	1 000	680	320	220	150	150	100	100
95	1 000	680	460	320	150	150	150	100
100	1 500	1 000	460	320	220	150	150	100

a
- Consult gearbox, bearing and lubricant manufacturers if a viscosity grade of less than 32 or greater than 3 200 is indicated.
- Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
- Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
- Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

b This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

c Determine pitch line velocity of all gear sets. Select viscosity grade for critical gear set taking into account cold startup conditions.

E.3 Lubricant performance characteristics

E.3.1 Lubricant assessment

While it is most desirable to have actual service experience with fluids this is not always practical or cost effective, especially with new fluids. Therefore, one must rely on laboratory test methods to assess the potential performance characteristics of a given fluid. The wind turbine application can be very demanding and, as such, many different performance areas must be addressed. As a minimum the following areas should be included in any assessment of a fluid:

- gear wear:

  - abrasive;
  - adhesive;
  - fatigue;
- bearing life;
- oxidation and thermal stability;
- corrosion protection:
  - ferrous;
  - non-ferrous;
- foaming and air entrainment;
- filterability;
- compatibility with other materials (paints, elastomers, filter media, etc.).

Many of the performance areas above can be addressed with standardized tests recognized in the lubricant industry. However, some of the more critical performance issues faced in the wind turbine application do not have standardized methods available at this time. Specifically, micropitting and bearing life are areas of concern in large turbines today, but the only methods available to the industry have not been standardized at this time. There also still remains questions about the relevance of some of the methods to actual practice which can only be resolved with further investigation and field evaluations. Table E.5 provides a list of standardized test methods to help define lubricant performance in some of the key areas mentioned above. The minimum requirements for some of these have not been established and should be negotiated between the lubricant supplier and end user/wind turbine manufacturer or component supplier.

Table E.5 – Standardized test methods for evaluating WT lubricants (fresh oil)

Property	Procedure name	Test method	Test conditions	Recommended minimum requirement
Gear wear (adhesive)	FZG scuffing test	ISO 14635-1	A/8.3/90	≥ Fail LS 12
Bearing wear	FE-8 stage 1	DIN 51819	7,5 r/min; 100 kN; 80 h; 80 °C	Rollers ≤30 mg Rippling: small on either roller or washer Micropitting: small
Bearing fatigue at mixed friction condition	FE-8 stage 2	DIN 51819	75 r/min; 100 kN; 800 h: 70 °C	Running time ≥ 800 h Roller wear ≤30 mg
Oxidative stability		ASTM D2893	312 h; 121 °C	≤6 % increase in KV100
Corrosion (ferrous)		ISO 7120 (ASTM D665) A. distd water B. syn sea water	24 h; 60 °C	Pass (no corrosion) A. for on-shore application B. for off-shore application
Corrosion (ferrous)	SKF Emcor	ISO 11007		2 rating max
Corrosion (non-ferrous)		ASTM D130 ISO 2160	3 h; 100 °C	1 rating max
Shear stability	KRL shear Test	CEC L-45-A99	20 h, 60 °C,	Stay in ISO VG class
Elastomer compatibility		ISO 13226	72NBR902 (100 °C) 75FKM585 (130 °C)	Vol -2 % to +5 % Hardness ±5 % Elongation <50 % change Tensile str <60 % change

Table E.6 represents a series of methods commonly used to help define the potential performance of a lubricant for wind turbine applications. Some of these methods are being reviewed for standardization at this time.

Table E.6 – Non-standardized test methods for lubricant performance (fresh oil)

Property	Procedure name	Test method	Test conditions	Recommended minimum requirement
Gear wear (fatigue)	FZG micropitting test	FVA 54/I-IV	CGF/8.3/60	≥LS 10
Gear wear (fatigue)	FZG micropitting test	FVA 54/I-IV	CGF/8.3/90	≥LS 10
Bearing fatigue at EHL condition	L11 stage 3	FAG Test	Brg 6206; 9 000 r/min; 8,5 kN; 700 h; no temp control	L50 ≥550h
Bearing fatigue behaviour and residue with added water	FE8 w/pre heating system – stage 4	FAG Test	81212MPB 750 r/min: 60 kN preheat system/water/control temp; 100 °C	Run time >600 h Filter blocking <2 Roller wear <30 mg Fatigue damage: no Residue at brg: mod/hvy Residue at preheat system: mod/hvy
Foaming	Flender foam test	ISO 12152 [a]	20 °C; 60 °C	≤17, 10 % respectively
Filterability	Hydac method single pass	ISO 13357-2 (modified)	Filter patch 5 µm and application filter media	Filterability index: ≥80 % for stage 1; ≥60 % for stage 2. Mean flow rate must be acceptable
Other material compatibility	Consult gearbox manufacturer			

[a] This standard is under development.

E.3.2 Scuffing test

The minimum requirement regarding scuffing load carrying capacity should be agreed upon between the gear manufacturer and lubricant supplier to satisfy design needs. In the absence of this interaction then a failure load stage minimum of 12 in the standard FZG scuffing test A/8,3/90 according to ISO 14635-1 (or equivalent) should be used. For the large gear drives used in power generator systems of large wind turbines, scuffing should not be a major problem and can be controlled by accurate gear design. There are many gearboxes in service, which do not require more than the minimum requirements from the DIN and ISO gear oil specifications. If the requirement regarding scuffing load carrying capacity (EP property of the lubricant) is too high it could have a negative influence on micropitting, pitting, low speed wear, bearing life, yellow metal compatibility, foaming and filterability properties of the oil.

E.3.3 Micropitting resistance

The micropitting load carrying capacity of gear oils in wind turbine gearboxes is a very important requirement. Much work has been done to develop methods and evaluate performance of fluids. Unfortunately, to date, there is no standardized method available to the industry. The FVA 54/I-IV represents the most widely used method, but there still remain some questions about the relevance of this method to actual service.

E.3.4 Bearing life/FAG FE8 bearing wear test

At present there are no standardized methods available to the industry that measure the effect of lubricant on bearing life. There are many factors that influence bearing life as highlighted in 7.3 of this standard. The choice of viscosity grade, base stock, additive and contaminants will have an influence on the fatigue life in service. There is no one test at the moment that can reliably predict the overall lubricant effect. At this time the multistage methods offered by bearing manufacturer FAG of Schweinfurt, Germany provide some guidance in this area. The Stage 1 FE-8 test is a fundamental requirement in DIN 51517-3.

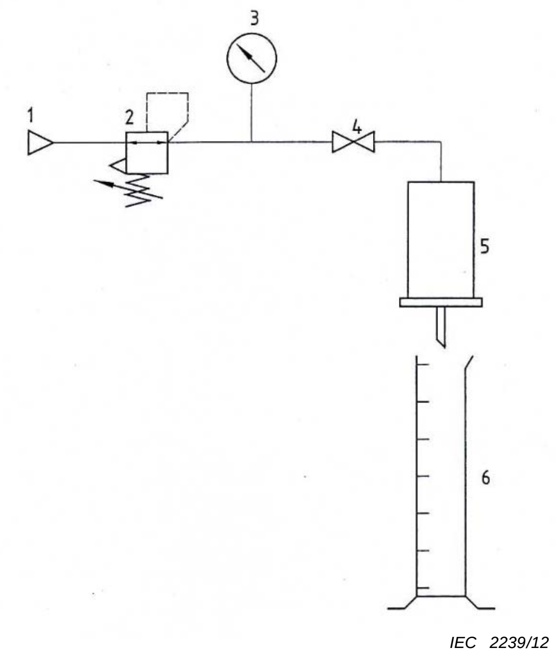
E.3.5 Filterability

E.3.5.1 Evaluating filterability

There are presently no standardized test procedures available for the filterability of gear oils (e.g. VG 320) so in-house tests have to be applied instead. The test procedures should be clarified between oil and filter supplier. An example for a filterability test procedure is described below. Because of the sensitivity of the behaviour of the oil with its additive packages in a filter system, it is recommended to test the performance in an additional accelerated multi-pass filterability test. In this test, the oil passes through the test filter multiple times in order to assess the compatibility of the filter media to the oil and its additive system. These tests should take into account that the filterability of the oil may decrease due to the oil aging process. Additionally, a filterability test with used oil, preferably from the field, should be considered.

E.3.5.2 Single-pass filterability test

This test serves to evaluate the general ability of the oil to pass through the filter media in a reasonable time and also to evaluate premature filter blockage caused by components of the oil. The test rig is similar to the apparatus described in ISO 13357-2 (see Figure E.2 below).



Key

1 source of compressed air or nitrogen
2 pressure regulator
3 pressure gauge
4 ball valve
5 pressure vessel with membrane support
6 measuring cylinder

Figure E.2 – Test apparatus for filterability evaluation

The main difference to ISO 13357-2 is the higher viscosity of gear oils e.g. VG320 compared to the scope of the ISO 13357-2 which is oil VG100 or lower viscosity grades, especially hydraulic oils. For this reason, the test procedures, parameters and interpretation have to be adapted for gear oils, e.g. the use of coarser filter media is recommended and should be specified in the in-house test procedure description.

Another method employed with high viscosity oils is the SKF in-house filterability test using a 12 μm cellulose nitrate membrane filter. This test should be carried out as part of the SKF approval process.

E.3.6 Foaming

For gear oils in wind turbine gearboxes the Flender foam test is well established, but it is not a standardized method. Although the ASTM D892 test is a standardized method there is some question regarding its relevance to wind turbine service. Therefore, it is not included here. A suggestion has also been made to combine the filterability and foaming evaluations into a single common method. This requires further investigation to validate its merits and use.

E.3.7 Corrosion protection

The SKF Emcor test is a typical requirement for gear oils in wind turbine gearboxes. For offshore applications this test and the ASTM D665 method, Part B using synthetic sea water should provide the user with an indicator of the corrosion protection afforded by the lubricant.

E.3.8 Oxidative stability

While wind turbines create a stressful climate for the lubricant, the thermal stress may not be severe with typical bulk lubricant operating temperatures in the 40 °C to 80 °C range. However, the ability of the oils to resist thermal stress and thermal cracking remain an important characteristic. Proprietary test methods for determining thermal stress resistance have been developed and used by lubricant formulators, but recognized international or national standard methods are not available at this time. The ASTM D2893 method provides a measure of the oxidative resistance of the oil. This method also measures the ability of the oil to prevent deposit and sludge build up over longer periods of time as might be expected in wind turbine service.

E.3.9 Shear stability

As lubricants change to meet the widening requirements of the application there is a growing need to monitor the potential for viscosity loss due to breakdown of polymeric components used in the formulations. The CEC L-45-A99 method has served the automotive sector well in predicting the losses that may occur due to mechanical shearing of fluid components. It is recommended that the lubricant maintains its viscosity within the original ISO VG class as a guideline for fluids used in wind turbine applications.

E.3.10 Compatibility

Due to the use of many different materials in the gearbox it is highly recommended to test the compatibility of those components with the oil. This should include static seals such as o-rings, dynamic seals, paints, and electronic components immersed in or in contact with the oil. The test procedure and limits of acceptance should be decided between the component and oil suppliers.

E.4 Recommendations for lubricant life and condition monitoring

E.4.1 Lubricant condition parameters

Lubricants begin to change immediately after being placed in service. The critical question becomes when should the lubricant be changed out and what parameters should be monitored to prevent costly repairs and downtime. This very much depends on the type of service, operating environment, type of lubricant, etc. However, if one considers that the basic functions of the lubricant include removing heat and minimizing friction and wear, then a few common sense parameters should be monitored. Key among those parameters would be viscosity, additive and wear elements, contaminants (solid or liquid), and general particulate levels. When monitored on a regular (and reasonably frequent) basis these parameters can provide valuable information about the state of the turbine in service. Suggestions for action based on their values are shown in Table E.7.

Table E.7– Guidelines for lubricant parameter limits

Parameter	Method	Acceptable limit	Cautionary level	Alarm level
Viscosity, ν_{40}	ISO 3104 ASTM D445	Nominal $\nu_{40} \pm 5$ %	Nominal $\nu_{40} \pm 8$ %	Nominal $\nu_{40} > \pm$ 10 %
Additive Elements (except Si antifoam)	ASTM D5185	New ± 10 %	New ± 20 %	New ± 30 %
Wear Elements [a] Fe Cr Cu, Al [a]	ASTM D5185	 $<$50 ppm [b] TBD $<$20 ppm	 50 ppm to150 ppm TBD 20 ppm to 50 ppm	 $>$150 ppm TBD $>$50 ppm
Cleanliness	ISO 4406	-/16/13	-/17/14	-/18/15
Water content (except PAG oils)	ISO 12937 ASTM D6304	$<$300 ppm	300 ppm to 600 ppm	$>$600 ppm

[a] Limit values for wear elements should be agreed between the gearbox manufacturer and wind turbine manufacturer. The limit values should reflect materials used in the gearbox and in the lubrication system.

[b] Parts per million.

None of the limits above should be considered absolutes, but rather guidelines or indicators of other issues taking place in the system. For example, changing viscosity beyond the acceptable limits shown in Table E.7, either increasing or decreasing, suggests different potential problems. If the viscosity is increasing it is a sign of degradation of the oil via oxidation. If the viscosity is decreasing this could be the result of shear thinning of some component in the formulation or from some contaminant. Decreasing viscosity reduces the film thickness of the fluid in high stress contact zones which in turn leads to a reduction in component life. The additive and wear elements provide a sign of the overall lubricant health. The levels should be monitored and viewed over time to assess trends for rates of wear that may suggest an adjustment in other parameters, e.g., a turbine showing higher rates of wear than similar equipment may require a higher viscosity fluid due to more stressful operation or it may be a sign of some internal problem with that turbine. Contaminants such as water or solid particulate are indicators of seal or breather malfunction and should be addressed quickly to prevent rapid increases in wear and component breakdown.

E.4.2 Lubricant condition monitoring

As previously noted, monitoring the condition of the lubricant parameters over time can provide useful information on the health of gears and bearings, and detect contamination or other adverse changes of the lubricant. To do this small quantities of lubricant are sampled from the gearbox and analysed after regular operation intervals. The results of this analysis are compared to the property limits as described in this annex and in 7.6.

E.4.3 Lubricant sampling

Whenever samples are taken, it is important that the same procedures be followed so that consistent samples are obtained. Once the monitoring program has begun, do not change sampling procedures or sampling points. It is recommended that a log be kept of samples for traceability.

E.4.4 Sampling techniques

E.4.4.1 General

Always use clean, lubricant compatible plastic or glass sample bottles and caps, and keep all sampling equipment scrupulously clean.

Prior to sampling, fill out the label completely and attach it to the sample bottle. Be sure to record the sample point, the date, the gearbox, site and turbine identifiers and other pertinent information.

Thoroughly clean the area around the sampling port valve (see 7.6.11.5 regarding sampling ports) before sampling. A flexible tube (preferably carbon steel or stainless steel) of sufficient length to reach a sample bottle should be attached to the sampling valve outlet port.

A plastic slip on or metal threaded cap should be attached to the open end of the sampling tube when it is not in use.

E.4.4.2 Sampling from the gearbox

For monitoring gearbox health, the sample should be taken from the gearbox sampling port while the oil is still warm. Discard any oil in the sampling port that may have been stagnant. Do this by turning on the valve, capturing the oil to be discarded in a separate bottle, and without touching the valve, obtain the sample to be analyzed in a fresh sample bottle.

E.4.4.3 Sampling from oil drums

For monitoring the quality of fresh lubricant, the sample should be taken from the lubricant drum. The procedure described below assumes that the objective is to test the general condition of the oil in the drum. Therefore, the sample is a mixture of oil taken from the top, middle and near the bottom of the drum. If the objective is to test for water contamination, sludge or sedimentation, the sample should be taken from the lowest point of the drum.

- Use a manual suction pump to draw the oil sample into the sample bottle.
- Sample from the top, middle and near the bottom of the drum in order to obtain a representative sample and avoid stratification. Use a sampling rod and attach the sampling tube to the rod with plastic ties. Position the sampling tube a few inches from the bottom of the rod to prevent the end of the tube from touching the sides and bottom of the drum. Take 1/3 of the sample with the rod touching the bottom of the drum. Take another 1/3 of the sample with the rod raised to mid-height of the oil level. Take the final 1/3 of the sample with the rod raised so that the sampling tube is just below the surface of the oil.
- Discard the sampling tube to avoid contaminating subsequent oil samples.

E.5 Maintaining lubricant cleanliness

E.5.1 Overview

Properly designed, installed and maintained mechanical filter systems have been found to adequately maintain wind turbine gearbox lubricant cleanliness levels as specified in 7.6.9. Depending on the operating environment, semi-annual or longer servicing intervals can be achieved.

E.5.2 Configuration of filtration systems

Wind turbine gearbox mechanical filtration systems can utilize one or more mechanical filter assemblies. Frequently, the filtration system is combined with the cooling system, see Figure E.3. Generally, the filtration system includes:

- an inline filter assembly adequately sized to minimally restrict the flow of oils up to ISO viscosity grade 460, VG 460, at the volumes required to lubricate the gears, bearings and other components;
- an offline filter assembly (recommended);
- a pressure strainer (recommended) in the oil distribution system to capture large particulate matter in the event of inline filter element failure, cold start conditions and clogged inline filter.

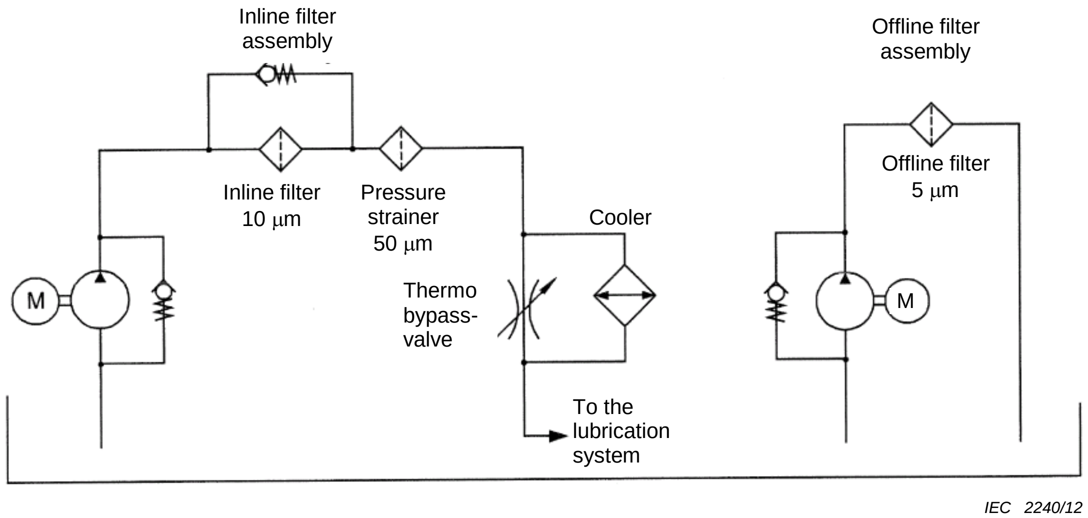


Figure E.3 – Example for circuit design of combined filtration and cooling system

E.5.3 Inline filter

The inline filter assembly should be sized to:

- provide the minimum restricted flow of the required volume and pressure of oil to the lubricated components during turbine operation;
- prohibit flow restriction on cold start-up that may damage or collapse filter element; and
- provide dirt retention capacity capable of achieving seven months of service when semi-annual filter element change cycles are desired, and fourteen months of service when annual filter element change cycles are required.

When choosing the filter micron rating, one should know the cleanliness class requirements (see Table 11) and the system properties such as dirt ingression rate, flow rate, and oil properties. A typical value is 10 µm with a beta ratio greater than or equal to 200 ($\beta_{10(c)} \geq 200$). The index (c) refers to the certified micron rating per ISO 16889. The filter assembly should include an automatic oil bypass to permit adequate oil flow to protect the lubricated components during cold start-up and when the filter media becomes clogged with contaminants. Sensors should be installed to monitor the clogging status and oil flow of the filter assembly. This may include:

- pressure drop across the filter element;
- the status of the bypass device; and
- positive oil flow or pressure upstream and downstream of the filter element.

E.5.4 Offline filter

The filter element should be sized to:

- prohibit flow restriction during initial operation that is sufficient to damage or collapse the filter element, and
- have a contaminant retention capacity capable of providing seven months of service when semi–annual filter element change cycles are desired, and fourteen months of service when annual filter element change cycles are required.

The selected filter micron rating should be appropriate to achieve the cleanliness class requirements and the system properties (for example, dirt ingression rate, flow rate, oil properties). A typical value for the micron rating is 5 µm with a beta ratio greater than or equal to 200 ($\beta_{5(c)} \geq 200$). The filter assembly should include an automatic oil bypass. Sensors should be installed to monitor the clogging status of the filter assembly. This may include:

- pressure loss across the filter element; and
- the status of the bypass device.

Optionally, offline filtration may be utilized to absorb free water in the oil.

E.5.5 Pressure strainer

A metal mesh strainer assembly may be installed in the pressurized oil distribution line downstream of the inline filter assembly. The function of this device is to prevent large particles from entering the oil distribution system when the inline filter bypass is open at cold start conditions and in event of an inline filter element mechanical failure, such as collapse. A stainless steel wire mesh element with openings of 50 μm is generally acceptable.

The strainer housing and element should be dimensionally correct to minimize the restriction of oil flow during cold start-up and normal gearbox operation. It is important to prevent damage to the lubrication system due to insufficient flow past a clogged pressure strainer. Suitable measures to avoid such a situation include, but are not limited to, adding a bypass valve around the strainer or adding a pressure transducer or flow switch to indicate clogging has occurred.

E.5.6 Filter and gear oil compatibility

The filter assemblies shall be sized appropriately and manufactured from materials that are chemically compatible with the lubricant. They should not have a detrimental effect on the long term performance of the lubricant or lubrication system. This includes:

- hardening, softening or degradation of the filter media or sealing components;
- filterability;
- removal or alteration of additives such as antifoam agents; and
- contributing to electrophoresis (static electricity build-up within the lubrication system).

E.5.7 Coolers and heaters

Coolers and heaters are used to maintain a proper oil temperature for start-up and operation of the gearbox. These devices may be installed in the main lubrication system, offline or a combination of both. Devices may be added to limit the pressure drop across coolers and heaters.

The components, if incorporated into the main lubrication system, must be sized appropriately to provide sufficient oil flow to the gears and bearings. All materials of these devices shall be compatible with the lubricant.

Annex F
(informative)

Design verification documentation

A listing of recommendations for the documentation of the gearbox design and its verification is shown in Table F.1. In the table is a description of the various documents, who issues the document and who receives it. Details regarding the necessary calculations and/or calculation methods are to be taken from the relevant clauses.

Table F.1 – Design validation and verification documentation

Document	Specification	Issued by	Recipient	Remark
General specification	Operating conditions as defined in 6.4	W	G, (B, L), C	
	Main technical data such as nominal speed and speed range, ratio, nominal torque, etc.		G, (B, L), C	
	Type and arrangement of brakes, couplings and gear support/mounting		G, (B), C	
	Schematic description of the system for cooling, lubrication and filtration		G, (B, L), C	
	Monitoring including list of sensors		G, (B, L), C	
	Supplementary equipment		G, (B, L), C	
	Requirements for tests and calculations		G, (B, L), C	
	Requirements for documentation		G, (B, L), C	
	Requirements for quality control		G, (B, L), C	
Load specification	Short description of the turbine including control and safety system	W	G, (B), C	
	Time series for fatigue and static strength calculations			
	Fatigue loads prepared as LDD's and RFC's for all relevant loads			
	Extreme loads including reverse torque			
	Additional loads if any created e.g. by vibration or deformation			
Test specification	Requirements regarding the test of the gearbox on the test rig: design of the test rig, measurement equipment, course of the test run, criteria of success	G, (B, C)	W, C	
	Requirements regarding the test of the gearbox on the turbine: site, measurement equipment and data to be measured, course of the test run, criteria of success	W	G (B, L), C	
	Requirements regarding the test of serial gearboxes on the test rig: design of test rig, measurement equipment, course of the test run, criteria of success	G, (B, L)	W, C	

Document	Specification	Issued by	Recipient	Remark
Drawings and bill of material	Outline dimension and assembly drawing	G	W, B, C	
	Drawings and specifications of toothed parts including material and heat treatment, tool data and profile- and lead modifications	G	W, C	
	Drawings and specifications for shafts and couplings/connections	G	W, B, C	
	Drawing and specification of housing parts including bearing seats, material and other relevant data	G	W, B, C	
	Schematic drawing of the lubrication system with information regarding pipe diameters and nozzles	G	W, B, L, C	
	Drawings of bearings including all information pertinent to calculations specified in 7.3.8	B	W, G, C	
	Bill of material of gearbox	G	W, G, C	
	Bill of material of cooling- and filtration system	W, G	W, G, L, C	
Gear analyses	Fatigue analysis against pitting and tooth root fracture according to the ISO 6336 series	G	W, C	
	Static strength analysis against pitting and tooth root fracture according to ISO 6336 series		W, C	
	Scuffing analysis		W, C	
	Calculation of face load factor, $K_{H\beta}$		W, C	
	Gear mesh frequencies for all gear meshes considering the specified speed range		W, C	
Bearing analyses	Assumptions and requirements for the bearing calculations	B	W, G, L, C	
	Calculation of rating life and static rating for all bearings according to 7.3.8		W, G, C	
	Calculation of axial load capacity		W, G, C	
	Evaluation of other possible failure modes e.g. skidding and slip		W, G, C	
	Inner and outer race frequencies for all bearings as well as cage rotating frequencies considering the specified speed range		W, G, C	
Other components	Fatigue and static strength analysis of structural components as defined in 7.4,fatigue and static strength analysis of shafts, shaft-hub connections and structurally loaded bolts	G	W, C	
	Fatigue and static strength analysis of torque arm, planet carrier and housing as FEA-report according to 7.5		W, C	
	Deformation of load carrying parts which may lead to significant changes of pressure distribution in gear meshes and/or bearings		W, B, C	
	Lubricant data sheet	L	W, G, B, C	
Verification of thermal capacity and available cooling capacity	Calculation with nominal power under consideration of the maximum permissible ambient temperature (according to ISO/TR 14179)	G	W, C	

Document	Specification	Issued by	Recipient	Remark
O&M manual	Required maintenance and inspection intervals	G	W, C	
	Admissible temperatures, pressures, etc. and necessary surveillance of the gearbox		W, B, C	
	Instructions regarding the start-up and run-in procedures		W, B, C	
	Change interval and recommended analysis limits for lubricants		W, B, L, C	
Prototype test on test rig	Assembly protocols of the prototype gearbox	G	W, B, L, C	
	Description of the test rig incl. pictures		W, B, L, C	
	Detailed test program		W, B, L, C	
	Calibration method and records		C	
	Gear contact pattern records e.g. pictures		W, C	
	Bearing contact pattern records		W, B, C	
	Measured data (temperatures, pressures, oil cleanliness, etc.)		W, B, L, C	
	Conclusions		W, B, L, C	
Prototype test in wind turbine	Assembly protocols of the prototype gearbox	G	W, B, L, C	
	Description of site and environmental conditions (e.g. wind speed and temperature during the test)	W (G)	W, B, L, C	
	Detailed test program describing the measurements and tests to be carried out	W, G	W, B, L, C	
	Calibration method and records	W, G	C	
	Description of the test turbine and operation during the test (e.g. availability, produced electricity, power curve)	W	W, B, L, C	
	Inspection protocols (incl. pictures of contact patterns) and oil analyses	G, (W)	W, B, L, C	
	Test results such as time series of measured data e.g. power, torque, temperatures, pressures, oil cleanliness	W, G	W, B, L, C	
	Conclusions	G, (W)	W, B, L, C	

W Wind turbine manufacturer

G Gearbox manufacturer

B Bearing manufacturer

L Lubricant supplier

C Certification body

() Optional

Annex G
(informative)

Bearing calculation documentation

G.1 Proposed documentation of bearing calculations

Maintaining a record of the assumptions and calculations of bearing loads, calculated fatigue life and response is rather critical to the gearbox design. The following is a proposed format for documenting these details. Note that brackets, [ ], are used to insert comments or actual configuration data.

G.2 Title page

Bearing calculations for:	[Name of gear unit manufacturer]
Gear unit:	[Gear unit designation]
Wind Turbine:	[Name of turbine manufacturer]
[Name of bearing supplier] reference:	[File reference number]
Supplied load spectrum:	[reference number]
Dated:	[date]
Prepared by:	[Name]
Approved by:	[Name]

Distribution list – Version

Company	Name
[Name of gear unit manufacturer]	[Name]
[Name of turbine manufacturer]	[Name]
[Name of bearing manufacturer]	[Name]

Revision	Date	Change
1.0	[Date]	Issue of the document

G.3 Drivetrain details

See attached drawing or sketch [should be provided for clear identification of the drivetrain including the coordinate system] and details of the wind turbine and drivetrain [include size, type, power, classification, etc.].

The calculations were carried out for a reduced load spectrum derived from the load spectrum specified in document.

The running conditions and the gear unit data are listed below and represent the base for the calculations.

G.4 Input data

G.4.1 Bearing position

No: [a]	Shaft position	Shaft label	Bearing label	Bearing designation
1	Low speed planet carrier	LS-Car	RS	
2			GS	
3	Low speed planet	LS-Plan	RS	
4			GS	
5	Intermediate speed planet carrier [b]	IS-Car	RS	
6			GS	
7	Intermediate speed planet [b]	IS-Plan	RS	
8			GS	
9	Low speed intermediate shaft [b]	LSIS	RS	
10			GS-in	
11			GS-out [b]	
12	High speed intermediate shaft	HSIS	RS	
13			GS-in	
14			GS-out [b]	
15	High speed shaft	HSS	RS	
16			GS-in	
17			GS-out	

[a] The numbering corresponds to the torque flow from input to output shaft looking from rotor towards generator.

[b] Optional, depending on gearbox design.

G.4.2 Bearing data

No: [a]	Bearing designation	D mm	D mm	$B/T/H$ mm	d_m mm	D_W mm	L_W mm	z	α_0 °	C_{dyn} kN	C_0 kN	C_u kN	n_{ref} rpm
1													
2													
3													
. . .													
17													

[a] The numbering corresponds to the torque flow from input to output shaft looking from rotor towards generator.

G.4.3 Gear data

G.4.3.1 General

Gear stage [c]	Designation	Centre distance[a] mm	Number of teeth					Module	Pressure angle	Helix angle
			z_{ring}	z_{planet}	z_{sun}	z_1	z_2	m_n mm	α_n °	β °
1	LS – planet	[a]							[20]	- [helix] [a]
2	IS – planet [d]	[a]							[20]	- [helix] [a]
3	cylindrical gears [d]	[a]							[20]	+ [helix] [b]
4	cylindrical gears [d]	[a]							[20]	- [helix] [b]

[a] Helix angle related to the sun wheel: "-" sign for Left and "+" sign for Right.

[b] Helix angle related to the pinion: "-" sign for Left and "+" sign for Right.

[c] Sequence of numeration according to torque flow.

[d] Optional, depending on gearbox design.

G.4.3.2 LS-planet

Load application point offset	Planet – sun gear mesh mm	Planet – ring wheel gear mesh mm
T_{eq}	Distance from middle towards [Rotor/Generator]	[Distance from middle towards [Rotor/Generator]
T_{max}	Distance from middle towards [Rotor/Generator]	[Distance from middle towards [Rotor/Generator]
T_{max_rev}	Distance from middle towards [Rotor/Generator]	Distance from middle towards

Or if only 1 value has been specified

$K_{H\beta}$ = [Total]: Planet-sun: (mm) shifted from the middle towards the [Rotor/Generator]
Planet-ring: (mm) shifted from the middle towards the [Rotor/Generator]

G.4.3.3 IS-planet

Load application point offset	Planet – sun gear mesh	Planet – ring wheel gear mesh
T_{eq}	[Distance from middle towards [Rotor/Generator]	Distance from middle towards [Rotor/Generator]
T_{max}	Distance from middle towards [Rotor/Generator]	[Distance from middle towards [Rotor/Generator]
T_{max_rev}	Distance from middle towards [Rotor/Generator]	Distance from middle towards [Rotor/Generator]

Or if only 1 value has been specified

$K_{H\beta}$ = [Total]: Planet-sun: (mm) shifted from the middle towards the [Rotor/Generator]
Planet-ring: (mm) shifted from the middle towards the [Rotor/Generator]

G.4.4 Power, torque, force and speed data

Nominal wind turbine ratings:

Power: $P_{nominal}$ = [power] MW

Torque: $T_{nominal}$ = [torque] kNm

Speed: $n_{nominal}$ = [speed] r/min

Input speed: n_1 = [input speed] r/min

Output speed: n_2 = [output speed] r/min.

Total gear ratio: [n_1 / n_2]

Load spectrum according to document [document name] of [date] and reduced by method per (for suggested methods, see Annex C):

Load case	**Torque, T_i** kNm	**Speed n_i** r/min	**[Time or revolutions]**	**[Time or revolutions] share** %
T max				
T max_rev				
T nom				
1				
2				
3				
4				
5				
6				
7				
8				
9				
10				
$T_{eq,}$ (exponent = 3.3)				

G.4.5 Lubrication and cleanliness data

Type of oil used is [mineral or synthetic], ISO VG [viscosity] gear oil [with/without EP additives].

No: [a]	Shaft position	Bearing designation	Bearing lubricant operating temperature °C [c]	Kappa, k	Lubrication cleanliness	Cleanliness used in calculation
1	Low speed planet carrier					
2						
3	Low speed planet					
4						
5	Intermediate speed planet carrier [b]					
6						
7	Intermediate speed planet wheel [b]					
8						
9	Low speed intermediate shaft [b]					
10						
11						
12	High speed intermediate shaft					
13						
14						
15	High speed shaft					
16						
17						

[a] The numbering corresponds to the torque flow from input to output shaft looking from rotor towards generator.

[b] Optional, depending on gearbox design.

[c] See 7.3.6.2.

G.4.6 Fits and estimated clearances in operation

No:[a]	Shaft position	Bearing designation	Fits µm		Material		Operating temperature °C [c]		Internal clearance µm	
			Shaft	Housing	Shaft	Housing	Shaft	Housing	Before mounting	In operation
1	Low speed planet carrier									
2										
3	Low speed planet wheel									
4										
5	Intermediate speed planet carrier [b]									
6										
7	Intermediate speed planet wheel [b]									
8										
9	Low speed intermediate shaft									
10										
11										
12	High speed intermediate shaft									
13										
14										
15	High speed shaft									
16										
17										

[a] The numbering corresponds to the torque flow from input to output shaft looking from rotor towards generator.

[b] Optional, depending on gearbox design.

[c] See 7.3.6.2.

G.5 Calculation results

G.5.1 Basic rating life of bearings

Basic rating life of bearings for equivalent load and speed including the load application point offset on the planets:

Equivalent torque, T_{eq} = [Torque] kNm

Equivalent speed, n_{eq} = [speed] r/min.

No: [a]	Shaft position	Bearing designation	Basic rating life, L_{h10} h	Modified reference rating life, L_{hmr} h	Basic reference rating life, L_{hr} h	L_{hmr} / L_{hr}	$\sigma_{H,max}$ N/mm^2
1	Low speed planet carrier						
2							
3	Low speed planet						
4							
5	Intermediate speed planet carrier [b]						
6							
7	Intermediate speed planet [b]						
8							
9	Low speed intermediate shaft						
10							
11							
12	High speed intermediate shaft						
13							
14							
15	High speed shaft						
16							
17							

[a] The numbering corresponds to the torque flow from input to output shaft looking from rotor towards generator.

[b] Optional, depending on gearbox design.

G.5.2 Bearing static rating

Static rating of bearings including the load application point offset on the planets for:

Maximum torque, T_{max} = [Torque] kNm
Maximum rev torque, T_{max_rev} = [Torque] kNm

No: [a]	Shaft position	Bearing designation	Maximum torque		Maximum rev. torque	
			$\sigma_{H\ max}$ N/mm^2	Static safety	$\sigma_{H\ max}$ N/mm^2	Static safety
1	Low speed planet carrier					
2						
3	Low speed planet					
4						
5	Intermediate speed planet carrier [b]					
6						
7	Intermediate speed planet [b]					
8						
9	Low speed intermediate shaft					
10						
11						
12	High speed intermediate shaft					
13						
14						
15	High speed shaft					
16						
17						

[a] The numbering corresponds to the torque flow from input to output shaft looking from rotor towards generator.

[b] Optional, depending on gearbox design.

G.6 Conclusions

[evaluation of application]

[pertinent comments]

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ISO 8579-2, *Acceptance code for gears – Part 2: determination of mechanical vibrations of gear units during acceptance testing*

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VDI 3834-1, *Measurement And Evaluation Of The Mechanical Vibration Of Wind Energy Turbines And Their Components, Verein Deutscher Ingenieure, 2009, www.vdi.de*

The following standards may be found relevant for the design of a wind turbine gearbox. This list also includes alternative test methods.

ISO/IEC 17025, *General requirements for the competence of testing and calibration laboratories*

ISO 1122-1, *Vocabulary of Gear Terms – Part 1: Definitions Related to Geometry*

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ISO 5593, *Rolling bearings – Vocabulary*

ISO 7120, *Petroleum Products and Lubricants –Petroleum Oils and Other Fluids – Determination of Rust-Preventing Characteristics in the Presence of Water*

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ISO 12107, *Metallic materials – Fatigue testing – Statistical planning and analysis of data*

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