

INTERNATIONAL STANDARD

NORME INTERNATIONALE



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

**Éoliennes –
Partie 4: Exigences de conception des boîtes de vitesses des éoliennes**



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COMMISSION

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INTERNATIONAL ELECTROTECHNICAL COMMISSION

WIND TURBINES –

Part 4: Design requirements for wind turbine gearboxes

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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 bilingual version (2020-02) corresponds to the monolingual English version, published in 2012-12.

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.

The French version of this standard has not been voted upon.

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,
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- amended.

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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)

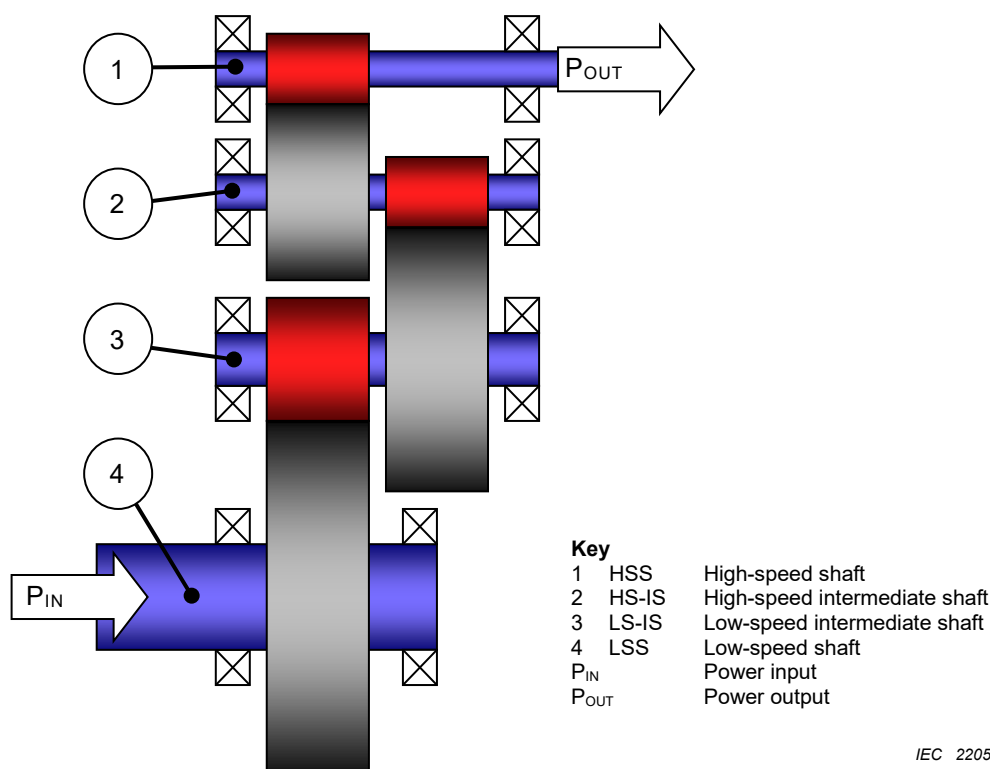
In case of paired bearings the following can be used:

- 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”.



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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.

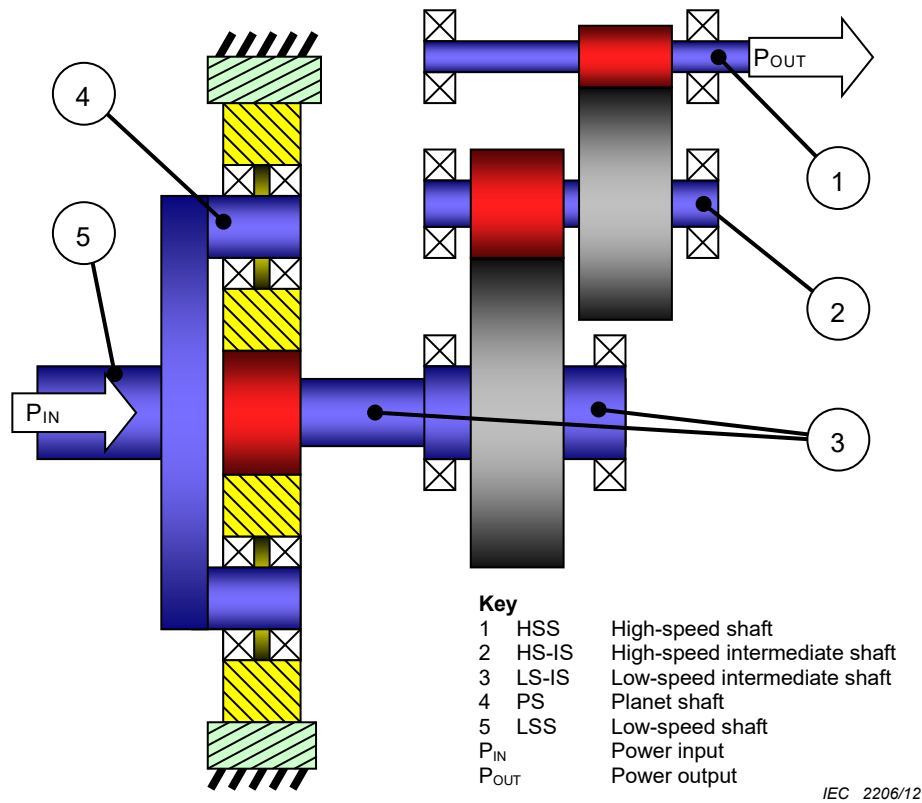


Figure 2 – Shaft designation in 3-stage gearboxes with one planet stage

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

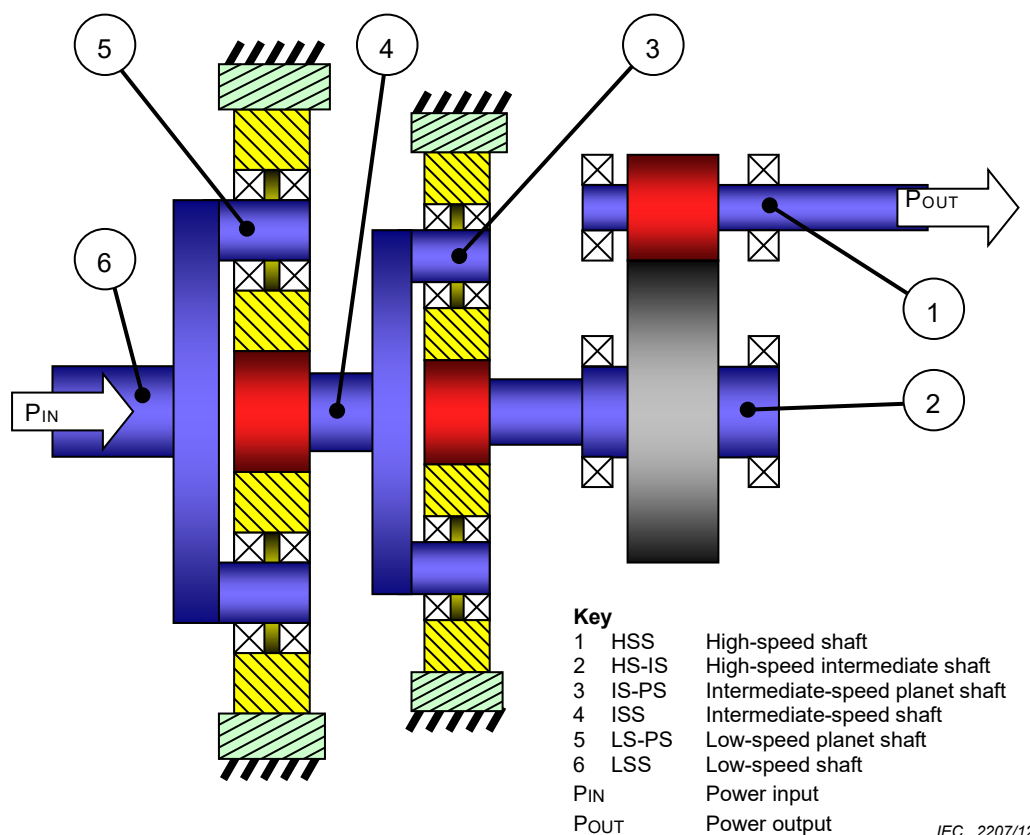


Figure 3 – Shaft designation in 3-stage gearboxes with two planet stages

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	–
$\eta_{pl,\sigma,GF}$	section factor for global failure local stress in relation to R_P	–
$\eta_{pl,\sigma,LF}$	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	$^\circ$
α_0	bearing nominal contact angle	$^\circ$
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	$^\circ\text{arc-min}$
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
$\sigma_{H \text{ lim}}$	allowable stress number (contact stress)	MPa
σ_I	maximum principal stress	MPa
σ_{III}	minimum principal stress	MPa
$\sigma_{ij, \text{pre}} (s)$	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
$\Sigma \rho_{\text{line}}$	curvature sum for line contact	–
$\Sigma \rho_{\text{point}}$	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.

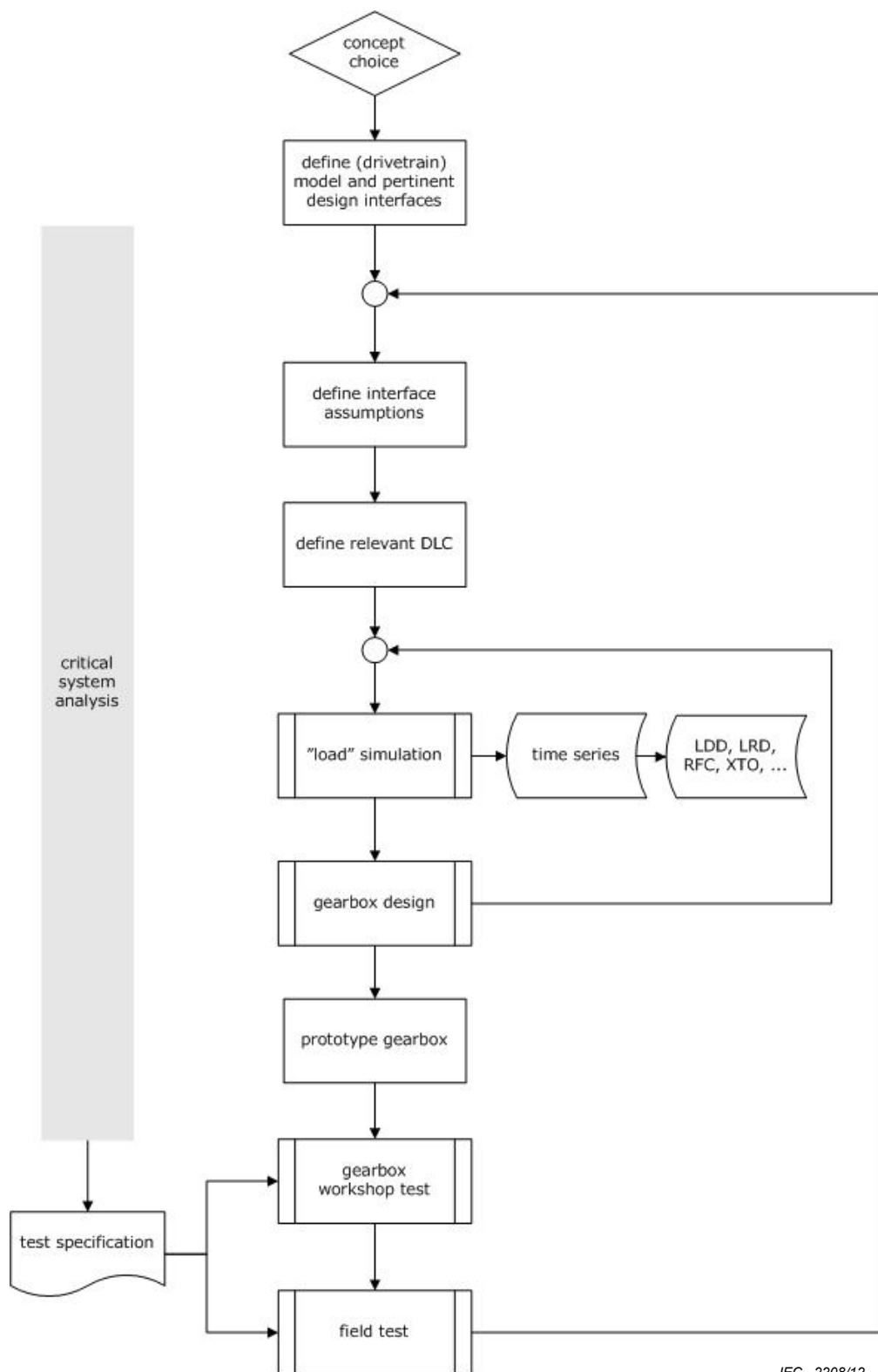
Scheduled maintenance and even scheduled replacement of components may be applied to increase the availability and reliability of the gearbox. Components with a design lifetime less than that specified for the entire gearbox shall be serviceable or replaceable. The expected design life of these components 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.



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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).

$$f_{ma} = \sqrt{f_{H\beta 1}^2 + f_{H\beta 2}^2 + f_{\Sigma\gamma}^2} \quad (1)$$

$$f_{ma} = f_{H\beta 1} + f_{H\beta 2} + f_{\Sigma\gamma} \quad (2)$$

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 $\sigma_{H\lim}$ 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 $0,25 \times m_n$. 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 $R_a = 0,8 \mu\text{m}$. Maximum flank surface roughness for internal gears shall be $R_a = 1,6 \mu\text{m}$. 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 $R_z \leq 4,8 \mu\text{m}$ for external gears and $R_z \leq 9,6 \mu\text{m}$ 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.

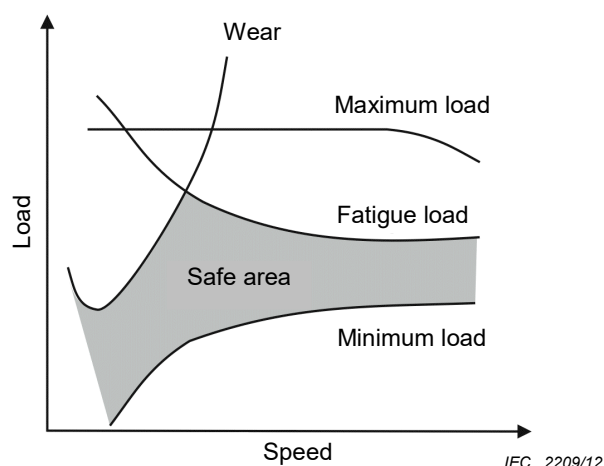


Figure 5 – Examples of bearing selection criteria

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.

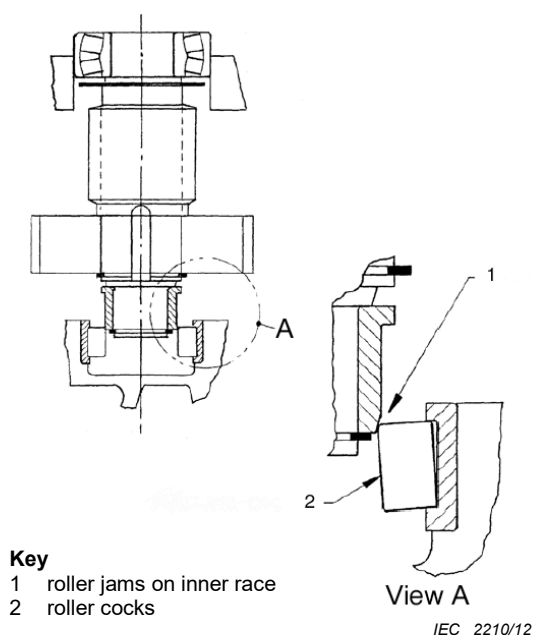


Figure 6 – Blind bearing assembly

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):

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

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;