BRITISH STANDARD

BS ISO 81400-4:2005

Wind turbines —

Part 4 : Design and specification of gearboxes

ICS 21.200; 27.180

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National foreword

This British Standard is the UK implementation of ISO 81400-4:2005, incorporating corrigendum December 2005.

NOTE ISO corrigendum December 2005 replaces the last two paragraphs of the ISO foreword.

The UK participation in its preparation was entrusted to Technical Committee MCE/5, Gears.

A list of organizations represented on this committee can be obtained on request to its secretary.

This publication does not purport to include all the necessary provisions of a contract. Users are responsible for its correct application.

Compliance with a British Standard cannot confer immunity from legal obligations.

This British Standard was published under the authority of the Standards Policy and Strategy Committee on 30 May 2008

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ISBN 978 0 580 54862 8

Amendments/corrigenda issued since publication

Date	Comments

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INTERNATIONAL STANDARD

ISO 81400-4

First edition
2005-10-01

Wind turbines —

Part 4:
Design and specification of gearboxes

Aérogénérateurs —

Partie 4: Conception et spécifications des boîtes de vitesses

Reference number
ISO 81400-4:2005(E)

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Full document details: BS ISO 81400-4:2005

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Full document details: BS ISO 81400-4:2005

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO 81400-4:2005 was prepared by AWEA and AGMA (as ANSI/AGMA/AWEA 6006-A03) and was adopted, under a special “fast-track procedure”, jointly by Technical Committee ISO/TC 60, *Gears*, and Technical Committee IEC/TC 88, *Wind turbines*, in parallel with its approval by the national bodies of ISO and IEC.

ISO 81400-4 is part of the IEC 61400 series. It is to be published as, and replaced by, IEC 61400-4 at the next revision.

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Introduction

The operation and loading of a wind turbine speed increasing gearbox is unlike most other gear applications. The intent of this standard is to describe the differences. Much of the information is based on field experience. This standard is a tool whereby wind turbine and gearbox manufacturers can communicate and understand each other's needs in developing a gearbox specification for wind turbine applications. The annexes present informative discussion of various issues specific to wind turbine applications and gear design.

A combined committee of the American Wind Energy Association (AWEA) and American Gear Manufacturers Association (AGMA) members representing international wind turbine manufacturers, operators, researchers, consultants; and gear, bearing, plus lubricant manufacturers were responsible for the drafting and development of this standard.

The committee first met in 1993 to develop AGMA/AWEA 921–A97, *Recommended Practices for Design and Specification of Gearboxes for Wind Turbine Generator Systems.* The AGMA Information Sheet was approved by the AGMA/AWEA Wind Turbine Gear Committee on October 25, 1996 and by the AGMA Technical Division Executive Committee on October 28, 1996. This standard superseded AGMA/AWEA 921–A97.

The first draft of ANSI/AGMA/AWEA 6006–A03 was made in March, 2000. It was approved by the AGMA membership in October, 2003. It was approved as an American National Standard (ANSI) on January 9, 2004.

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Wind turbines – Part 4: Design and specification of gearboxes

1 Scope

This standard applies to gearboxes for wind turbines with power capacities ranging from 40 kW to 2 MW. It applies to all parallel axis, one stage epicyclic, and combined one stage epicyclic and parallel shaft enclosed gearboxes. The provisions made in this standard are based on field experience with wind turbines having the above power capacities and configurations.

Guidelines of this standard may be applied to higher capacity wind turbines provided the specifications are appropriately modified to accommodate the characteristics of higher capacity wind turbines.

Life requirements apply to wind turbines with a minimum design lifetime of 20 years.

2 Normative references

The following standards contain provisions which, through reference in this text, constitute provisions of this standard. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this standard are encouraged to investigate the possibility of applying the most recent editions of the documents indicated below.

AGMA 901-A92, *A Rational Procedure for Preliminary Design of Minimum Volume Gears*

AGMA 913-A98, *Method for Specifying the Geometry of Spur and Helical Gears*

AGMA 925-A03, *Effect of Lubrication on Gear Surface Distress*

AMS 2301, *Aircraft quality steel cleanliness, magnetic particle inspection procedure*

ANSI Y12.3-1968, *Letter symbols for quantities used in mechanics of solids*

ANSI/AGMA 1012-F90, *Gear Nomenclature, Definitions of Terms with Symbols*

ANSI/AGMA 2101-D04, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*

ANSI/AGMA 6000-B96, *Specification for Measurement of Linear Vibration on Gear Units*

ANSI/AGMA 6001-D97, *Design and Selection of Components for Enclosed Gear Drives*

ANSI/AGMA 6025-D98, *Sound for Enclosed Helical, Herringbone, and Spiral Bevel Gear Drives*

ANSI/AGMA 6110-F97, *Standard for Spur, Helical, Herringbone and Bevel Enclosed Drives*

ANSI/AGMA 6123-A88, *Design Manual for Enclosed Epicyclic Metric Module Gear Drives*

ANSI/AGMA 9005-E02, *Industrial Gear Lubrication*

ASTM A534, *Standard specification for carburizing steels for anti-friction bearings*

Det Norske Veritas Classification AS, Classification Notes No. 41.2, *Calculation of Gear Rating for Marine Transmissions,* July 1993

DIN ISO 281 Bbl. 4:2003, *Dynamische Tragzahl und nominelle Lebensdauer – Verfahren zur Berechnung der modifizierten Referenzlebensdauer für allgemein belastete Wälzlager (Dynamic load ratings and life – Method for calculation of the modified reference rating life for generally loaded rolling bearings)*[1)]

DIN 743:2000, *Tragfähigkeitsberechnung von Wellen und Achsen (Calculation of load capacity of shafts and axles)*

DIN 6885-2:1967, *Drive Type Fastenings without Taper Action; Parallel Keys, Keyways*

DIN 7190:2001, *Interference fits – Calculation and design rules*

[1)] English translation available as ISO TC 4/SC 8 N254a

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ISO 76:1987, *Rolling bearings – Static load ratings*

ISO 281:1990, *Rolling bearings – Dynamic load rating and rating life*

ISO R773:1969, *Rectangular or square parallel keys and their corresponding keyways (dimensions in millimeters)*

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 4406:1999 (SAE J1165), *Hydraulic fluid power – Fluids – Method for coding the level of contamination by solid particles*

ISO 6336- 1: 1996, *Calculation of load capacity of spur and helical gears- Part 1: Basic principles, introduction and general influence factors*

ISO 6336- 2: 1996, *Calculation of load capacity of spur and helical gears- Part 2: Calculation of surface durability (pitting)*

ISO 6336- 3: 1996, *Calculation of load capacity of spur and helical gears- Part 3: Calculation of tooth bending strength*

ISO 6336-5: 1996, *Calculation of load capacity of spur and helical gears- Part 5: Strength and quality of materials*

ISO/DIS 6336-6[2)], *Calculation of load capacity of spur and helical gears – Part 6: Calculation of service life under variable load*

ISO 8579-1:2002, *Acceptance code for gears – Part 1: Determination of airborne sound power levels emitted by gear units*

ISO 8579-2:1993, *Acceptance code for gears – Part 2: Determination of mechanical vibration of gear units during acceptance testing*

ISO/TR 13593:1999, *Enclosed gear drives for industrial applications*

ISO/TR 13989-1:2000, *Calculation of scuffing load capacity of cylindrical, bevel and hypoid gears – Part 1: Flash temperature method*

ISO 14104:1995, *Gears – Surface temper etch inspection after grinding*

ISO/TR 14179-1:2001, *Gears – Thermal capacity – Part 1: Rating gear drives with thermal equilibrium at 95 °C sump temperature*

[2)] Presently at the development stage.

3 Definitions and symbols

3.0 Terms and definitions

For the purposes of this document, the terms and definitions given in 3.2 through 3.4 and the following apply, wherever applicable, conforming to ANSI/AGMA 1012-F90, and ANSI Y12.3-1968.

3.1 Symbols

The symbols, terms and units used in this standard are shown in table 1.

NOTE: The symbols and terms contained in this document may vary from those used in other AGMA standards. Users of this standard should assure themselves that they are using these symbols and terms in the manner indicated herein.

3.2 Wind turbine terms

active yaw: A system to rotate the nacelle relative to the changing direction of the wind. See passive yaw.

airfoil: Two dimensional cross section of a blade.

annual average wind speed: The time averaged, mean, horizontal wind speed for one calendar year at a particular site and a specified height.

annual average turbulence intensity: A measure of the short-time and spatial variation of the inflow wind speed about its long time average.

availability: The ratio of the number of hours that a turbine could operate to the total number of hours in that period, usually expressed as a percentage. Downtime due to faults or maintenance (scheduled or otherwise) generally make up the unavailable time.

bedplate: In a modular system, the structure that supports the drive train components and nacelle cover. Also called a main frame.

blade: The component of the rotor that converts wind energy into rotation of the rotor shaft.

brake: A device capable of stopping rotation of the rotor or reducing its speed.

certification: Procedure by which a third party gives written assurance that a product, process or service conforms to specified requirements, also known as conformity assessment.

certification standard: Standard that has specific rules for procedures and management to carry out certification of conformity.

control system: A system that monitors the wind turbine and its environment and adjusts the wind turbine to keep it within operating limits.

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Table 1 - Symbols

Symbol	Term	Units	Where first used
C	Basic dynamic load rating	N	Eq 1
C_0	Basic static load rating	N	5.1.3.1
f_{ma}	Mesh misalignment	- -	5.1.1.3
K_A	Ratio between the equivalent and the nominal torque	- -	5.1.1.5
$K_{H\beta}$	Load distribution factor	- -	5.1.1.2
K_{lc}	Ratio of maximum contact pressure to contact pressure for line contact without misalignment	- -	Eq 4
K_m	Ratio of maximum contact pressure with misalignment to maximum contact pressure without misalignment	- -	Eq 4
K_v	Dynamic factor	- -	5.1.1.1
k	Load sharing factor for the maximum loaded roller	- -	Eq 2
L_{adv}	Combined advanced rating life	hours	5.1.3.2.3
$L_{adv,\,i}$	Advanced rating life on the ith load level	hours	Eq 5
L_{h10}	Basic rating life	hours	Eq 1
L_{we}	Effective roller length	mm	Eq 3
L_{10r}	Combined nominal reference rating life	hours	5.1.3.2.3
n	Rotational speed	rpm	Eq 1
P	Dynamic equivalent bearing load	N	Eq 1
P_o	Equivalent static bearing load	N	5.1.3.1
P_t	Rated power of wind turbine	kW	Eq 6
p	Exponent in bearing life equation	- -	Eq 1
p_{line}	Contact pressure for line contact	MPa	Eq 3
p_{max}	Maximum contact stress	MPa	Table 3
Q	Single roller maximum load for a clearance free bearing	N	Eq 2
Q_{ty}	Recommended oil quantity	liters	Eq 6
q_i	Time share on the ith load level	- -	Eq 5
Ra	Roughness average	µm	5.2.8.2
Rz	Mean peak-to-valley height	µm	5.2.8.2
S_F	Safety factor for bending strength	- -	5.1.1.4
S_H	Safety factor for pitting resistance	- -	5.1.1.4
Y_N	Stress cycle factor for bending strength	- -	5.1.1.4
Y_{NT}	Life factor for bending	- -	5.1.1.5
Z	Total number of rolling elements	- -	Eq 2
Z_N	Stress cycle factor for pitting resistance	- -	5.1.1.4
Z_{NT}	Life factor for pitting resistance	- -	5.1.1.5
α_0	Nominal contact angle of the bearing	degrees	Eq 2
$\Sigma\rho_{line}$	Curvature sum for line contact	- -	Eq 3
κ	Viscosity ratio	- -	5.1.3.3

cut-in wind speed: The minimum wind speed at hub height at which the control system calls for the turbine to produce power.

cut-out wind speed: The maximum wind speed at hub height at which the control system calls for the turbine to produce power.

damped yaw: A device used to slow yaw motions.

design life: The period of real time that the system is expected to continue functioning. Includes operating, idling and stopped time.

downwind turbine: A HAWT where the wind passes the tower before the rotor.

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dynamic equivalent bearing load: A hypothetical load, constant in magnitude and direction, acting radially on radial bearings or axially on thrust bearings, which if applied, would have the same influence on bearing life as the actual loads to which the bearing is subjected.

emergency shutdown: A rapid shutdown of the wind turbine triggered by the control system, a protection system or manual intervention.

extreme load: The extreme load is that load from any source, either operating or non-operating, that is the largest single load that the gearbox will see during its design life beyond which the gearbox no longer satisfies the design requirements. This load can be either forces, moments, torques, or a combination of the three. This load, supplied by the wind turbine manufacturer, includes all partial load safety factors.

extreme torque: The extreme torque is that torque from any source that is the largest single torque that the gearbox will see during its design life beyond which the gearbox no longer satisfies the design requirements.

extreme wind speed: The highest short-term average wind speed that is likely to be experienced by the wind turbine during its service lifetime. It is typically based on statistical estimates of the long term behavior of the wind speed.

feathering: In a variable pitch HAWT, the action of pitching the blades to a minimum power production position.

fixed pitch rotor: A rotor with blades that do not change pitch during operation. The pitch angle of the rotor blades may be changed manually for site specific or seasonal wind spectrum changes.

free yaw: See passive yaw.

HAWT: Horizontal axis wind turbine. The rotational axis of the rotor is approximately parallel to the horizon.

horizontal axis: The axis of rotor rotation is approximately parallel to the horizon.

hub height: For a HAWT, the height to the center of the rotor.

hub: The structure that attaches the blades to the rotor shaft.

idling: Operating condition where the rotor is rotating and the generator is not producing power.

input or mechanical power: The mechanical power measured at the gearbox low speed shaft or the wind turbine rotor shaft.

input shaft: See rotor shaft.

integrated system: A system architecture in which the gearbox housing supports the rotor directly, and in some cases, the generator(s) and other components. See modular system.

lock: The use of a mechanical device to prevent movement of the rotor or yaw drive.

main frame: See bedplate.

main shaft: See rotor shaft.

maximum operating load: The maximum operating load is the highest load in the load spectrum.

maximum power: The highest level of net electrical power delivered by a wind turbine in normal operation.

Miner's sum dynamic equivalent bearing load: The dynamic equivalent bearing load obtained by combining loads and speeds in a wind spectrum using Miner's rule.

modular system: A system architecture in which the rotor shaft assembly, gearbox, generator(s) and, possibly, a yaw drive, are separate components mounted to a common main frame. See integrated system.

motoring: Operating condition where the generator is consuming power.

nacelle: The structure that contains the drive train and other components located at the top of a HAWT.

nacelle cover: The housing that covers the nacelle.

nominal speed: The gearbox low speed shaft speed at which mechanical power is defined.

non-rotating: Operating condition where the rotor is not rotating.

normal shutdown: Transitional operating condition where the rotor decelerates from operating speed to standstill or idling and the generator ceases to generate power.

operational wind speed range: The range of wind speeds between the cut-in and the cut-out speed.

output shaft: See high speed shaft.

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parked: Operating condition where the rotor is not rotating because the parking brake is applied.

parking brake: A device capable of preventing rotor rotation.

passive yaw: The forces of the wind are used to align the nacelle (rotor disk) relative to the changing direction of the wind. See active yaw.

pitch: The angular position of the rotor blades about their long axis.

pitch control: Rotor shaft torque limiting is accomplished by actively adjusting the pitch.

preventive maintenance: Scheduled work intended to prevent failure or unscheduled repairs.

rated power: The continuous electrical power output assigned by the WTGS manufacturer that the wind turbine is designed to achieve under normal operating conditions at rated wind speed.

rated wind speed: The specified wind speed, assigned by the WTGS manufacturer, at which the rated power is produced.

rotor: The hub/blade assembly.

rotor bearing(s): The bearing(s) that supports the rotor shaft.

rotor diameter (horizontal axis): Diameter of the disk swept by the rotation of the blades.

rotor shaft: The shaft that supports the rotor and transmits the rotor torque to the gearbox. Also called the main shaft.

rotor speed: The rotational speed of the wind turbine rotor about its axis, in revolutions per minute.

stall control: Rotor shaft torque limiting is accomplished by aerodynamic design (airfoil selection, blade taper, blade twist, blade pitch, rotor speed).

standstill: See non–rotating.

startup: Transitional condition where the rotor accelerates from standstill or idling to operating speed and the generator begins to generate power.

tower: The structure that supports the nacelle in a HAWT.

turbulence intensity: A statistical measure of the variation in the wind speed. The ratio of the standard deviation of the wind speed to the mean wind speed.

upwind turbine: A HAWT where the wind passes the rotor before the tower.

variable pitch rotor: A rotor whose blade pitch can be varied during operation. The pitch angle may be actively controlled to optimize power or limit loads in response to the conditions.

variable speed: Rotor shaft torque limiting is accomplished by using high voltage electronic components and special generator designs to allow a wide range of rotor speeds. This method utilizes changes in inertial energy in the rotor to absorb the effect of wind gusts.

VAWT: Vertical axis wind turbine. The rotational axis of the rotor is approximately perpendicular to the horizon. This kind of turbine is beyond the scope of this standard.

wind turbine generator system (WTGS): A system that converts the kinetic energy of the wind into electrical power.

wind turbine manufacturer: Entity that designs, manufactures and warrants wind turbines.

wind turbine operator: Entity that operates and maintains wind turbines.

yaw: Rotation of a HAWT's nacelle about the long axis of its tower. Used to orientate the nacelle (rotor disk) with respect to the prevailing wind.

yaw bearing: The bearing system that supports the nacelle in a HAWT. It permits the nacelle to rotate about the tower axis.

yaw drive: The system of components used to cause yaw motion.

3.3 Gearbox terms

alloy steel: Steel containing significant quantities of alloying elements such as nickel, chrome, or molybdenum to improve its properties such as hardenability or toughness.

ambient temperature: The dry bulb air temperature within the immediate vicinity of the gearbox.

annulus gear: Gear wheel with teeth on the inner surface of a cylinder. Also known as an internal gear.

aspect ratio: The ratio of the pinion face width to the pinion operating pitch diameter.

bearing basic rating life: The life where adjustment factors for reliability, material and environment are taken as unity (1.0).

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bearing manufacturer: Entity that designs, manufactures and warrants bearings for wind turbine gearboxes.

bulk oil: The oil that is most representative of the overall physical condition of the lubricant within the lubrication system. With splash lubricated gearboxes, the location of this lubricant is at or near the midpoint of the oil sump level shortly after the drive is shut down at operating temperature. With pressure fed lubrication systems, this is represented by the oil within the pressure line between the oil pump and filter assembly during system operation.

carburizing: A heat treatment process where gears are heated in a carbon rich atmosphere (usually gas carburizing) that causes carbon to diffuse into the surface layers of the gear teeth. The gears are hardened by either quenching from the carburizing temperature or they are cooled, reheated and quenched. The carburizing and hardening is followed by tempering where the gears are reheated to a relatively low temperature and slowly cooled.

coupling: A device that connects two rotating shafts to transmit power, accommodate misalignment, and compensate for axial movement.

double helical gear: Gear wheel with both right-hand and left-hand helices. The teeth are separated by a gap between the helices.

epicyclic: Gear arrangement consisting of multiple parallel axis gears including a sun pinion, several planets that mesh with the sun, planet carrier, and an annulus gear that meshes with the planets.

gear: Of two gears in a gearset, the one with the larger number of teeth is the gear. Also known as the wheel. See pinion.

gearbox: A complete assembly of gears, shafts, bearings, housing, seals, lubrication system and associated components.

gearbox manufacturer: Entity that designs, manufactures and warrants gearboxes for wind turbines.

gear ratio: The ratio of the larger to the smaller number of teeth in a pair of gears.

gearset: A pinion and gear that are intended to run together.

grinding notch: A discontinuity produced by a grinding tool between the start of active profile and tooth root that increases the tooth root stress.

helical gear: A gear with teeth that are inclined to the gear axis like a helical screw.

helix modification: A manufacturing modification of a pinion or gear obtained by changing the shape of the tooth flank along the face width.

high speed shaft: The highest speed shaft in a gearbox that drives the generator.

housing: The enclosure that contains the gearbox components such as gears, shafts, bearings and associated components.

inner ring: In a bearing, the material between the inside dimension of the roller/ball and the outside diameter of the part the bearing is mounted on.

involute profile modification: A manufacturing modification of a pinion or gear where a small variable amount of material is removed along the tooth profile in the root to tip direction.

low speed shaft: The lowest speed shaft in a gearbox. See rotor shaft.

lubricant manufacturer: Entity that designs, manufactures and warrants lubricants for wind turbine gearboxes.

module: The ratio of the pitch diameter in millimeters to the number of teeth in a gear.

nitriding: Heat treatment process where gears are heated in a nitrogen atmosphere that causes nitrogen to diffuse into surface layers of gear teeth and form hard nitrides. Distortion is small, because nitriding is done at low temperatures and there is no quench.

outer ring: In a bearing, the material between the outside dimension of the roller/ball and the bore of the part the bearing is mounted within.

parallel shaft: A gear arrangement where the pinion and gear mesh on parallel axes.

pinion: Of two gears in a gearset, the one with fewer number of teeth is the pinion. See gear.

planetary: An epicyclic gear arrangement where the annulus is fixed, the planets rotate about their own axes, and the planet carrier rotates.

power take-off (PTO): Additional output shaft for driving auxiliary equipment, such as oil pumps.

profile shift: A modification of a gear where the tooth profile is radially shifted.

protuberance cutter: A tool for cutting gear teeth that cuts a relief in the profile of the gear teeth to avoid grinding notches.

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purchaser: Entity that issues purchase contracts for wind turbine gearboxes.

rim thickness: The radial distance from the roots of the teeth to the inner diameter of the rim or to the bore on an external gear, and to the outside diameter on an internal gear.

split power path: A gear arrangement consisting of multiple parallel axis gears. The arrangement is analogous to epicyclic gears except there is no annulus gear.

spur gear: Gear with teeth that are parallel to the gear axis.

star: An epicyclic gear arrangement where the annulus rotates, the planets (stars) rotate about their own axes, and the planet carrier is fixed.

through hardening: Heat treatment process where gear blanks are austenitized, rapidly quenched to obtain a predominantly martensitic microstructure, and tempered. Gear teeth are machined after through hardening to avoid distortion.

> **NOTE:** Through hardening does not imply that the part has equivalent hardness throughout the entire cross-section.

torque arm: A structural component that attaches to the housing of a shaft mounted gearbox and prevents rotation of the gearbox about the rotor shaft.

wheel: Of two gears in a gearset, the one with the larger number of teeth is the wheel. Also known as the gear.

3.4 Filtration terms

cleanliness level: The cleanliness level as defined by ISO 4406 is a code system used to quantify the number of particles of a certain size in a given volume of oil. See annex F.

filter: A device for removing solid particles from a liquid stream, typically by means of porous media.

inline filter: A filter installed in the main oil circulation system that supplies gears and bearings with oil.

offline filter: A filtration device independent of the main lubrication system, typically with a separate pump, that operates continuously to improve oil cleanliness. Also called bypass filter, kidney loop filter, or side stream filter.

4 Design specification

4.1 Introduction

This clause provides the minimum required information for the specification of wind turbine gearboxes. It is important for the purchaser to identify what is expected of the gearbox manufacturer. A thorough specification is the method by which this is done. The scope of the specification may range from only performance and life requirements to detailed design and method of calculation requirements. If wind turbine certification is required, the purchaser shall clearly specify all certification documents relevant to the gearbox. The specification should contain the information noted in annex E.

4.2 Specification introduction

An introduction shall be provided that identifies the intent of the procurement specification. This shall include a description of the type of wind turbine, its basic modes of operation, the application of the gearbox in the wind turbine, and a description of the interfaces to the gearbox such as generator, rotor, bedplate, torque arm, lubricant system and accessories.

4.3 Gearbox configuration

4.3.1 Configuration

The general configuration of the gearbox shall be specified. This may include: the type of mounting; the type of gearing; the gear arrangement; the number of high speed shafts; the location and type of power take-off gears (PTO); and the method of lubrication.

All requirements for the geometric configuration of the gearbox shall be specified. This may include: the overall length, width, or height; the distance between shaft centers; length of shaft extensions; angle of shaft tilt or offset; gear housing split plane; the maximum weight, or other features.

A detailed description of all components interfaced to the gearbox shall be provided. Each interface shall be detailed for mounting, support and loading.

4.3.2 Rotor speed

The rotor speed, or speed range, shall be specified. This shall include expected speed during power production and idling mode. The direction of rotation for each of these situations shall be specified.

4.3.3 Gear ratio

The overall gear ratio and its tolerance shall be specified for the drive gears and any PTO gears.

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The overall gear ratio of the gearbox is set by the requirements for rotor speed and generator speed. However, if there is more than one stage of gears, the gearbox manufacturer can select gear ratios for each stage to maximize load capacity and minimize weight (see AGMA 901-A92).

4.4 Loading

4.4.1 Description of loads

It is the responsibility of the wind turbine manufacturer to provide all loads applied to the gearbox to allow adequate evaluation of the design life requirements for all gears, bearings, shafts and the housing (see annex B). The details of this load description are presented in the following sections.

The loads should be thoroughly detailed in a load description document. This document should include:

- torque-frequency histogram including all operating loads;
- transient loads described as annotated time series. Refer to section B.5.2.2 and figure B.2 for a sample of an annotated brake event;
- torque-speed relationships; and
- other structural loads described in fatigue-based cycle counts at pertinent interfaces. These loads can be presented as a representative time series of the loads or the results of a Rainflow Count [1] with mean value, amplitude (peak-to-peak), and frequency of occurrence.

The purchaser shall indicate in the loads document the partial safety factors and load uncertainty factors used in deriving the loads. Any additional multipliers to be applied to the loads shall be explicitly stated. The source and rationale for the use of the safety factors or multipliers or both shall be described or sufficiently referenced.

4.4.2 Torque loads

4.4.2.1 Fatigue

The low speed shaft torque spectrum shall be specified in bins with:

- torque level;
- cycles or revolutions per torque level;
- nominal speed for design;
- idling speed.

It shall be clearly stated as to which portion of the turbine lifetime the spectrum refers.

For variable speed wind turbines, it may be necessary to separate each torque bin into several speed bins.

Specified torque level of each bin shall represent the highest level of torque represented in that bin. To avoid excessive conservatism, sufficient quantity of bins (at least 40) shall be used. Bin width need not be uniform, and, in fact, finer resolution at the highest torque bins is preferred. The load spectrum shall also contain one bin that accounts for idling and stopped time. The load spectrum total time will then match the design life of the turbine.

The torque spectrum shall include all fatigue loads, including all external transient loads such as brake loads, if applicable. If more than a single driven load, such as multiple generators, pump drives, or other PTO's exist, the torque spectrum for each driven load shall be defined.

4.4.2.2 Extreme torque loads

Extreme torque shall be specified by the wind turbine manufacturer:

- torque level;
- number of occurrences;
- source, such as rotor, generator or brake.

Extreme loads shall not be included in the load spectrum.

4.4.3 Structural loads

4.4.3.1 Non-torque load sources

In the case that the wind turbine rotor operation imparts non-torque loads to the gearbox low speed shaft, these loads shall be sufficiently described in the specification. Such loads may occur in any operating mode of the wind turbine including idling mode or when the turbine is parked. In modular arrangements the shafts are subjected to loads that need to be tolerated and transferred to the base mount (see A.5). Also, the generator, brake and other interfaced components can affect reaction loads on the gearbox and shafts. Such loads may occur in any operating mode of the wind turbine including idling mode or when the turbine is parked. These loads shall be sufficiently described in the specification. Stiffness in all loading directions of compliant supports, such as elastomeric bushings, shall be specified by the purchaser.

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4.4.3.2 Structural fatigue loads

For each type of external load applied to the gearbox, fatigue loads shall be defined at the interface in a prescribed coordinate system as moments and forces in three directions.

Loads for each axis shall be defined in a spectrum, specified by bins, with:

- moment level;
- force level;
- cycles per revolution per moment/force level.

Multi-axis loads shall be provided in such a way that the phase relationship is preserved. The interpretation and use of data shall be a joint effort between the wind turbine manufacturer and the gearbox manufacturer.

4.4.3.3 Structural extreme loads

For each type of external loading to the gearbox, extreme loads shall be defined at the interface, in a prescribed coordinate system, with moments and forces in three directions.

4.4.4 Idling, parking and transient operation

Features such as duration and frequency of speeds and loads, method of lubrication, and temperature ranges during idling, parking and transient operation shall be specified.

4.4.4.1 Idling

Rotors should be allowed to idle whenever possible to avoid false brinelling, fretting corrosion, and corrosion on gear teeth, splines, bearing rollers and bearing raceways.

4.4.4.2 Parking

Parking should be minimized to avoid false brinelling, fretting corrosion, and corrosion on gear teeth, splines, bearing rollers, and bearing raceways. Dynamic loads on a parked wind turbine shall be specified in detail, for example, by an annotated time series.

4.4.4.3 Transient operations

Transient load events such as braking, cut-in, cut-out, generator shift, and blade pitch operations, shall be specified in detail, for example, by an annotated time series.

4.4.5 Dynamic loading

The loads specified by the purchaser shall include effects from the system's dynamics. Depending on the layout of the drive train and nacelle, the purchaser should quantify static and dynamic relative displacements of the different subsystems. The purchaser should also specify absolute movements and accelerations of the gearbox. Implementation of this dynamic analysis is a joint effort between purchaser and gearbox manufacturer. To enable the purchaser to perform the dynamic analysis during the development process, the gearbox manufacturer should supply general gearbox data, such as center of gravity, stiffness, inertia, damping, and clearances.

4.5 Certification

Wind turbines are usually certified to facilitate due diligence efforts and insurance requirements. All requirements for certification shall be described in the specification, including:

- name of classification society;
- standard or certification document name, number, and revision level;
- applicable section or paragraphs;
- any exceptions to the above documents.

Safety factors in excess of certification standard requirements shall be specified. It is the responsibility of the purchaser that the purchase specification is consistent with the relevant certification standards.

4.6 Operating environment

The expected operating environment of the gearbox shall be specified. As a minimum this shall include: the ambient temperature range, air temperature and air velocity across the gearbox, maximum relative humidity, extent of exposure to direct sunlight, precipitation and airborne particulates. See annex E for additional information on this subject.

4.7 Control and monitoring

Requirements for monitoring and control sensors associated with the gearbox shall be specified. This may include: lubricant level, temperature, pressure sensors, vibration sensors, filter bypass sensors, particulate accumulation sensors, or others.

4.8 Qualification testing

4.8.1 Prototype tests

Type and conditions of any qualification testing shall be specified. These tests shall be performed on a

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prototype unit of the gearbox with gears and housings previously measured and recorded. Test duration shall be agreed to between the purchaser and gearbox manufacturer. The test shall observe the development of contact pattern with rising load (documented at defined steps). The test shall be performed up to the design load for the helix and profile modifications. The test results shall be well documented by photos and shall be compared to the design calculations in order to verify the calculation model. In addition, bearing and oil temperatures should be reported. Typical test length should be a function of the time required for system temperatures to stabilize.

In addition to the contact pattern development and thermal capacity test, an instrumented overload, long duration prototype qualification test may be required. Test conditions for such a test are to be agreed to between the purchaser and the gearbox manufacturer at the time of purchase.

4.8.2 Sound emission testing

Acceptance tests for verification of the sound emissions from the gearbox shall be specified. Testing conditions for the acceptance measurements shall also be specified. ANSI/AGMA 6025-D98 (sound pressure measurement) and ISO 8579-1 (sound power measurement) are two possible methods for measurement. The specification of test conditions may include: speed, torque, temperature, type of lubricant, direction of rotation, mounting of gearbox, type of drive motor, type of measurement device, range of frequencies, and others. In addition, it may be desirable to specify the tonal character of the sound emission.

4.8.3 Vibration testing

Acceptance tests for verification of the vibration of the gearbox shall be specified. Testing conditions for the acceptance measurements shall also be specified. ANSI/AGMA 6000-B96 and ISO 8579-2 are two possible methods for measurement. The specification of test conditions may include: speed, torque, temperature, type of lubricant, direction of rotation, mounting of gearbox, type of drive motor, type of measurement device, range of vibrational velocity or acceleration, and others.

4.9 Startup requirements

Wind turbines are subject to operating at their full design load capabilities immediately after installation. At this point, the roughness profile of gears and bearings is still dominated by the texture generated in the finishing process. High local contact stress at roughness peaks may cause micropitting, surface distress, scuffing or other irreversible wear processes. The surface finish achieved during the manufacturing process significantly influences these risks. Methods like super-finishing, surface plating or run-in at controlled power levels may be required to reduce these risks.

The wind turbine manufacturer and the gearbox manufacturer shall mutually agree among the above methods with consideration of the actual geometry, manufacturing processes, stress level, slide ratio, λ-ratio, and operating modes of the wind turbine. The suitability of the selected means shall be documented, for example, by prototype testing. When possible, the wind turbine operator should also agree on the above methods.

Uncertainties of the wind make it difficult to achieve proper run-in conditions during field commissioning, therefore a run-in will typically be performed at the factory. Operating and load conditions of a run-in shall be selected with care to avoid initial damage to gears and bearings. Refer to table 17 for oil cleanliness at the run-in test.

The wind turbine manufacturer and the gearbox manufacturer shall mutually determine and agree to startup, commissioning, operating and maintenance procedures designed to avoid gearbox damage during initial turbine startup. The wind turbine manufacturer shall include these procedures in the startup, commissioning, operating and maintenance documentation. The wind turbine operator shall follow the procedures as outlined in the operating and maintenance documentation.

4.10 Transport and construction

There are many possible sources of damage to the gearbox during transport, storage, and construction. Of particular concern are contamination, corrosion, fretting corrosion and false brinelling. It shall be the responsibility of the gear manufacturer to ensure appropriate steps are taken to minimize potential for contamination, corrosion, fretting corrosion and false brinelling during storage at their facilities and transport. It shall be the responsibility of the purchaser to ensure appropriate steps are taken to minimize potential for contamination, corrosion, fretting corrosion and false brinelling during storage, transport, and construction. This is especially true

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for turbines that are parked after blades are installed prior to commissioning and operations.

5 Gearbox design and manufacturing requirements

5.1 Component rating

The required design life shall be specified for each of the major subsystems of the gearbox including gears, bearings, housings, shafts and seals. To the extent possible, the specification should identify the minimum required life, the reliability associated with the life calculation, and the method or standard to be used for the calculation.

The gearbox shall be designed to withstand a momentary, maximum load as defined in annex B and specified by the purchaser. All major components of the gearbox including gears, keys, shafts, splines, bearings, housing and fasteners shall be capable of withstanding the extreme load without incurring significant permanent deformation that will affect performance. The maximum load capacity should be calculated in accordance with ANSI/AGMA 6110–F97, or ANSI/AGMA 6123-A88, or ANSI/AGMA 6001-D97, or ISO/TR 13593, or DIN 743 as agreed to between the purchaser and the gearbox manufacturer.

5.1.1 Gear life rating

Pitting resistance and bending strength shall be rated in accordance with ANSI/AGMA 2101-D04, or ISO 6336:1996. The following clauses give application rules for each of these standards. The purchaser and gearbox manufacturer shall agree to the applicable standard.

5.1.1.1 Dynamic factor, K_v

The dynamic factor, K_v, significantly affects gear rating. A minimum value of $K_v = 1.05$ shall be used unless a detailed dynamic analysis using a proven multi-body simulation, shows otherwise.

5.1.1.2 Load distribution factor, $K_{H\beta}$

The load distribution factor, $K_{H\beta}$, significantly affects gear rating. Load distribution along the face width is influenced by many parameters such as elastic deflections, manufacturing tolerances, and thermal deformations as described in clause 15 of ANSI/AGMA 2101–C95 and clauses 7.1 and 9.1 of ISO 6336-1:1996. Profile and helix modifications shall be used to compensate for the detrimental effects of these deviations.

5.1.1.3 Advanced contact analysis

Load distribution factors shall be determined by numerical analysis with an advanced contact analysis. If the advanced 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. Lower values may apply to self adjusting systems. The load distribution shall be validated by testing as described in clause 4.8.1.

The design load for profile and helix modification should correspond to the load that contributes most to the surface fatigue. The value for $K_{H\beta}$ established at this load shall be held constant across the entire load spectrum for gear rating. Additionally, maximum operating loads and extreme tolerance combinations shall be checked with their resulting contact stress. Special care shall be taken to avoid stress risers at the extremities of the contact area.

There are numerous computer codes available for analyzing the load distribution along the face width. The design process of a wind turbine gear shall employ a code that allows analysis of the load distribution in helix and profile directions at the same time, providing full information of the local loading in the entire contact area. In addition to the requirements of 5.1.1.2, an advanced theoretical contact analysis of the load distribution shall at least account for:

- load distribution in helix and profile directions;
- influence of adjacent meshes;
- influence of local discontinuities in the stiffness at the extremities of the contact area.

The effect of production variation on shaft parallelism and tooth alignment of pinion and gear should be included in the value of mesh misalignment, f_{ma}. Annex G suggests a method that can be used.

5.1.1.4 Gear rating according to ANSI/AGMA 2101-D04

Miner's Rule (see ISO 6336-6) shall be used to calculate gear life using a load spectrum supplied by the purchaser. Gear life calculations shall be based on a reliability of 99% and the lower curves for stress cycle factors, Z_N and Y_N. Pitting and bending fatigue lives shall be a minimum number of hours specified by the purchaser, but not less than the design life hours. The minimum safety factor for pitting

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resistance shall be S_H = 1.0. The minimum safety factor for bending strength shall be S_F = 1.0.

Scuffing resistance shall be rated in accordance with AGMA 925–A03. If scuffing temperature is determined from FZG tests, one stage lower than the fail load stage shall be used for the scuffing analysis. The risk of scuffing shall be less than 5%.

5.1.1.5 Gear rating according to ISO 6336:1996

Miner's Rule (see ISO 6336-6) shall be used to calculate safety factors using a load spectrum supplied by the purchaser. Life factors, Z_{NT} and Y_{NT} shall be reduced to 0.85 at 10^{10} cycles. Pitting and bending fatigue lives shall be a minimum number of hours specified by the purchaser, but not less than the design life hours. Alternatively, a rating calculation using an application factor, K_A, in accordance with annex H, may be used if agreed to by the purchaser and the gearbox manufacturer. In this case, K_A is defined as the ratio between the equivalent and the nominal torque.

In both cases, safety factor calculations shall be based on a reliability of 99%. 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.

ISO 6336:1996 does not provide a rating method for scuffing. The purchaser and gearbox manufacturer shall therefore agree on the rating method for this failure mode. It is highly recommended to use a method based on total contact temperature, such as AGMA 925-A03, or ISO/TR 13989-1, or DNV Classification Note 41.2. If scuffing temperature is determined from FZG tests, one stage lower than the fail load stage shall be used for the scuffing analysis. When using AGMA 925-A03, the risk of scuffing shall be less than 5%. When using ISO/TR 13989-1 or DNV 41.2, the minimum safety factor shall be 1.25

5.1.2 Gearbox thermal rating

Gearbox thermal power rating should be calculated in accordance with ANSI/AGMA 6110–F97, or ANSI/AGMA 6123–A88, or ISO/TR 14179-1. These documents provide tools to determine maximum operating temperatures of a given gear design. If the purchaser and the gearbox manufacturer agree, alternate methods can be used to compute the gearbox thermal rating.

5.1.3 Bearing rating

Bearings shall be rated according to 5.1.3.1 and 5.1.3.2. These requirements are empirical guidelines. Calculated bearing lives are valid for comparison of different bearing options, but may not reflect actual bearing lives under service conditions.

5.1.3.1 Static rating

The static safety factor of any bearing shall be greater than 3.0 at the maximum operating load on that bearing, and 2.0 at the extreme load on the bearing.

The static safety factor should be calculated from the actual internal load distribution. If not available, the ratio C_0/P_0 according to ISO 76 shall be used.

5.1.3.2 Rating life

Bearing life shall be calculated using DIN ISO 281 Bbl. 4:2003 (see 5.1.3.2.3). Proprietary advanced life calculation methods developed by many bearing manufacturers may be used. When proprietary methods are utilized, the results shall be compared to DIN ISO 281 Bbl. 4:2003 and the values produced by 5.1.3.2.1 through 5.1.3.2.2. Discrepancies should be rationalized between the bearing manufacturer, gearbox manufacturer, and purchaser.

5.1.3.2.1 Basic rating life

For a preliminary selection of bearings in the design process of the gearbox, the basic rating life calculation should be used.

This standardized calculation method for dynamically loaded rolling bearings is based on equivalent load, speed, bearing dynamic load rating and simplified load distribution assumptions. See ISO 281 and the catalogues of rolling bearing manufacturers. Table 2 offers minimum values for the basic rating life, L_{h10}, which is determined in accordance with ISO 281:

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

where

L_{h10} is basic rating life, hours;

n is rotational speed, rpm;

C is basic dynamic load rating according to ISO 281, N;

P is dynamic equivalent bearing load, N;

p is life exponent.

Miner's rule shall be used to combine loads and speeds given in the load spectrum supplied by the purchaser. The exponent, p, shall be 3.0 for ball bearings and 10/3 for roller bearings.

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Table 2 – Minimum basic rating life, L_{h10}

Bearing position	Required life, L_{h10}, hr
High speed shaft	30,000
High speed intermediate shaft	40,000
Low speed intermediate shaft	80,000
Planet	100,000
Low speed shaft	100,000
NOTE: Values in this table are valid for a design life of 20 years and shall be adjusted for designs with different design life.	

5.1.3.2.2 Contact stress

The contact stress using the Miner's sum dynamic equivalent bearing load should not exceed the values listed in table 3. Contact stress is calculated with equation 4 or methods described in 5.1.3.2.3, and differences between these methods should be resolved between the bearing manufacturer, the gearbox manufacturer and the wind turbine manufacturer. The results of an advanced contact analysis per 5.1.3.2.3 are preferred.

Table 3 – Guide values for maximum contact stress for rolling element bearings at Miner's sum dynamic equivalent bearing load

Bearing position	Maximum contact stress, p_{max}, MPa
High speed shaft	1300
High speed intermediate shaft	1650
Low speed intermediate shaft	1650
Planet	1450
Low speed shaft	(there is no equivalent load on the input shaft)
NOTE: Values in this table are valid for a design life of 20 years and shall be adjusted for designs with different design life.	

A simplified method to approximate this contact stress in radial roller bearings is as follows:

$$Q = \frac{P_0}{Z \cos \alpha_0} k \qquad (2)$$

where

Q is single roller maximum load for a clearance free bearing, N;

P_0 is equivalent static bearing load, N;

Z is total number of rolling elements;

α_0 is nominal contact angle of the bearing, degrees;

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

$$p_{line} = 270 \sqrt{\frac{1}{2} \left(\frac{Q}{L_{we}}\right) \Sigma \rho_{line}} \qquad (3)$$

where

p_{line} is contact pressure for line contact, MPa;

L_{we} is effective roller length, mm;

$\Sigma\rho_{line}$ is curvature sum for line contact.

$$p_{max} = K_{lc} K_m p_{line} \qquad (4)$$

where

p_{max} is maximum contact stress, MPa;

K_{lc} is ratio of maximum contact pressure to contact pressure for line contact without misalignment;

K_m is ratio of maximum contact pressure with misalignment to maximum contact pressure without misalignment.

For more details refer to annex I.

5.1.3.2.3 Advanced rating life

The advanced rating life shall be equal to or greater than the design life specified by the wind turbine manufacturer. Calculations shall be performed bin-by-bin using the load spectrum specified by the wind turbine manufacturer. If the combined advanced rating life, L_{adv}, is greater than 10 × L_{10r} (combined nominal reference rating life according to DIN ISO 281 Bbl. 4:2003), then the combined advanced rating life, L_{adv}, shall be set equal to 10 × L_{10r}. Note that L_{10r} is expressed in number of rotations in the DIN ISO 281 Bbl. 4:2003 document. This needs to be converted to hours for comparison with L_{adv}.

Miner's rule shall be used to reduce the number of bins in the spectrum, to facilitate data processing, but not less than 10 load bins shall be used. The life exponent used for spectrum reduction should be the same as used in the advanced life rating calculation.

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Combined rating life shall be calculated by the use of equation 5:

$$L_{adv} = \frac{\Sigma q_i}{\Sigma \frac{q_i}{L_{adv,\,i}}} \qquad (5)$$

where

L_{adv} is combined advanced rating life, hours;

q_i is time share on the ith load level;

$L_{adv,\,i}$ is adjusted rating life on the ith load level.

The advanced calculation shall include the effects of:

- radial, axial and moment loads;
- internal design of bearings;
- operating internal clearance (see 5.1.3.4);
- elasticity of bearings and shafts;
- load sharing between rolling elements;
- load distribution along roller length, considering actual roller and raceway profiles;
- truncation of ball/raceway contact;
- lubricant viscosity at operating temperature (see 5.1.3.3);
- operating lubricant cleanliness (see 5.1.3.5).

The advanced calculation may further account for elasticity of mounting. Advanced calculations may further be used to study the sensitivity of load sharing and load distribution against misalignment and manufacturing variation. Statistical tolerance analysis should be used to determine the misalignment. Annex G suggests a procedure that can be used. Special care shall be taken to avoid excessive stress risers at the roller ends and contact truncations.

5.1.3.3 Bearing operating temperature

The viscosity ratio, κ, shall be calculated at the operating temperature of the lubricant supplied to the bearing as defined in table 4. For splash lubricated bearings, the operating temperature can be verified by measuring the temperature of the non-rotating ring. The temperature gradient given for pressure lubricated bearings requires that the oil flow to the bearing provides sufficient cooling to the bearing.

The oil sump temperature is the steady-state operating oil sump temperature. The oil inlet temperature is the steady-state temperature at the gearbox inlet port.

On-site tests of the prototype wind turbine shall verify that the cooling system is sufficient to maintain these temperature levels. If operating oil temperature is unknown, 5 °C below the shutdown temperature specified by the purchaser shall be used.

The operating temperatures of the bearings shall be verified during the prototype test of the gearbox.

Table 4 – Bearing lubricant operating temperature for calculation of viscosity ratio, κ

Bearing position	Operating temperature for splash lubricated bearings	Operating temperature for pressure lubricated bearings
High speed shaft	Oil sump +15 °C	Oil inlet temperature +5 °C
High speed intermediate shaft	Oil sump +10 °C	
Low speed intermediate shaft	Oil sump +5 °C	
Planet	Oil sump +5 °C	
Low speed shaft	Oil sump	

5.1.3.4 Operating clearance

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

- bearing initial clearance;
- fitting tolerances of shaft and housing;
- smoothing of fit for the specified surface finishes;
- temperature gradient between inner and outer ring according to table 5;
- thermal expansion coefficient of the materials;
- operating temperature (see 5.1.3.3).

It may further consider static deflection of mounting.

Statistical tolerance analysis should be used to determine operating clearance. If this is not available, the RMS-value of all extremes shall be used. In addition, extreme tolerance combinations shall be checked.

Table 5 provides typical temperature gradients observed under constant operation. Higher gradients may occur, for example under startup conditions, and these extremes may be relevant for selecting bearing internal clearance. The operating

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clearance used in the advanced life rating, 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 as a function of rated power of the turbine	
	≤ 500 kW	> 500 kW
High speed shaft	5 to 10 °C	10 to 20 °C
High speed intermediate shaft	3 to 8 °C	5 to 15 °C
Low speed intermediate shaft	3 to 8 °C	0 to 10 °C
Planet	−5 to 0 °C	−5 to 0 °C
Low speed shaft	0 to 5 °C	0 to 5 °C

5.1.3.5 Lubricant cleanliness

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.

An oil cleanliness level of –/21/18 based on ISO 4406 shall be used when calculating bearing life for non-filtered systems unless operating cleanliness is demonstrated to be better.

In cases where cleanliness levels better than the defaults are used in the calculation, the value used shall be one class worse than the documented cleanliness.

5.1.3.6 Minimum required operating load

When selecting bearing size and type, one should consider the risk of skidding damage between rolling elements and raceways at minimum load. The risk of skidding damage is affected by factors such as:

- bearing design and size;
- speed;
- acceleration;
- rate of change from unloaded to loaded conditions;
- lubrication factors, such as quantity, viscosity, temperature and additives;
- operating internal clearance.

This issue shall be negotiated between the gearbox manufacturer, bearing manufacturer, and purchaser.

5.1.4 Shaft life rating

Shafts shall be designed and rated for fatigue and yielding in accordance with ANSI/AGMA 6001–D97, or ISO/TR 13593, or DIN 743. All shaft loads shall be considered including applied structural loads. Fatigue life at 99% reliability shall exceed the specified design life. The purchaser shall specify applicable partial factors on loads and material in accordance with the certification rules. The gearbox manufacturer shall select and document minimum safety factors in accordance with the calculation methods selected.

5.1.5 Housings

Housings, planet carriers and other parts that support gears or bearings shall be checked for deflection under load, either by calculation or extrapolation of test data from previous similar parts. Planet carriers shall be treated the same as shafts for fatigue calculation. The influence of planet carrier deflection on gear mesh alignment shall also be checked.

5.1.6 Shaft seals

Seals can have limited life, thus the type of seal and its expected life should be discussed between the gearbox manufacturer and purchaser. Labyrinth seals are preferred over lip seals because lip seals have relatively short life and are difficult to replace in the turbine. See 7.1 for additional information.

5.2 Gear elements

Requirements for features of each gear element should be specified. The following list is recommended as a minimum. For resolution of parts deviating from these specifications refer to annex C.

5.2.1 Aspect ratio

The aspect ratio is an indicator of how sensitive a gearset is to misalignment. To achieve good load distribution on spur and single helical gears, the aspect ratio should be less than 1.25. For double helical gears the aspect ratio should be less than 2.0 (see AGMA 901-A92). Different limits may apply to self adjusting systems.

5.2.2 Profile shift

Profile shift is used to:

- prevent undercut;
- balance specific sliding;

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- balance flash temperature;
- balance bending fatigue life;
- avoid narrow top lands.

The profile shift should be large enough to avoid undercut and small enough to avoid narrow top lands (see AGMA 913-A98). For wind turbine gears, which are speed increasers, it is usually best to design the profile shift for balanced specific sliding.

5.2.3 Involute profile modification

Involute profile modification shall be used to minimize detrimental effects of tooth deflections, assembly tolerances, and tooth variations. Proper involute profile modification increases load capacity and reduces noise. Since the loads on wind turbine gears are variable, the design load point for the involute profile modification must be chosen carefully because the involute profile modification can be designed for only one load, and over modification is detrimental. The design modification shall account for effects of maximum loads, scuffing risk, maximum manufacturing variations, and transverse contact ratios at low loads.

5.2.4 Helix modification

Helix modification shall be used to minimize the detrimental effects of bending and torsional deflections of teeth, shafts, bearings, housing, and manufacturing variations. Proper helix modification increases load capacity and reduces noise. Since the loads on wind turbine gears are variable, the design load 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.

5.2.5 Planet gear rim thickness

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

5.2.6 Gear materials

The required properties and heat treat response of the gear material should be specified. The following items are recommended as a minimum.

5.2.6.1 External gears

All external gears shall be made from carburizing grade alloy steels with sufficient hardenability to obtain case and core properties meeting the requirements for grade 2 material in accordance with ANSI/AGMA 2101–C95, or grade MQ material in accordance with ISO 6336-5:1996 respectively, with the following exceptions:

- core hardness should be 28-45 HRC;
- certification in accordance with AMS 2301 or ASTM A534 should be required;
- the gear teeth shall be inspected for surface temper in accordance with 5.2.8.3.

Low hardenability alloys (such as AISI 8620) should not be used for modules coarser than 5 without special quality assurance procedures. The steel alloy shall be selected, and the heat treatment process shall be controlled, to obtain a microstructure that has strength and fracture toughness meeting the requirements for grade 2 material in accordance with ANSI/AGMA 2101–C95, or grade MQ material in accordance with ISO 6336–5:1996. PTO gears are not covered by this clause.

5.2.6.2 Internal gears

Internal gears shall be made from alloy steels with sufficient hardenability to obtain properties meeting the requirements for grade 2 material in accordance with ANSI/AGMA 2101-D04, or grade MQ material in accordance with ISO 6336-5:1996. The gears shall be heat treated to obtain a microstructure with strength and fracture toughness meeting the requirements of the application. The strength of the rim, the influence on tooth strength and the effects of bolt holes and threads shall be considered. Carburized or nitrided internal gears have better wear properties than through hardened gears.

5.2.6.3 Case depth

For both external and internal gears, case depth shall be selected based on the stress distribution. Attention shall be paid to the risk of embrittlement, subcase fatigue and case/core separation. Heat treatment distortion and resulting variation of case depth from grinding shall be evaluated for carburized and ground gears.

The effective case depth should be designed to avoid subcase fatigue failures and be in accordance with ANSI/AGMA 2101-D04 or ISO 6336-5:1996.

5.2.7 Gear accuracy

Geometric accuracy of gear elements shall be specified in accordance with ISO 1328-1. The maximum grades used shall be per table 6.

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,

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grinding after assembly with shafts should be considered.

Table 6 – Required gear accuracy

Gear type	Heat treatment	Maximum accuracy per ISO 1328–1
External	Carburized	6
Internal	Carburized	7
Internal	Nitrided	7 (with 8 for run-out and total cumulative pitch deviation)
Internal	Through hardened	8

5.2.8 Gear manufacturing

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

5.2.8.1 Method of manufacturing

All external gear teeth should be cut with protuberance cutters with minimum tool tip radius of 0.25 × normal module.

All external gear teeth should be ground on the flanks only. Grinding notches in the tooth flanks or in the root fillets are not permitted. The grinding process requires appropriate design, selection of tools, and planning of the production process.

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 chamfer should be specified on the drawings.

5.2.8.2 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. Maximum surface roughness shall be Ra=0.8 µm or Rz = 5 µm. Maximum surface roughness for internal gears shall be Ra = 1.6 µm. To reduce the risk of micropitting, the surface roughness listed in table 7 is recommended.

Active flanks of gear teeth should not be shot peened after final finishing because shot peened flanks are more likely to produce micropitting on mating gear teeth.

See annex F for further information about surface roughness and boundary lubrication.

Table 7 – Recommended gear tooth surface roughness

Gear	Recommended maximum roughness, Ra, as manufactured, µm
High speed pinion and gear	0.7
Intermediate pinion and gear	0.7
Low speed pinion and gear	0.6
Low speed sun and planet	0.5

5.2.8.3 Surface temper inspection after grinding

Surface temper inspection after grinding shall be specified by a sampling plan agreed to between the gearbox manufacturer and the purchaser. Sampling rate shall be based on effectiveness of process control and rejection rate on similar parts.

A well established, reliable inspection method is surface temper etch inspection in accordance with ISO 14104. All other inspection methods shall explicitly be agreed to between the purchaser and the gearbox manufacturer.

5.2.9 Gear arrangements

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

5.2.10 Lifting holes

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

5.3 Bearings

Requirements for features of each bearing element should be specified. The following list is recommended as a minimum.

5.3.1 Bearing types and arrangements

Tables 8 through 16 provide guidance for selecting appropriate bearing types and arrangements for a wind turbine gearbox. 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 not experienced.

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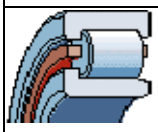
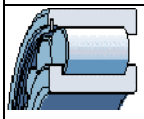
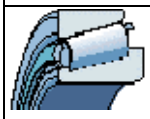
Table 8 – Bearings for combined loads

Symbol	Type	Abbreviation	General comments
	spherical roller bearing	SRB	The high load and misalignment capability of the SRB can be advantageous in wind turbine gearboxes. A careful evaluation of operating conditions (see notes in bearing selection matrix tables) shall be performed on a case-by-case basis.
	cylindrical roller bearing NJ-design	CRB	If this bearing is chosen for supporting axial load, flange strength of the inner ring shall be carefully evaluated in respect to fatigue bending and shock loads, as well as heat dissipation. Proper support of the flange by a well designed abutment is mandatory. NUP and NJ+HJ type may be used as locating bearings, if the necessary precautions are taken to prevent looseness between the bearing and the HJ ring.
	full-complement cylindrical roller bearing	fc CRB	Full complement bearings should only be selected where centrifugal forces do not contribute significantly to contact forces between rollers. If significant contact forces occur, special considerations need to be taken. If this bearing is chosen for supporting axial load, flange strength of the inner ring shall be carefully evaluated in respect to fatigue bending and shock loads, as well as heat dissipation. Proper support of the flange by a well designed abutment is mandatory.
	tapered roller bearing	TRB	Tapered roller bearings shall preferably be used as paired bearings. It shall be evaluated depending on the application whether the pairs need to be matched together to achieve the designated clearance in operation. Single-row TRB in cross-locating arrangement shall be used only where endplay can be controlled within acceptable limits under typical operation conditions.
	deep-groove ball bearing	BB	The suitability of this type may be restricted by its limits on load carrying capacity.
	angular contact ball bearing	ACBB	The suitability of this type may be restricted by its limits on load carrying capacity.
	double-row full-complement cylindrical roller bearing	dr fc CRB	Full complement bearings should only be selected where centrifugal forces do not contribute significantly to contact forces between rollers. If significant contact forces occur, special considerations need to be taken. If this bearing is chosen for supporting axial load, flange strength of the inner ring shall be carefully evaluated in respect to fatigue bending and shock loads, as well as heat dissipation. Proper support of the flange by a well designed abutment is mandatory.

Bearing recommendations are valid assuming:

- the bearing is appropriately sized in accordance with 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.

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Bearing types shown in table 8 support radial and axial loads.

5.3.2 Bearing arrangement

Except for sun pinions, pinions should be mounted between bearings. Overhung pinions should not be used. Sun pinions should be designed without bearings to achieve load sharing between planet gears.

Bearing types shown in table 9 support only radial loads. They may be used as a non-locating bearing or in combination with an additional bearing to support the axial load. Outer and inner rings shall be axially fixed in their journals.

Bearing types shown in table 10 support only axial loads. They shall be used only in combination with an additional bearing to support the radial load. The additional bearing should be a pure radial bearing to avoid axial load sharing. It shall be ensured that the axial bearing itself does not share any radial load, for example, by providing clearance in the radial direction at the outer ring. Appropriate means shall be taken to prevent the outer ring from spinning, for example, by a locking pin.

Table 9 - Bearings for pure radial load

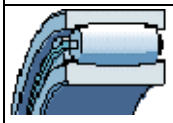
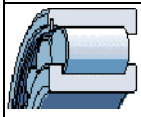
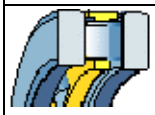
Symbol	Type	Abbreviation	General comments
	cylindrical roller bearing NU-design	CRB NU	This type has an outer ring with two flanges and inner ring without flanges. The outer ring can retain a small amount of oil to avoid dry startup.
	cylindrical roller bearing N-design	CRB N	N-designs may be considered on rotating shaft or rotating outer ring to minimize rollers skidding at the inner ring contact under low load running conditions. When this standard was developed, there was little documented experience for N-designs in wind turbine gearboxes.
	toroidal bearing	TORB	Axial displacement of the inner ring relative to the outer ring affects the radial clearance of the bearing. This effect shall be considered when determining the gear mesh alignment.
	full-complement cylindrical roller bearing	fc CRB	Full-complement bearings should only be selected where centrifugal forces do not contribute significantly to contact forces between rollers. If significant contact forces occur, special considerations shall be taken.

Table 10 - Bearings for pure axial loads

Symbol	Type	Abbreviation	General comments
	four-point contact ball bearing	4PCBB	Alternating axial loads may cause oscillating contact points and excessive cage wear. Application of a 4PCBB should therefore be avoided in applications with loads alternating at high frequency.
	spherical roller thrust bearing	SRTB	This bearing shall have sufficient axial load under all operating conditions - refer to bearing manufacturer's recommendations.
	taper roller bearing	TRB	This bearing shall have sufficient axial load under all operating conditions - refer to bearing manufacturer's recommendations.
	cylindrical roller thrust bearing	CRTB	This bearing shall have sufficient axial load under all operating conditions - refer to bearing manufacturer's recommendations. A CRTB has unfavorable sliding and requires special properties of the lubricant. The bearing selected should have a small mean diameter and a small sectional height to minimize sliding.

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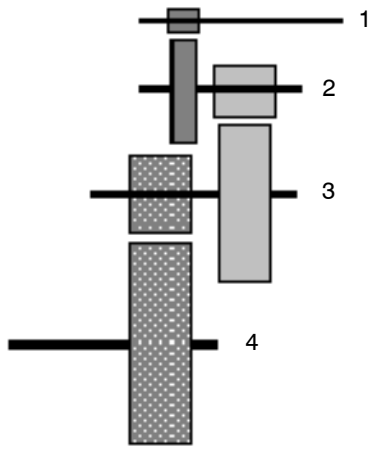
Each shaft should be supported by one of the arrangements shown in the selection matrices tables 12 through 16. For interpretation of the matrices refer to 5.3.2.2 and table 11.

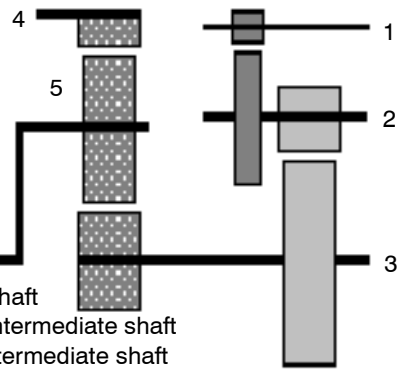
5.3.2.1 Terms and definitions

shaft designations: Figures 1 and 2 show the designation for 3-stage parallel shaft gearboxes and 3-stage planet/helical hybrids. In 4-stage gearboxes, the intermediate shafts shall be called "low speed intermediate shaft", "medium speed intermediate shaft", and "high speed intermediate shaft". Depending on the speed range, tables 13 or 14 may apply for selection of bearings for the medium speed intermediate shaft. Gearboxes with more than one planet stage are not covered by these recommendations.



Key
1 high speed shaft
2 high speed intermediate shaft
3 low speed intermediate shaft
4 low speed shaft

Figure 1 - 3-stage parallel shaft gearbox



Key
1 high speed shaft
2 high speed intermediate shaft
3 low speed intermediate shaft
4 internal ring gear
5 planet
6 low speed shaft

Figure 2 - 3-stage planet/helical hybrid

bearing arrangements:

- paired bearings: Two bearings of the same type at the same location so their radial capacities complement and their axial capacities are opposite. Examples are two TRB or two ACBB in face-to-face or back-to-back arrangement.
- combined bearings: Two bearings of different types at the same location to fulfil separate functions. An example is 4PCBB with CRB-NU.
- tandem bearings: Two bearings of the same type at the same location so their radial and axial capacities complement.
- double row bearings: Paired bearings with common inner rings, outer rings, or both. For example, fc CRB are available as double-row bearings.

bearing function:

- locating: A bearing supporting axial forces in both directions. Another common term for this arrangement is "fixed bearing". The locating function can be achieved by one bearing suitable for combined loads (see table 8) or by combined bearings.
- non-locating: A bearing supporting only radial load. Another common term for this arrangement is "floating bearing". The non-locating function can be achieved by a bearing suitable for pure radial loads (see table 9) or a bearing suitable for combined loads (see table 8) with one ring floating on its shaft or in its bore. The non-locating bearing shall be free to absorb any axial growth due to thermal change. Bearings that shift internally (like CRB) are preferred because their rings can have tight fits.
- cross-locating: A bearing arrangement where either bearing supports the axial load depending on the direction of the axial load. The bearing and shaft assembly shall have sufficient axial clearance to absorb any axial growth due to thermal change. Bearings that shift internally (like CRB) are preferred because their rings can have tight fits.

5.3.2.2 Bearing selection matrices

The following rules apply to tables 12 through 16:

- The symbol for a NJ-CRB is used for all caged CRB. However, NU- or N- bearings should be used in locations without axial load.
- Where one of the axial bearings (4PCBB, SRTB, CRTB) is recommended as the locating bearing, this shall implicitly mean this axial bearing combined with a radial bearing. The radial bearing shall be selected from the "non-locating" row, preferably one of the true radial bearings.
- Where TRB or ACBB are recommended as locating bearings, this always refers to paired

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bearings arranged face-to-face or back-to-back.

5.3.3 Bearing shaft and housing fits

Bearings in wind turbine gearboxes require very heavy duty fitting practices as defined by bearing manufacturers. Tight fits are necessary to prevent damage to the bearing or housing. One bearing ring shall be tight; the other ring should be tight or adequate means should be provided to prevent spinning of the inner or outer bearing rings. Planet bearing outer rings should be fitted to planet bores with a tolerance of at least R6 for metric bearings as a general guideline. Other fits may be required depending on operating conditions and planet wheel design. Tolerances selected should be reviewed with the bearing manufacturer. Planet bearing inner rings should be clamped to avoid rotation, except when spherical roller bearings are used.

5.3.4 Bearing cages

Bearing cages that both guide and separate the rollers (such as SRB) shall be steel or brass.

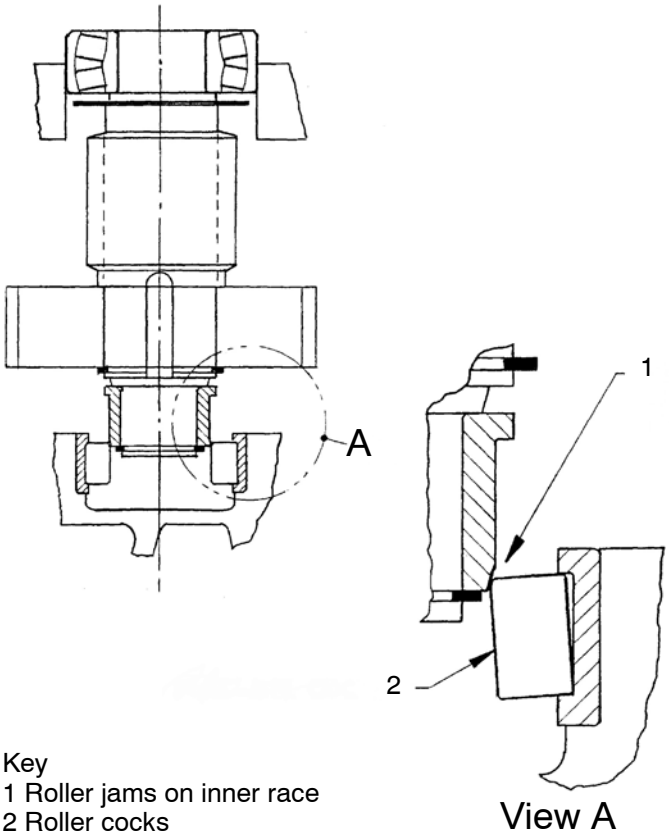
For bearing cages that only separate the rollers, other cage material may be considered, if they are proven to be resistant to embedding of hard particles and resistant to ageing due to operating temperature and lubricant for the design life. The suitability of other cage materials shall be agreed to between the purchaser and the gearbox manufacturer and shall be based on thorough evaluation of field experience.

5.3.5 Bearing internal clearance

Internal clearance shall be designed to accommodate heavy interference fits and temperature differences (see table 5). The appropriate selection shall be checked at worst tolerance combination at startup condition. Radial clearance shall be controlled to limit misalignment of the gear meshes. This is especially important for the low speed shaft of gearboxes that support rotor forces.

5.3.6 Bearing assembly

Bearings can be easily damaged during installation. For example, figure 3 shows how blind installation of cylindrical roller bearings can be especially risky. Therefore, the gearbox manufacturer should avoid blind assembly. Bearings shall be installed using appropriate tools and techniques that minimize the risk of damage.



Key
1 Roller jams on inner race
2 Roller cocks

Figure 3 - Bearing assembly

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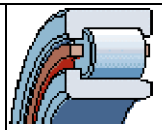
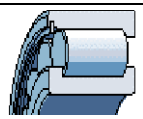
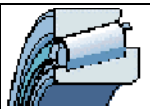
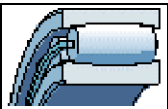
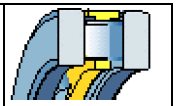
Table 11 - Bearing selection matrix - legend to symbols

Symbol	Description	General comments
●	suitable	This bearing type has proven suitable for wind turbine gearboxes.
◐	suitable with restrictions	This bearing may be suitable if certain restrictions and additional precautions are complied with. See individual tables for details.
○	not experienced	This bearing either has not been used in wind turbine applications or has insufficient field experience in wind turbines to confirm acceptable performance. Any use of this type shall be preceded by detailed theoretical analysis, approval from the bearing manufacturer, and appropriate testing for the specific application.
⊖	not suitable	This bearing type either is not to be used for the specific conditions listed or has proven unsuitable for wind turbine gearboxes.

Table 12 - Bearing selection matrix for the low speed shaft/planet carrier

Type										
	SRB	CRB	fc CRB	TRB	BB	4PCBB	ACBB	SRTB	TORB	CRTB
locating	◐ 6)	○	○ 5)	◐ 3)	●	○	○	○	⊖	○
non-locating	◐ 1)	●	◐ 4)	○	◐ 1)	⊖	⊖	⊖	○	⊖
cross-locating	◐ 1), 6)	●	●	◐ 2)	◐ 1)	⊖	⊖	⊖	⊖	⊖

NOTES:

1) When this bearing type is used as non-locating bearing or in a cross-locating arrangement, the outer ring fit shall allow accommodation of thermal expansion by movement of the ring in the bore. The outer ring shall be prevented from spinning.

2) Single-row TRB in cross-locating arrangement shall only be used where endplay can be controlled within acceptable limits under typical operating conditions.

3) Performance of paired double-row TRB depends on appropriate internal clearance.

4) The radial lip shall be free to allow for axial movement. The outer ring shall be prevented from spinning.

5) This is intended for bearings that can take thrust in both directions.

6) Influences of variation of loads and shaft movement (amplitude and frequency) within the bearing internal clearance should be carefully analyzed.

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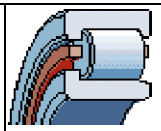
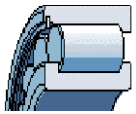
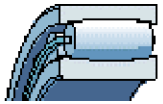
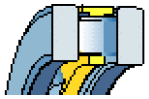
Table 13 - Bearing selection matrix for the low speed intermediate shaft

Type										
	SRB	**CRB**	**fc CRB**	**TRB**	**BB**	**4PCBB**	**ACBB**	**SRTB**	**TORB**	**CRTB**
locating	◐ [6]	○	○	◐ [3]	●	○	○	●	⊖	◐ [5]
non-locating	◐ [1]	●	◐ [4]	○	◐ [1]	⊖	⊖	⊖	●	⊖
cross-locating	◐ [1], [6]	●	●	◐ [2]	◐ [1]	⊖	⊖	⊖	⊖	⊖

NOTES:

[1] When this bearing type is used as non-locating bearing or in a cross-locating arrangement, the outer ring fit shall allow accommodation of thermal expansion by movement of the ring in the bore. The outer ring shall be prevented from spinning.

[2] Single-row TRB in cross-locating arrangement shall only be used where endplay can be controlled within acceptable limits under typical operating conditions.

[3] Performance of paired double-row TRB depends on appropriate internal clearance.

[4] The radial lip shall be free to allow for axial movement. The outer ring shall be prevented from spinning.

[5] Sliding motion occurs in roller contacts. This bearing shall be spring loaded.

[6] Influences of variation of loads and shaft movement (amplitude and frequency) within the bearing internal clearance should be carefully analyzed.

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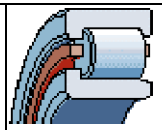
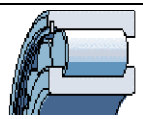
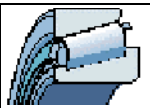
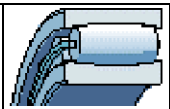
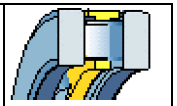
Table 14 - Bearing selection matrix for the high speed intermediate shaft

Type	SRB	 CRB	 fc CRB	 TRB	BB	4PCBB	ACBB	SRTB	 TORB	 CRTB
locating	◐ 4)	◐ 5)	⊖	◐ 3)	○	●	○	○	⊖	○
non-locating	◐ 1)	●	⊖	○	○	⊖	⊖	⊖	◐ 8)	⊖
cross-locating	◐ 1) 4)	◐ 6)	⊖	◐ 2) 7)	○	⊖	⊖	⊖	⊖	⊖

NOTES:

1) When this bearing type is used as non-locating bearing or in a cross-locating arrangement, the outer ring fit shall allow accommodation of thermal expansion by movement of the ring in the bore. The outer ring shall be prevented from spinning.

2) Single-row TRB in cross-locating arrangement shall only be used where endplay can be controlled within acceptable limits under typical operating conditions.

3) Performance of paired double-row TRB depends on appropriate internal clearance.

4) Influence of variation of gear loads generated by torque and/or axial movement of the high speed intermediate shaft (amplitude and frequency) within the bearing internal clearance should be carefully analyzed.

5) If NUP or NJ+HJ type are used as locating bearings, precautions shall be taken to prevent looseness between the bearing and the HJ ring.

6) This bearing requires consideration for cooling requirements due to its methods of carrying axial load. The ratio of axial to radial load shall also be reviewed.

7) This design is successful when used with short shafts that do not have large thermal expansions and housings with small elastic deformations.

8) There is currently short term experience with this bearing type in wind turbine service.

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