BS ISO 81400-4:2005

Wind turbines —

Part 4 : Design and specification of gearboxes

ICS 21.200; 27.180

British Standards

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.

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Contents

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.

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

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.

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.

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.

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).

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.

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.

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.

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 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 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 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:

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

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.

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 $10 \times L_{10r}$ (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 $10 \times L_{10r}$. 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.

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

$$L_{\text{adv}} = \frac{\Sigma q_{\text{i}}}{\Sigma \frac{q_{\text{i}}}{L_{\text{adv, i}}}} \tag{5}$$

where

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

q_{i} is time share on the ith load level;

$L_{\text{adv, i}}$ 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 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;

- 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, 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.

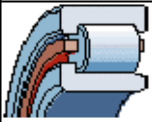
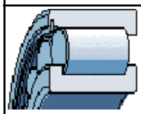
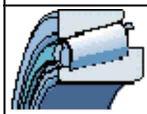
5.3 Bearings

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.

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.

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

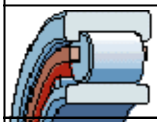
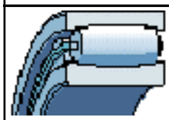
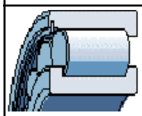
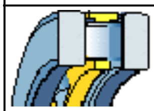
Symbol	Type	Abbreviation	General comments
	cylindrical roller bearing NU-design	CRB NU	This type has an outer ring with two flanges and inner ring without flanges. The outer ring can retain a small amount of oil to avoid dry startup.
	cylindrical roller bearing N-design	CRB N	N-designs may be considered 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.

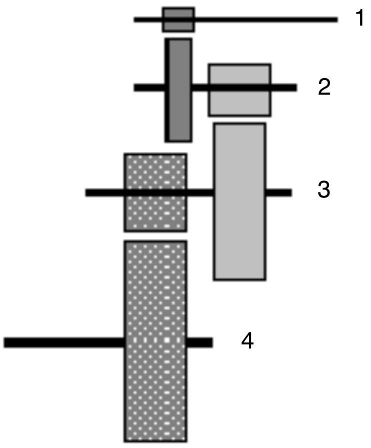
Table 10 - Bearings for pure axial loads

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.

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.

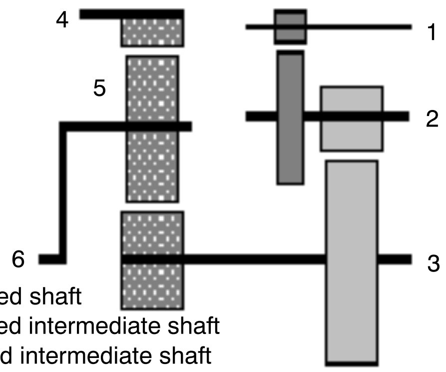
5.3.2.1 Terms and definitions

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

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.

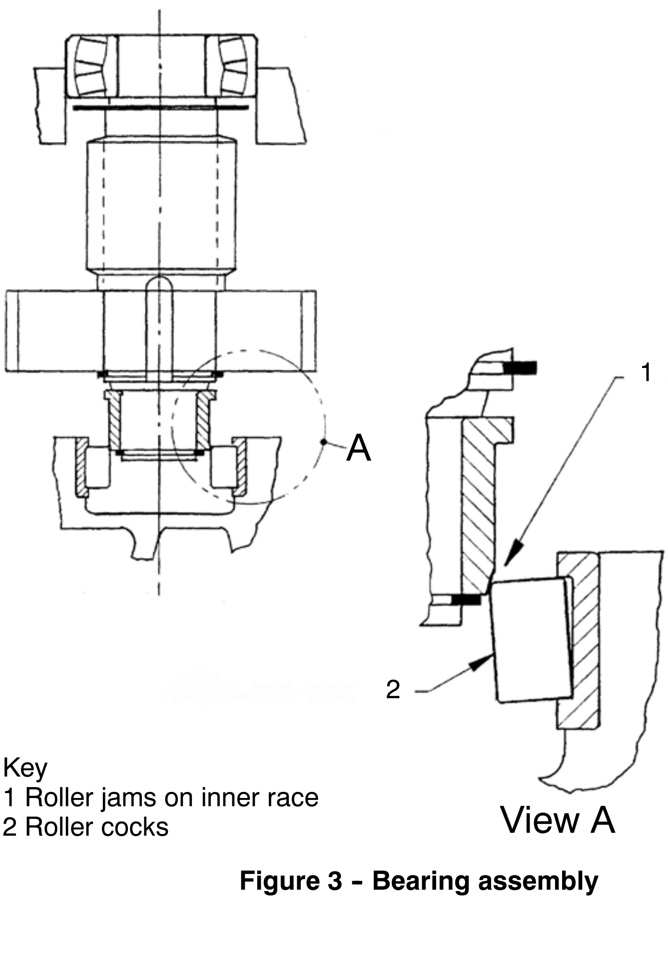
For bearing cages that only separate the rollers, other cage material may be considered, if they are proven to be resistant to embedding of hard particles and resistant to ageing due to operating temperature and lubricant for the design life. The suitability of other cage materials shall be agreed to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

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.

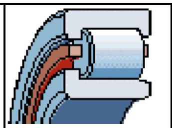
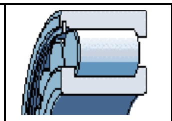
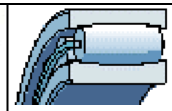
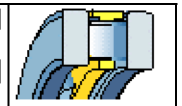
3) Performance of paired double-row TRB depends on appropriate internal clearance.

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.

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.

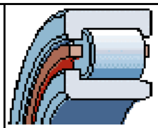
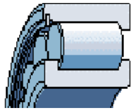
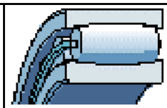
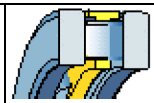
3) Performance of paired double-row TRB depends on appropriate internal clearance.

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.

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.

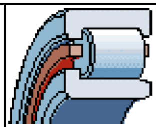
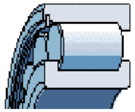
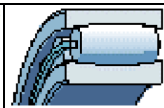
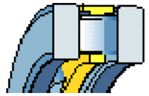
5) If NUP or NJ+HJ type are used as locating bearings, precautions shall be taken to prevent looseness between the bearing and the HJ ring.

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.

Table 15 - Bearing selection matrix for the high speed shaft

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

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) High axial to radial load ratio loading conditions as well as influence of variation of gear loads generated by torque, or axial movement, or both of the high speed shaft (amplitude and frequency) within the bearing internal clearance should be carefully analyzed.

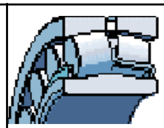
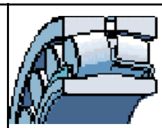
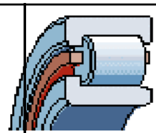
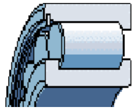
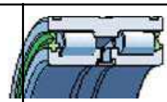
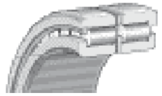
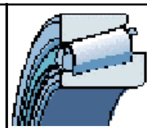
5) This design is successful when used with short shafts that do not have large thermal expansions and housings with small elastic deformations.

6) Review cooling requirements when using this bearing type.

7) Special attention to alternating axial loads, for example from couplings, is required when using this bearing type.

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

Table 16 - Bearing selection matrix for the planet wheel

Type							
	SRB	**SRB**	**CRB**	**fc CRB**	**dr fc CRB**	**dr fc CRB**	**TRB**
arrange-ment	single	two bearings	2 bearings cross-locate	2 bearings cross-locate	single	2 bearings	2 bearings cross-locate
spur gears	◐ 6)	◐ 1)	●	◐ 2)	◐ 2)	◐ 2)	◐ 4)
single helical gears 5)	⊖	◐ 1)	●	◐ 2)	◐ 2), 6)	◐ 2), 3)	○

NOTES:

1) Two SRB may be used when appropriate fit allows sufficient axial movement of the inner ring for load sharing. The risk of wear on the inner rings and their shaft shall be evaluated. When closely spaced, total capacity may be less than the two bearing single capacities (see ISO 281 for details).

2) Full-complement bearings shall be used only for low speed stages where centrifugal forces do not contribute to contact forces between rollers.

3) The helix angle generates a tilt moment on the planet that causes uneven load sharing between the bearing rows. The elastic deflection of the planet wheel along the width can also affect load sharing. These effects shall be considered in detail.

4) Cross-locating TRB in planet wheels should be assembled with zero clearance or light pre-load. This enables a stiff support for the mesh. Consequently, planet carrier deflection (due to production deviations, carrier deflection or clearance in carrier bearings) will lead to mesh misalignment that shall be compensated for in the design.

5) Extensive field experience with helical planet stages began in the year 2000.

NOTE: The raceway of cylindrical, tapered, and spherical roller bearings may be integrated into the planet bore. This allows for compact designs and avoids the risk of spinning outer rings. The integration does define additional requirements to the metallurgy, heat treatment, and production processes of the gears. This option was under evaluation and field test when the standard was issued, but no experience was available.

6) Rotating outer ring in combination with misaligned axis of inner ring relative to the outer ring causes dynamic misalignment with axial sliding of rollers against the outer ring.

5.4 Shafts, keys and splines

There are many requirements for shafts, keys and splines that may be specified. The following items are recommended as a minimum.

5.4.1 Shaft material

All shafts shall have sufficient surface hardness and should be made from alloy steels with sufficient hardenability to obtain a microstructure that has strength and fracture toughness meeting the requirements of the application (see 5.1.4).

5.4.2 Lifting holes

All shaft ends shall have threaded holes designed to accept eye bolts for lifting.

5.4.3 Keyless fits

Interference fits without keys (keyless fits) shall be designed per ANSI/AGMA 6001–D97 or DIN 7190.

5.4.4 Interference fits with keys

Interference fits with keys shall be designed to transmit the maximum reversing torque only by interference. No benefit from keys shall be considered when calculating the reverse torque capacity.

5.4.5 Keys

All keys should be designed in accordance with ANSI/AGMA 6001-D97 or ISO/TR 13593. Key slots should not extend into the bearing journals. Intersection of key slots with diameter changes should be avoided. Edges of key slots should be uniformly deburred or chamfered, free of nicks, gouges or any sharp transitions that may act as stress risers.

5.4.6 Key material

All keys shall be made from steel with sufficient hardenability to obtain a microstructure that has strength and fracture toughness meeting the requirements of the application.

5.4.7 Key hardness

All keys should have a surface hardness that is consistent with the strength requirements of the application as determined from calculations in accordance with ANSI/AGMA 6001-D97 or ISO/TR 13593.

5.4.8 Key geometry and shaft fit

Key, shaft and hub shall be designed in accordance with ISO R773 or DIN 6885-2. All keys shall be fitted to their shaft with an interference fit.

5.4.9 Flexible splines

Flexible splines that transmit significant torque shall be designed to prevent fretting corrosion. Refer to the internal couplings clause of ANSI/AGMA 6123–A88 for additional information on this topic. External and internal teeth should be case hardened, preferably by nitriding. Lubrication should be adequate to prevent fretting corrosion. Pressure fed lubrication is preferred. The oil flow through the connection shall be directed to flush out all debris and return channels shall direct oil back into the sump.

5.5 Housings

There are many requirements for the housing that may be specified. The following items are recommended as a minimum.

The location of gearbox components, particularly those that require maintenance, should be agreed to by the gearbox manufacturer and the purchaser to assure safe, accessible maintenance (see annex E for details). An outline dimension drawing of the installation location should be provided by the purchaser that shows space constraints and location of the components.

Gearboxes are subject to being climbed over, stepped on, or used as work surfaces. Therefore, components should be placed or shielded so that they are not damaged during maintenance activities.

Two piece housings, split on a plane through the bearing bores, are preferred for parallel shaft designs. One piece housings make assembly of gears and bearings more difficult, and risk damaging bearings. However, they may allow larger bearing sizes.

5.5.1 Housing material

All housings for integrated systems should be cast ductile iron, cast steel or fabricated steel. Modular systems may have cast iron housings.

5.5.2 Housing distortion

The housing should be designed and constructed to prevent harmful distortions resulting from thermal deformation, mechanical deformation or both.

5.5.3 Housing accuracy

The housing bores shall be machined to an accuracy that ensures all gearsets and all bearings contact correctly. Bearing journal cylindrical form and form tolerance of abutment shoulders shall be in accordance with bearing manufacturer's recommendations. The tolerances for parallelism of shafts shall match the shaft alignment assumed in the determination of $K_{H\beta}$ (see 5.1.1.2). For further information see ISO/TR 10064-3.

5.5.4 Inspection covers

Removable inspection covers with reusable gaskets should be provided for field inspection of the face width of all parallel shaft gear meshes. The inspection openings should be a minimum of 90% of the gear face width(s). They should be large enough to allow easy access for direct visual inspection of the gear teeth and lubricant as well as the measurement of bearing axial or radial endplay.

The inspection covers, especially those on top of the gearbox, should be easy to clean of any debris buildup prior to their removal. They should be light and small enough to be easily handled within the confines of the turbine nacelle. They may be of one or multi-piece construction. They should have raised mounting faces on the housing to minimize the potential of contaminants.

Threaded holes should be provided to permit separating inspection covers from the housing with jack screws. Tapped holes for securing the covers to the housing should not penetrate into the housing interior.

5.5.5 Epicyclic gear inspection

Adequate access to sun, planet and internal gears shall be provided. Inspection ports and planet carriers shall be designed to allow visual inspection of the sun pinions. Ports shall be provided for borescope inspection if necessary. Removable sun pinions are preferred for inspection or replacement on site.

5.5.6 Bore covers

All bore covers, bearing retainer caps or seal retainer caps should be sealed with O-rings or suitable sealing compound and should be provided with threaded holes to permit removal with jack screws. Ports should be provided to access the tapped end of the gear shafts to allow end play measurements.

5.5.7 Housing joint

If the housing has a split plane for gear removal, it should employ a flat metal-to-metal joint that is maintained oil tight with O-rings or suitable sealing compound. O-rings and sealing compounds that are compatible with the lubricant should be used. Apply all sealing compounds in accordance with the manufacturer's recommendations. Do not apply excessive amounts of sealing compounds. In the uncured state, these materials may dissolve in the lubricant and lead to residue formation on critical surfaces during gearbox operation. Split plane housings shall have positive locking devices such as dowel pins. Bearings shall not be used for centering housing splits.

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

6 Lubrication

The selection, installation and monitoring of the correct lubricant are essential for optimum service life and performance of the WTGS. The specification of performance requirements for the lubricant shall be the joint responsibility of the purchaser and gearbox manufacturer. The selection of a specific brand or type of lubricant shall also be approved by the purchaser and gearbox manufacturer. The bearing manufacturer shall also be consulted by the gearbox manufacturer during this approval process. Minimum physical and performance specifications for various types of lubricants are provided in ANSI/AGMA 9005-E02. Additional information related to the lubrication of WTGSs can be found in annex F. In addition to the gears and bearings, there are many other components that come in contact with the gearbox lubricant. The requirements of components such as seals, paints, pumps, heat exchangers, and filters should be considered when selecting the appropriate gearbox lubricant. Therefore, many requirements of a WTGS lubrication system should be specified. As a minimum these include the type of lubricant, lubricant viscosity, the method of lubrication, operating conditions and system maintenance.

6.1 Type of lubricant

WTGS gears operate at low to moderate pitch line velocity with high to very high contact loads. These conditions require the use of lubricants fortified with antiscuff additives and of the highest practical viscosity. The base fluids of these lubricants should be highly refined mineral oils, full synthetic fluids, or semi-synthetic blends (mixtures of highly refined mineral oils and synthetic fluids). The choice of lubricant depends on many factors including viscosity, viscosity index, pour point, additives, and overall lubrication costs. Site specific operating conditions, WTGS performance, and serviceability influence the selection of the most cost effective gearbox lubricant. It is imperative to maintain the proper lubricant viscosity at the gearbox operating temperature. Therefore, parameters such as cold start and operating temperature within the nacelle should be closely monitored. Lubricant type and viscosity should be appropriate for the operating conditions. Micropitting resistance of a lubricant is very important in this application and should be tested. See annex F for additional information.

6.2 Lubricant viscosity

Viscosity is the most important physical property of a lubricant. The viscosity, at operating conditions of temperature, load, and velocity, has direct impact on gearbox performance and durability. The correct viscosity at cold startup is important to achieve adequate lubricant flow to all critical surfaces without channeling or creating excessive drag. The correct viscosity at operating temperature is required to minimize metal to metal friction and wear without contributing to excess parasitic losses such as churning by the gear, or fluid friction drag in bearings.

Maintaining proper viscosity over the entire operating temperature range of the gearbox also minimizes the potential for foaming and air entrainment. Excessive parasitic losses elevate the operating temperature of the gearbox, which increases the oxidation rate of the lubricating oil. The useful service life of a lubricant is reduced when its oxidation rate is increased. Byproducts of oxidation include residue-forming material such as varnish and sludge that can plug filters and small oil passages such as oil spray nozzles, as well as deposit on critical surfaces.

Selecting the correct viscosity grade of the lubricating oil for a gearbox should be based on operating, not startup conditions. If the correct viscosity at operating temperature results in excessive viscosity during cold startup, several options are available. Lubricant manufacturers often can provide synthetic, semi-synthetic, or mineral oil based products, with enhanced low temperature fluid properties, such as low pour point, very high viscosity index, and low dynamic viscosity at low temperatures. Another option is to incorporate a low-power-density heater in the gearbox sump or a space heater within the nacelle to warm the lubricating oil to achieve the proper startup viscosity.

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

6.3 Method of lubrication

The required application and method of lubrication may be specified. Depending on configuration, features such as pump location, type of tubing, hoses, fittings, type of filter element, filter change interval and quantity of oil in reservoirs may be specified.

6.3.1 Splash lubrication

Splash lubrication is the simplest system. The low speed gear should dip into the oil bath for at least twice the tooth depth to provide adequate splash for gears and bearings. The oil level should be designed to minimize churning while providing adequate lubrication to all bearings and gears. The gear housing should have troughs to capture the oil flowing down the housing walls, and channels to the bearings.

Splash systems without an offline filtration system to control contamination and prevent the distribution of particles to critical gear and bearing surfaces are not recommended. The offline filtration system shall be designed to maintain an oil cleanliness level one class better than the assumption made in bearing life calculations (see 5.1.3.5).

Splash lubrication works well when a WTGS control operating strategy uses rotor idling, so that lubricant is always present in rolling and sliding interfaces of gears and bearings. When a parking brake is set during cut-out or before cut-in, splash lubrication may not prevent direct metal-to-metal contact in gears and bearings. This is especially true with a high speed shaft parking brake.

6.3.2 Pressure fed lubrication

Gearboxes 500 kW and above shall be lubricated by an oil circulation system capable of maintaining an oil cleanliness level as specified in table 17 by inline filters, offline filters or a combination of both. Pressure fed systems can also have a heat exchanger to cool the oil. These systems can ensure adequate lubrication of all rotating elements and prolong the life of the lubricant and components. To ensure adequate lubrication and control lubricant temperature, the system must be properly designed considering viscosity, flow rate, feed pressure, and the size, number and placement of the jets. All bearings except those that submerge into the sump operating oil level shall be fed by the circulation system. The WTGS control strategy can, during idling or parking, activate a pressure fed system periodically to minimize damage caused by lack of lubricant.

Table 17 - Lubricant cleanliness

Source of oil sample	Required cleanliness per ISO 4406
Oil added into gearbox at any location	–/14/11
Bulk oil from gearbox after factory test at the gearbox manufacturer's facility	–/15/12
Bulk oil from gearbox after having been in service 24 to 72 hours after commissioning of the WTGS (pressure fed systems only)	-/15/12
Bulk oil from gearbox sampled per the operating and maintenance manual (pressure fed systems only) (see 6.7)	–/16/13

The pressure level in the oil distribution system and pressure drop at the orifices shall be selected as low as possible to reduce entrained air. For the same reason, hard oil spray against rotating parts shall be avoided. Pressure fed lubrication may not require oil sprayed on the gears and bearings by spray nozzles. Pressure spray lubrication should be required at pitch line velocities greater than 25 meters per second.

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

6.3.3 Combined lubrication systems

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

6.3.4 Oil scrapers

Lubricant can be transferred from gears by scrapers that either do not contact to avoid wear, or are elastic, or spring loaded, to accommodate the motion of the gears. Scrapers provide a flow of oil to troughs or channels that distribute it to gears and bearings. Care must be taken to ensure that scraper fasteners cannot loosen.

6.4 Operating temperature

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

6.4.1 Bulk oil temperature

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

6.4.2 Gear mesh temperature

This is the temperature of the oil measured as it exits the gear tooth mesh. This is a close approximation of the gear tooth surface temperature that is important for scuffing resistance calculations. The accuracy of the gear tooth surface temperature measurement is increased by making the measurement as close to the mesh exit as possible.

6.4.3 Bearing oil temperature

This is the temperature of the oil as measured in the vicinity of the rolling element bearing. The accuracy of the bearing oil temperature measurement is increased by making the measurement as close to the rolling elements as possible. This is different from the bearing temperature that is measured on the outer ring/cup of the bearing.

6.5 Oil quantity

Minimum quantity of oil in the lubrication system should be:

$$Q_{ty} = 0.15\,P_t + 20 \tag{6}$$

where

Q_{ty} is recommended oil quantity, liters;

P_t is rated power of wind turbine, kW.

These recommendations are based on experience with typical multistage gearboxes where the gear housing forms the oil reservoir. If the design is not a multistage gearbox or a separate oil tank is used, the above information may not apply.

6.6 Temperature control

Gearbox operating temperature should be controlled through all phases of operation. If necessary, heaters and coolers should be used to control gearbox temperature. Controls should be set as specified in 6.6.1 and 6.6.2.

6.6.1 Sump temperature

Maximum sump temperature above ambient and maximum absolute sump temperature shall be specified. The specified limits shall correspond to the values used in rating calculations for gears and bearings (see 5.1.3.3). As an absolute maximum, controls shall be set to shutdown the turbine when the sump temperature exceeds 85°C for ten minutes within any continuous sixty minutes of operation. Sump temperature should be maintained at least 5°C above the lubricant pour point so lubricant will circulate freely prior to startup. A heat source may be needed to achieve this. The gearbox shall be equipped with appropriate monitoring devices to ensure the lubricant system provides adequate flow during periods of cold weather to critical components such as gears, bearings, splines, filtration devices, and external oil reservoirs. To meet these requirements, the overall gearbox and any external systems may need to be heated. This could be done by a number of methods including:

- air heating in the nacelle;
- heat exchangers in a pressurized lubrication system;
- using turbine idling with controlled speed to raise the oil temperature in the bearings and gear meshes. Make sure that upper bearings have sufficient lubrication at feathered controlled speed;
- wrapping external lubrication lines with resistive heating devices;
- continual pumped circulation of oil to maintain fluidity and utilize viscous heating of the lubricant.

6.6.2 Bearing temperature

Maximum bearing temperature above ambient and above the oil sump temperature, as well as maximum absolute bearing temperature, shall be specified. The specified limits shall correspond to the values used in the calculation of bearing rating life (see 5.1.3.3). As absolute maximum, controls shall be set to shutdown the turbine when the bearing outer ring temperature exceeds 105°C for ten minutes within any continuous sixty minutes of operation. Maximum permissible continuous bearing temperature measured at the bearing outside diameter shall not exceed 95°C. This temperature limit may need to be reduced for some lubricants.

6.7 Lubricant condition monitoring

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

- oil cleanliness;
- viscosity;
- water content;
- wear metals;
- measure of oil oxidation.

Refer to annex F for further guidance. The wind turbine manufacturer is responsible for preparing the operation and maintenance manual in accordance with the requirements specified by the gearbox manufacturer.

The first sample should be drawn from the gearbox within 72 hours of commissioning. The second sample should be drawn from the gearbox not later than 1000 operating hours after commissioning of the wind turbine. Six months has proven to be a suitable interval for subsequent samples.

6.8 Lubricant cleanliness

For maximum gear and bearing life, the lubricant must be clean. Guidelines for maintaining cleanliness are:

- filter new lubricant before adding to gear boxes;
- determine oil cleanliness after factory test to determine assembly cleanliness (see table 17);
- filter oil in service and maintain an oil cleanliness level as specified in table 17;
- monitor during service to detect contamination or other adverse changes to lubricant according to 6.7.

Table 17 and annex F give lubricant cleanliness criteria.

The gearbox manufacturer shall demonstrate by test results that the test oil used in production/acceptance testing of every gearbox meets the cleanliness shown in table 17. The purchaser shall demonstrate by test results that the oil in the gearbox at commissioning meets the cleanliness shown in table 17. Additionally, during the warranty period, the purchaser shall document by test results that the oil has been maintained at the cleanliness shown in table 17 in accordance with the requirements stated in the operating and maintenance manual. The user should continue to monitor and document the cleanliness of the oil in the same manner throughout the service life of the gearbox.

6.9 Lubricant filter(s)

Filtration devices shall remove ingested atmospheric and internally generated debris faster than they can accumulate within the gearbox. The filtration capabilities of the lubrication system shall maintain the bulk oil at a minimum cleanliness as specified in 6.8. Responsibility for determining the appropriate components, dimensional sizes of the filter elements, the nominal sizes of filter media pores, and the efficiency ratings of the filter media should be shared by the purchaser, gearbox manufacturer, and filter manufacturer. The purchaser shall provide the gearbox manufacturer with environmental conditions expected at the turbine installation site (see 4.6 and E.1.3). Refer to F.4 for further information on filtration.

6.10 Ports

Requirements for the type and size of ports on the gearbox shall be agreed to between the purchaser and the gearbox manufacturer. The type of port may include an oil drain, pressure feed, suction strainer, oil level, oil pressure, oil filler, oil sampling, or breather. The following items are recommended, depending on configuration.

6.10.1 Drain and fill plugs

Drain and fill plugs should be placed where lines are easily attached. The drain should be at the bottom of a sloped sump so that the oil can be completely drained. The drain opening should be a minimum of 25 millimeters to allow the oil to drain in a reasonable time. A ball valve should be provided for draining the lubricant in the field. A plug should be provided to seal the drain valve.

6.10.2 Pressurized ports

If an oil circulation system is used, all external ports for suction and return should have O-ring type seals. Drawings and manufacturer's manuals shall clearly identify the connection points for the oil circulation and offline filter system.

6.10.3 Non-pressurized ports

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

6.11 Oil level indicator

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

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

6.12 Magnetic plugs

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

6.13 Breather

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

6.14 Oil sampling ports

Splash lubrication gearboxes should incorporate a threaded port on each side of the housing at an elevation equal to the midpoint of the operating oil level. Each port should be in a location of free lubricant circulation within the sump. Pressure fed lubrication systems should incorporate two oil sampling ports. One should be between the pump and the filter, and the other directly after the filter. See F.5.1.1 for sampling techniques.

7 Other important items

7.1 Seals

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

7.2 Interfaces

Requirements for all the interfaces to the gearbox shall be specified. These may include a wide range of features depending on gearbox configuration. The following items should be considered depending on the application.

7.2.1 Low speed shaft

Features such as shaft size, length, extension, tolerance, strength and loading should be specified.

7.2.2 High speed shaft(s)

Features such as shaft size, length, extension, tolerance, strength and loading should be specified.

7.2.3 Mounting

Depending on configuration, features such as service access, geometry, size, bolt patterns, tolerance, allowable deflection and loading shall be specified.

7.2.4 Torque arm

If the gearbox is a shaft mount, the housing shall have mounting lugs for the torque arm. Mounting lugs should be positioned to minimize housing movement relative to the rotor shaft during normal operation, feathering or braking. Features such as position, size, geometry, tolerance, and loading shall be specified. If elastomeric bushings are used, their elastic properties shall be specified.

7.2.5 Generator

If a generator is mounted directly to the gearbox housing, features such as service access, geometry, bolt patterns, tolerance, generator overall size, weight, overhang, rotating inertia, type of coupling and loading shall be specified.

7.2.6 Pitch system

If an actuation device for the pitch system is mounted directly to the gearbox housing, features such as service access, geometry, bolt patterns, actuator overall size, weight, overhang and loading shall be specified.

7.2.7 Yaw system

If an actuation device for the yaw system is mounted directly to the gearbox housing, features such as service access, geometry, bolt patterns, actuator overall size, weight, overhang and loading shall be specified.

7.2.8 Lifting points

Features such as through holes in the housing or eye bolts should be specified to facilitate lifting, assembly and handling of the gearbox. Requirements such as load rating and surrounding geometry should be specified.

7.2.9 Brake

If a shaft brake is mounted on the housing, pads shall be provided for mounting the brake. Features such as geometry, mounting surfaces, bolt patterns, and loading shall be specified.

7.2.10 Sensors

For each of the monitoring sensors (for example; pressure, oil level and temperature), features such as service access, placement, geometry, tolerance, expected quantity, range and required electrical connectors shall be specified.

7.2.11 Safety systems

Depending on configuration, there may be several safety systems whose interface should be specified. These may include a rotor lock, a rotor brake, a yaw lock or others. For each system, features such as service access, geometry, bolt patterns, mounting surfaces, tolerances, and loading shall be specified.

7.2.12 Personnel

During periods of testing, maintenance and repair, technicians may be required to be adjacent to or on top of the gearbox. Depending on configuration, the interface with personnel may include ladders, safety clip points, shaft guards, heat shields or other protective features to prevent injury. Requirements for each of the interfaces should be specified.

7.2.13 Miscellaneous

Depending on configuration, there may be several other interfaces that should be specified. These may include work platforms, nacelle covers or electrical, hydraulic or pneumatic systems.

7.3 Hardware

All fasteners shall be metric grade 8.8 or better; grade 12.9 shall not be used under dynamic loading. Fastener size, tightening torque and engagement should be in accordance with ANSI/AGMA 6001–D97 or qualified by certified test. Wherever feasible, hardware shall be standardized to common sizes and finishes.

7.3.1 High strength hardware

High strength hardware is sometimes used where high loads are concentrated, such as yaw bearings.

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

Purchasing of high strength hardware should be source controlled. Substitutes or changes in the plating process should not be allowed. The hardware should be independently sampled and tested using a stress/strain device.

7.3.2 Internal fasteners

Staking is not permitted. Hardware such as set screws, bolts, nuts, pins, and fittings should not be used inside the housing unless they are secured in a manner approved by the purchaser.

7.4 Surface coatings

Exterior and interior surface coatings for corrosion protection and esthetics shall be specified to the extent necessary. The specifications may include conformance to accelerated corrosion tests, such as salt spray, or identification of particular products and application methods. Interior surfaces should be painted with a light colored protective coating compatible with the lubricant to provide a hard smooth surface that is easy to clean. A particular color for the final topcoat of exterior paint may be specified.

7.5 Quality assurance

All requirements for quality assurance of the gearbox shall be specified. As a minimum, each of the items of annex C should be addressed.

7.6 Analysis, drawings and data

The purchaser shall specify the requirements for analysis and documentation of the design. The level of detail, type of analysis, format of documentation, method for review and acceptance should be specified. A checklist for the required items is given in annex E.

Annex A
(informative)
Wind turbine architecture

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ISO 81400-4, *Wind turbines - Part 4: Design and specification of gearboxes*]

A. Purpose

A wind turbine is an energy conversion system that harnesses wind as a power source for the production of electricity. This annex describes the primary components that define the configuration as well as the type of control systems used to regulate the power output.

A.1 Overview

All wind energy conversion systems have a rotor consisting of one or more blades that convert wind energy to shaft torque. The nature of aerodynamics limit the speed of the rotor to levels below that required by standard generators. The speed difference frequently requires a speed increasing gearbox to operate the turbine economically.

Unfortunately, the blades transfer unwanted loads from the wind. Changes in wind speed and direction cause loads on the wind turbine that are very unsteady and vary significantly in both magnitude and direction. These forces need to be properly transferred to the ground via the rotor shaft, bed-plate, tower and foundation.

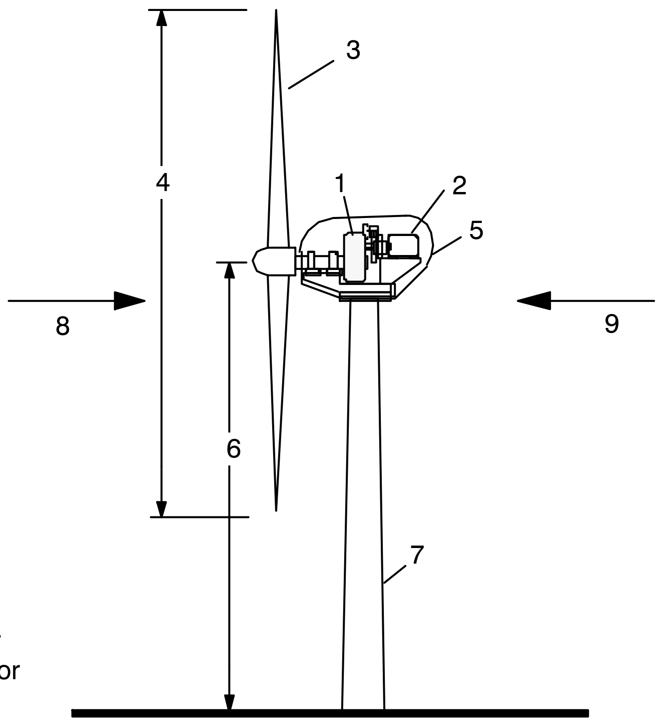
Additionally, mechanical and electrical controls are required to limit rotor loads and to provide grid synchronization and safe operation.

A.2 Turbine configuration

There are two main configurations of turbines; horizontal axis wind turbines, HAWT, and vertical axis wind turbines, VAWT. Only the HAWT is discussed in this document.

A.2.1 Horizontal axis wind turbine

A typical HAWT is shown in figure A.1. In this case the rotor shaft rotates horizontally, or nearly so, atop a tower. HAWTs, the most common turbines, have the gearbox and generator located at the top of the tower. Advantages of HAWTs are lower fatigue loads, and the ability to utilize improved blade geometry to further reduce loads and costs.



Key
1 Gearbox
2 Generator
3 Rotor blade
4 Rotor diameter
5 Nacelle
6 Hub height
7 Tower
8 Wind direction for an upwind rotor
9 Wind direction for a downwind rotor

Figure A.1 - Horizontal axis wind turbine (HAWT)

A.2.2 System components

Most wind turbines are composed of the following major subsystems:

- rotor assembly;
- rotor shaft assembly (may be part of the gearbox assembly);
- gearbox assembly;
- yaw assembly (on upwind HAWTs);
- generator assembly;
- tower;
- control system;
- stopping, braking and fail safe systems.

A.2.3 Nacelle layout

A typical nacelle layout for a HAWT is shown in figure A.2. This particular nacelle shows a modular drive train (see A.5.2). The rotor weight and loads are supported by rotor bearings that transfer the loads through the bedplate to the tower top. The bedplate holds the generator rotor bearings, gearbox and all the control systems required at the tower top. The components of a HAWT nacelle can be arranged in integrated, modular or hybrid layouts. For further discussion of these options, see A.5.

A.3 Rotor control

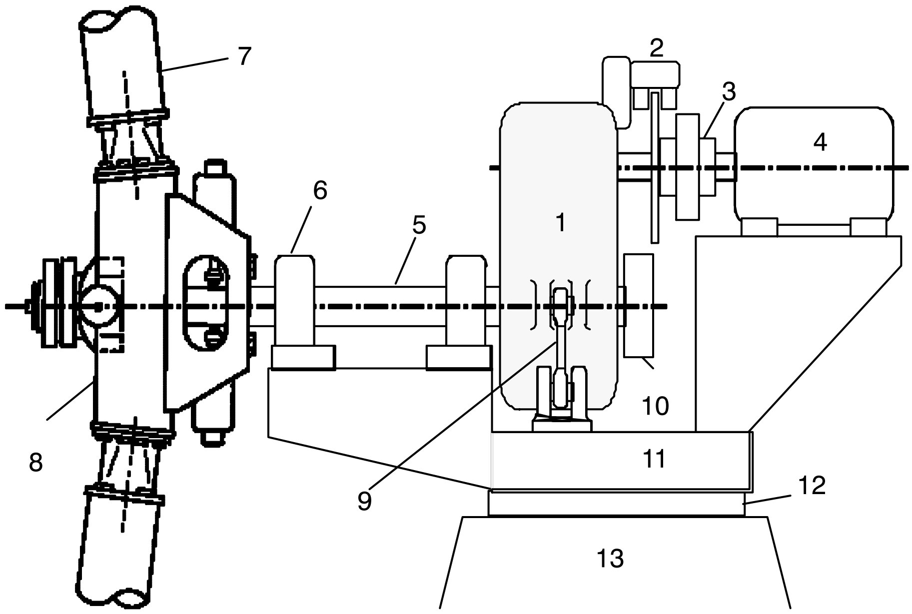
The control system is an integral part of the design philosophy. Most turbines are designed around a fixed pitch or variable pitch rotor.

A.3.1 Pitch control

Variable pitch machines regulate the power output by pitching the blades. They may pitch the whole blade, a control surface, or just the tip. Methods used to pitch blades may be mechanical, electromechanical, hydraulic or a combination of these. The blades are also pitched to optimize the amount of energy produced under varying wind conditions. Although the blades may not pitch fast enough to react to the dynamics of the wind, they can be pitched back and regulated to reduce transient loads caused by large gusts in high wind conditions. The blades can also be pitched during startup and shut-down to reduce the loads on the gearbox. Blades may pitch very rapidly to feather during emergency shutdown.

A.3.2 Stall control

Fixed pitch machines use the stall characteristics of the airfoil to regulate the amount of power produced. The blades are designed to produce a specific power at a given wind speed. At higher wind speeds, the airfoil stalls and does not allow the rotor to produce more power. Blade pitch angle may be changed manually to account for site specific or seasonal wind spectrum changes.



Key
1 Gearbox
2 Brake
3 Couping
4 Generator
5 Rotor shaft
6 Rotor bearing
7 Blade
8 Hub assembly
9 Torque arm
10 Shrink disc
11 Bedplate
12 Yaw bearing
13 Tower

Figure A.2 - Schematic of nacelle layout for a HAWT

The fail safe method for stopping these machines is usually a rotating blade tip, a mechanical brake or both. The brake is sometimes placed on the high speed shaft of the gearbox. This practice can result in tremendous shock loads on the gearbox. Extreme care must be given to quantify these loads to insure proper design of the gearbox.

A.3.3 Other control options

Some systems use sophisticated power electronics to limit torque and mitigate gust loads by allowing the rotor speed to vary. Others use yaw control, variable blade coning and nacelle pitching to limit power.

A.4 Yaw system

A HAWT typically requires that the turbine be able to rotate or yaw about the tower axis. This yawing action allows the rotor axis to align to the prevalent wind direction.

HAWTs have either passive or active yaw. Passive yaw turbines are oriented directly by the wind force acting on the blades or nacelle. Active yaw turbines have an electromechanical, mechanical or hydraulic system, which is controlled by a direction sensor to orient the nacelle.

A.4.1 Yaw bearing

One of two yaw bearing designs are commonly used. In a “sliding shoe” design, the turbine weight is carried by a bearing material installed between the nacelle and the tower and turbine thrust is resisted by devices to limit horizontal and vertical motion. The other common choice is a “slewing ring” which is a large diameter rolling element bearing.

If a sliding shoe design is selected, the ability to replace the bearing material should be incorporated into the maintenance requirements. If a slewing ring design is used, the bearing manufacturer’s recommendations for mounting, surface flatness, stiffness and bolting should be followed. Standard engineering practice for joint design can be used for review of bolt capacity, while the bearing capacity should be reviewed using standard bearing formulas and data. When investigating bearing capacity, it should be recognized that the application is “non-rotational”, since the turbine is oriented in one primary direction for extended periods of time. This situation can cause inadequate lubrication and damage to the bearing ring.

A.4.2 Yaw drive

Active yaw drive systems typically consist of one or more electric or hydraulic motor-driven reduction gears driving an internal or external ring gear fixed to the tower top.

A.5 Gearbox

A wind turbine gearbox is a speed increaser that increases the relatively slow speed of the rotor to the speed of the generator. The application is a demanding one that requires careful consideration of the load spectrum to ensure that the gearbox has adequate load capacity, and is within constraints on size and weight. Gearboxes must be designed to maximize efficiency while minimizing sound level. Because of the large number of wind turbines, and the limited accessibility for maintenance, reliability and maintainability are important considerations. The operational environment requires gearboxes that are resistant to temperature extremes, contamination and corrosion. For information on gearbox design see [2], [3], [4], [5], [6], [7] and AGMA 901-A92.

Many manufacturers have used standard shaft mount and foot mount gearboxes in their turbine configurations. Because these gearboxes are not specifically designed for use in a wind turbine, some bearing configurations may not be adequate for the dynamic loading conditions of a WTGS. Also, housings may not be stiff enough to provide proper bearing and gear alignment. It is suggested that full load testing be performed on these types of gearboxes before specifying them for use in a WTGS.

A.5.1 Integrated gearbox

In an integrated system the gearbox housing provides the bearing supports for the rotor and interfaces for other components such as generator, brake and yaw drives. Because the rotor shaft is integral with the gearbox, the gearbox housing is subjected to rotor loads. The housing must be carefully designed so that the load path transfers the rotor support loads to the tower without causing high stresses or excessive deflection in the gearbox housing. The complex shape of the gearbox housing usually requires that it be designed using finite element methods, FEM. These calculations should be verified experimentally. It is especially important to ensure that the housing deflection caused by installation or operation does not misalign the gears.

Field problems with the gearbox, rotor shaft or rotor bearings usually require that the rotor blades be removed and the generator, gearbox and rotor shaft be removed as an assembly. The combined weight of the assembly must be within the load capacity of normally available cranes and soil conditions. In large WTGS, this can be a major impediment to service. For instance, repairing and servicing components in WTGS with integrated gearbox systems may necessitate blade removal and re-placement, which can be significantly delayed (and cause significant downtime) by periods of high wind.

Integrated systems can reduce misalignment between the rotor and gearbox. It allows the gearbox lubricant to be used to lubricate the enclosed rotor shaft bearings. With an integrated system, seals must be carefully designed because there is the chance that the gearbox lubricant may leak into the generator, or wear debris or other contaminants from the generator or rotor may get into the gearbox.

A.5.2 Modular drive train

A modular system consists of a rotor shaft assembly, gearbox, generator and possibly a yaw drive, which are separately mounted to a common bedplate. In contrast to an integrated system, a modular system transfers the rotor support loads to the tower through bearings that are separate from the gearbox.

With a modular system, an individual component such as the generator, gearbox or rotor shaft can be removed for repairs without disturbing the other components.

A.5.2.1 Shaft mounted gearbox

Modular gearboxes may be either shaft mounted to the rotor shaft or foot mounted to the bedplate.

Shaft mounted gearboxes are fixed to the rotor shaft with a rigid coupling rather than a flexible coupling. This eliminates the need for aligning and lubricating the low speed coupling, and may save space.

Shaft mounted gearboxes have a torque arm that subjects the rotor shaft to a bending moment. Unless it is properly designed and mounted, the torque arm may twist the gearbox, causing the low speed gear mesh to misalign. Two torque arms may be used to cancel the bending moment, but this adds weight. Most shaft mounts use shrink disks that may be susceptible to fretting corrosion. Shrink disks require maintenance to ensure that the mounting bolts remain tight.

A shaft mounted gearbox requires a properly designed and mounted torque arm to maintain alignment of the high speed coupling. It also requires periodic maintenance to align the high speed coupling.

A.5.2.2 Foot mounted gearbox

Foot mounted gearboxes require a coupling that provides torsional compliance and damping to reduce the rotor dynamic loads before they reach the gearbox and gear teeth. Stresses on the rotor shaft are reduced because there is no bending moment from a torque arm, or gearbox weight, on the rotor shaft or rotor bearings.

Foot mounted gearboxes require an accurate and rigid bedplate to maintain alignment of the high speed and low speed couplings. They also require periodic maintenance to align the couplings. They may not be as compact as shaft mounted gearboxes.

A.5.3 Hybrid drive train

A hybrid drive train format is a combination of the integrated and modular designs.

A.6 Turbine operation

At startup, a WTGS is normally induced by the wind to accelerate up to speed at which point the controller closes the generator contactor. It will produce power as long as the wind speed is within operational range. In the event of a fault, excess power production, or loss of line, the rotor is stopped by either actuating rotor control surfaces or applying a brake.

Starting, running and stopping each have unique loads that must be considered when designing a gearbox. For procedures to include these effects in the load spectra, see B.5.

A.6.1 Startup

Depending on the type of electrical controls, the contactor closure may not always be at synchronous speed. This can result in transient loads that can be several times the rated generator capacity. At some sites such starts can happen quite frequently.

Some turbines use the generator as a motor to run the turbine up to speed. This can also cause high and reversing loads. WTGSs with two speed operation may also result in transient loading as one generator or generator winding is de-energized and the other energized.

A.6.2 Shutdown

Braking can have dramatic load effects on the gearbox. During high wind conditions or emergencies, the turbine may need to be stopped while producing rated power. This can also produce high loads.

A.6.3 Idling

Many hundreds of hours of operation can be experienced without power production. This can be either at low wind idling or negative power operation. Idling is operation at winds sufficient to start the rotor, but insufficient to sustain power production. Negative power production is the result of practical limitations of control during operation near the cut-in wind speed. The gear designer should be made aware of the relative amount of such operation.

A.6.4 Non-rotation

Due to low or no-wind periods during the year, the turbine will not rotate for weeks or months in some locations. Also, due to system faults causing shutdowns there can be periods when the parked rotor is subjected to wind loading. This can lead to fretting corrosion and other problems if the brake acts through the gearbox to react to parked rotor torque.

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Annex B
(informative)
Wind turbine load description

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ISO 81400-4, *Wind turbines - Part 4: Design and specification of gearboxes*]

B. Purpose

Gearbox problems have plagued the modern wind power industry since its initial growth period in the 1980s. Many of these problems stem from a lack of understanding by both gearbox manufacturers and wind turbine manufacturers of the severity of the wind turbine operational environment. More recently, a better understanding of the unique load characteristics of wind turbines has enabled more reliable wind turbine gearboxes to be developed. Experience has shown that to be successful, the wind turbine manufacturer should work with the gearbox manufacturer to assure that the gearbox loading is accurately defined for the wind turbine configuration being developed. This annex provides guidance on how this can be achieved.

B.1 General discussion

Primarily, the wind turbine manufacturer must provide the torsional loads on the drive train components for the lifetime of the wind turbine when specifying a gearbox. In addition, all other loads acting axially and laterally on the gearbox shafts or critical interfaces must be provided. A discussion of these structural loads is provided in B.6.

Torque loads on a wind turbine gearbox are driven by the response of the rotor and drive train to fluctuating wind conditions and to actions of the wind turbine's control sub-systems, such as the contactors, brakes, pitch control devices and over speed protection devices. The magnitude and dynamic coupling of these loads are difficult to predict and extremely variable. Additionally, loads arising from unsteady aerodynamics and atmospheric turbulence are complex and difficult to model. It is the responsibility of the wind turbine manufacturer to include each load condition and prepare a load spectrum that characterizes the true loading of the wind turbine.

B.2 Application, service and safety factors

Traditionally, the gear industry has used the terms "application factor" and "service factor" to describe multipliers that relate the calculated load capacity of a gearbox to the applied load. The term "factor of safety" or "safety factor" is commonly used by machine designers to define the ratio of allowable stress to applied stress. There are no universally accepted definitions for application factor, service factor or safety factor that relate to gearboxes. One must have a clear understanding of the meaning and significance of these terms when comparing gear load capacity using different industrial standards. Most of these factors work best when they are empirically determined from long experience with a particular application where the loads are well defined and do not vary significantly.

There has been sufficient experience with wind turbines to preclude the use of a single multiplier such as a service factor to specify a wind turbine gearbox. Wind turbine gearbox requirements vary significantly and are dependent on many factors such as wind turbine design class (see B.4.1), turbine configuration and local/regional regulatory requirements. Therefore, gearbox performance must be calculated based on a turbine specific load spectrum and a cumulative fatigue damage criterion, such as Miner's Rule [8] and [9].

B.3 Wind turbine configuration

The loads on a wind turbine system are greatly affected by the turbine system configuration and the control strategy. It is the responsibility of the wind turbine manufacturer to include the effects of each of the specific control functions and operational states that may be unique to the wind turbine's configuration. Two wind turbines with the same generator power rating may have extremely different gearbox torque load spectra. Some of the configuration features that influence the loading include, but are not limited to, the following:

- number of blades;
- method of regulating peak power;
- upwind or downwind rotor orientation;
- method of yaw control (passive or active);
- mechanical and aerodynamic braking methods and control strategy;

- weight of rotating and stationary components;

- configuration of integrated components (such as generator mount, rotor support, yaw bearing support and pitch control mechanism);

- controller response to fault conditions, for example, variable speed control.

The wind turbine manufacturer should understand how each feature affects the gearbox loading and convey to the gearbox manufacturer the importance of critical features that affect gearbox loading. The wind turbine manufacturer should provide a configuration description to the gearbox manufacturer to show how the gearbox fits into the drive train.

B.4 Wind turbine operating conditions

In determining the load spectrum for a wind turbine gearbox, the designer must make specific assumptions about the wind conditions at the design site. The site conditions will define the load spectrum. If the wind turbine is operated in other locations where the wind conditions differ, then the load spectrum could differ significantly.

B.4.1 IEC turbine classes

One effort to standardize the design methodology for wind turbines, which includes siting, has been conducted by the International Electrotechnical Commission, IEC. Under the IEC-61400-1 initiative, wind turbine design classes have been established that define the basic turbine site parameters for generic wind regimes of varying severity. The parameters include a reference extreme wind speed, annual average wind speed and annual average turbulence intensity. The reference extreme wind speed is used to define the extreme conditions that the wind turbine must survive at the site. The annual average wind speed, and annual average turbulence intensity are used to define a wind distribution. This is then used as a basis to determine energy capture potential and fatigue life. A Class 1 wind turbine, for example, operates in the most severe design site with a reference extreme wind speed of 50 meters per second, an annual average wind speed of 10 meters per second and a turbulence intensity of 0.17. The methodology for applying these parameters to the actual design load cases is described in IEC-61400-1.

B.4.2 Wind conditions

The loads on a wind turbine are directly related to the site and the design of the wind turbine. Wind turbine sites are typically represented statistically using a Weibull or a Rayleigh probability density function to describe the annual average wind speed distribution. The Weibull distribution is a two parameter function defined by the average wind speed and a “shape factor”. The Rayleigh distribution is a special case of the Weibull with a “shape factor” of 2. These distributions give the designer the annual average magnitude and frequency of occurrence of the wind speed at the site. The severity of the average turbulence is also provided. Wind distributions can be determined from wind measurements taken over several years [10], or generalized for typical site conditions using the statistical functions mentioned above.

Wind distributions do not give the designer information about other important inflow characteristics such as wind shear, wind direction changes, gust magnitude or complex three dimensional vorticity. These are necessary inputs for determining dynamic turbine behavior. These characteristics should be modeled using synthesized wind speed time series data. The algorithms for modeling the inflow are included in the IEC-61400-1 standard. Analytical wind turbine simulation models use the synthesized time series data as input to generate dynamic wind turbine loads [20].

It should be noted that the results of these models are no more accurate than the inflow data supplied, and it is difficult to fully characterize the true complexity of the wind across the rotor disc or for a period of time long enough to represent a full service life. The nature and significance of such uncertainty should be included in the loads description to the gear designer.

B.4.3 Wind turbine load conditions

It is the responsibility of the wind turbine designer to consider each operational condition as a potential load source and prepare a load spectrum that characterizes the aggregate loading on the wind turbine gearbox. Load fluctuations may result from, but are not limited to, any of the following conditions:

- turbulent wind fluctuations due to terrain, boundary layer and atmospheric effects, and wakes from other turbines;

- vertical and horizontal wind shear;

- gravity loads on overhung or cantilevered components;
- yawing motion of the rotor;
- off-axis yawed operation;
- unsteady loading due to the blades passing through the tower wake;
- transient starting loads due to generator control actions;
- loads due to motoring;
- transient stopping loads from aerodynamic and mechanical brakes;
- rotor mass imbalance or aerodynamic imbalance due to blade pitch differences;
- buffeting during parked rotor conditions;
- transportation and assembly;
- fault induced control actions.

The loading is a combination of normal operating loads and transient loads. The transient load cases may be numerous. The wind turbine designer must determine the torque load spectrum for each condition that affects the gearbox using an acceptable method.

Table B.1 shows a matrix of design load cases that typically have to be considered in developing a load spectrum for wind turbine gearboxes. This is derived from the IEC-61400-1 wind turbine design standard. Normally, a fully developed loads document is assembled from modeling the rotor and the system dynamic response to the wind and control actions. The results of analysis from the various load cases may, or may not be pertinent to the gearbox loading. In assembling the load part of the specification, the wind turbine designer has to choose which of these analyses to include. Any unusual transient events must be included in the assembled load spectrum, but time series descriptions should also be provided to the gearbox designer.

B.5 Gearbox design loads

B.5.1 General description

The wind turbine manufacturer should supply the design load spectrum to the gearbox manufacturer. It should be presented in tabular form, listing torque level and percent time at each torque level.

The load spectrum should specify all the torques that the gearbox will experience during its design life including the full range of operation. A more detailed description of the loading will be required for integrated gearbox designs to account for loading at all of the interfaces, in addition to the torque inputs. In this case a complete fatigue load spectrum consisting of cycle count matrices of mean and alternating loads will be required.

Determining the load spectrum is a difficult task due to a large amount of uncertainty associated with predicting operating and transient loads. This is especially true when extrapolating limited data to predict infrequent high load events. In light of these difficulties, the discussion is separated into normal operating loads and full range of loads.

Table B.1 - Wind turbine design load cases

	Design load case	Wind speeds	Other conditions	Load spectrum
1	Normal power production	Cut-in to cut-out		Normal
2	Power production plus fault	Cut-in to cut-out	Fault	Normal
3	Startup	Cut-in to cut-out		Show time series trace
4	Normal stop	Cut-in to cut-out		Show time series trace
5	Emergency stop	Cut-in to cut-out	Over speed or loss of line	Show time series trace
6	Parked (faulted)	Up to extreme wind speed	Fault or loss of line	Normal
7	Idling	Up to extreme wind speed	Fault	Normal
8	Transport/assembly			

B.5.2 Fatigue load spectrum

B.5.2.1 Normal operating loads

The normal operating loads for a wind turbine gearbox should be determined for the complete design life of the gearbox. This life is typically set to between 20 and 30 years in a given wind regime. The operating torque loads can be determined from analysis, experimental measurements, or a combination of both.

For new designs it is not usually possible to rely solely on experimental measurements since the design must precede the fabrication. If experimental measurements are used, they should be taken from a wind turbine with the same gearbox, turbine configuration, and operating environment, or scaled from a wind turbine of a similar configuration and size. Extreme caution should be used when scaling data. Differences in the turbine's configuration or operation that would change the loads must be included in the scaling process.

Several analytical codes are available to assist in developing the load spectrum. It is relatively easy to predict the mean torque level at a given wind speed for a wind turbine configuration using a computer code [11]. It is more difficult to determine the cyclic load variations about the mean values. A dynamic simulation code may be necessary to calculate the fluctuating loads since steady state models usually give only the mean values. Some research [12] has indicated that high magnitude cyclic operating loads contribute greatly to fatigue damage.

B.5.2.1.1 Generating load histograms

No single method has been adopted for determining the normal operating torque spectrum for a wind turbine. A generalized approach is presented here.

First, divide the operating wind speed range into bins no larger than 1 meters per second. For each wind speed bin, establish an operating torque load history using an appropriate dynamic wind turbine simulation time series model, and run the model for at least an hour in each bin. The mean wind speed of the time series input should correspond with the bin center wind speed. Next, convert the data from each bin into time-at-torque histograms. Weight each histogram by the corresponding wind speed probability from the design annual wind speed distribution. Combine the weighted histograms at the individual wind speed bins and scale the total time to the specified gearbox life. The resulting histogram will be a non-conservative approximation of the operating loads, because most analytical models are not able to predict the rare extreme load events that may contribute significantly to the overall fatigue damage. The subject of extreme load events is covered in B.5.3.

If experimental data in the form of a gearbox design spectrum is available from a turbine of similar size and configuration, then the designer may scale the distribution to account for turbine differences. If a torque load spectrum is not available but experimental measurements of shaft torque are available, then the above analytical procedure should be used, substituting the experimental data for the analytical time history. This method is described in reference [12] although the treatment of extreme loads is ignored.

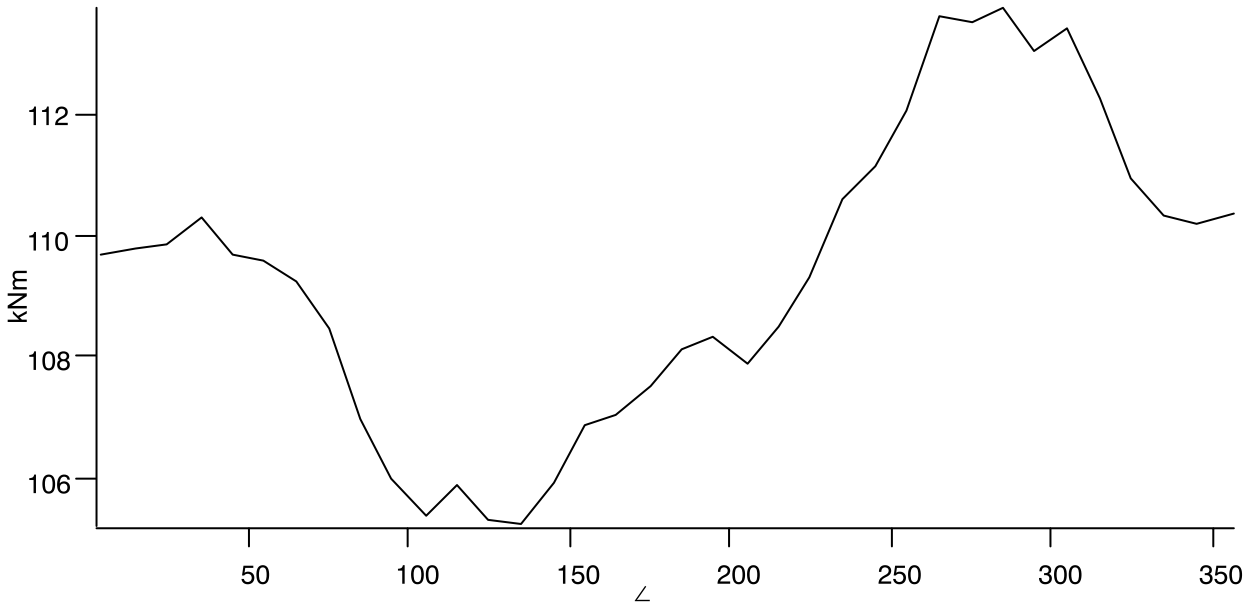
B.5.2.1.2 Simplified load histograms

A simplified approach is proposed in reference [13]. Wind speed variations are correlated with wind turbine power and low speed shaft torque fluctuations. The distribution about the mean torque for a given wind speed was shown to be Gaussian for several wind turbine configurations over the full operating wind speed range. The standard deviation of the Gaussian distribution is determined using the turbulence characteristics for the design wind site. The method was demonstrated on several wind turbines to determine the operating load conditions. However, this method does not consider the influence of transient conditions such as braking events, but the influence of extreme loading is accounted for in the tails of the Gaussian distributions.

B.5.2.1.3 Periodic loading

The peak deterministic load of each rotational cycle of the rotor shaft occurs at or near the same angular location most of the time. The analyses described above assume that all of the gear teeth share the load equally and that no tooth is loaded more severely than any other. Unfortunately, this is not always the case.

Figure B.1 shows the torque variation on the rotor shaft for a typical wind turbine averaged over more than 8000 rotations. At 280 degrees rotation the shaft experiences a maximum that occurs every revolution at about the same angular position. Similarly, the minimum occurs at about 120 degrees. The total azimuthal deviation is approximately 8 to 10 percent of the average torque load.



Torque and azimuth angle, ∠, experimental field data from 144 minutes of operation

Figure B.1 - Typical periodic variation in shaft torque

This may be especially important in designing gearbox components that do not change position with respect to the rotor over time (for example the low speed gear). It may also be important in selecting the number of teeth on the intermediate and high speed shafts. The number of teeth should be selected so the same pinion and gear teeth are not always engaged at the maximum torque position of the rotor. This can be achieved by using hunting tooth combinations.

B.5.2.1.4 Maximum operating loads

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

Over its design life, a wind turbine may experience many high load cycles during normal operation that cannot be witnessed by either experiment or analysis due to their infrequency. These events may be due to extreme wind speeds or rare three dimensional turbulent flow conditions. These load events may contribute significantly to the fatigue life of the wind turbine and gearbox, and their effects must be included when developing the fatigue load spectrum.

Several analytical techniques are available to calculate these loads using a limited amount of data. Typical methods involve extrapolating the extreme values from the tail of the measured or calculated torque load distributions. Extrapolation techniques are discussed in references [14], [15], [16] and [17]. None of these methods have been fully validated, but they provide a framework for addressing the problem.

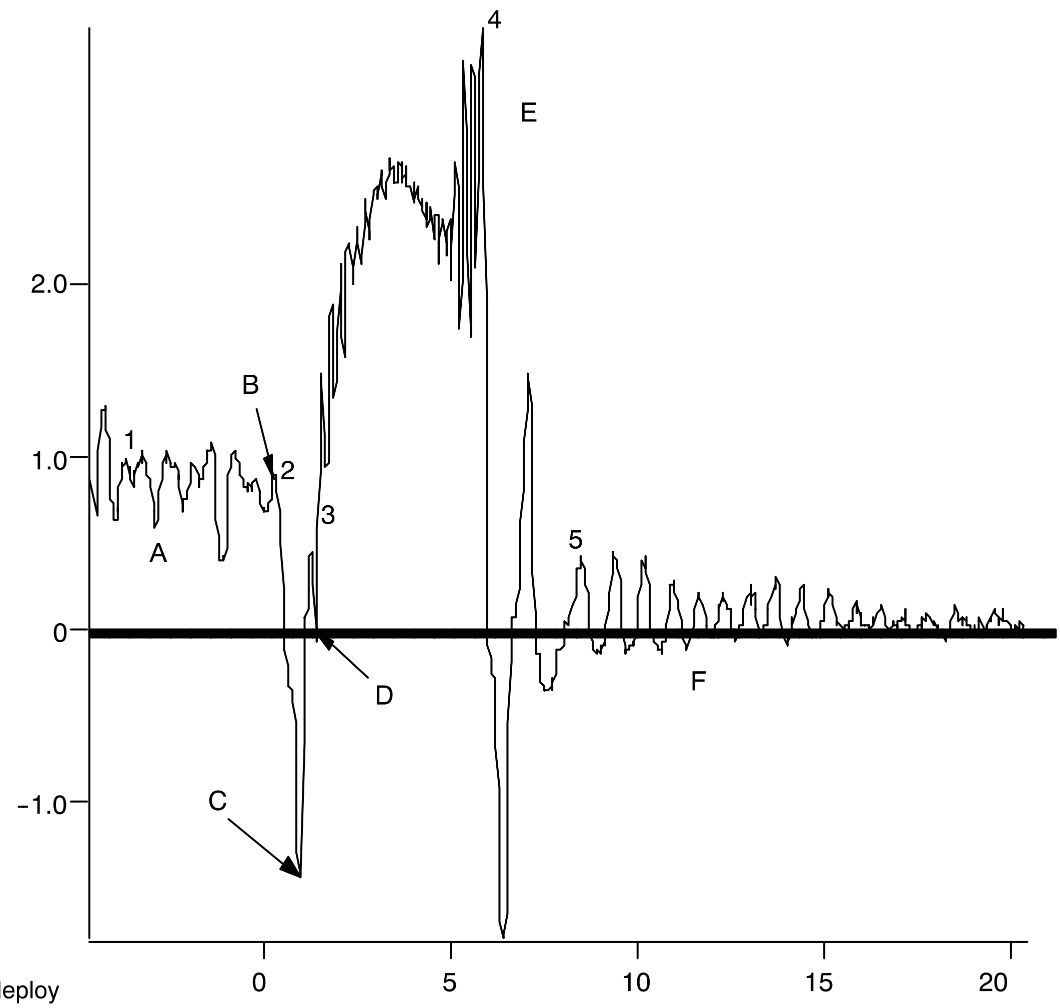
B.5.2.2 Transient loads

In many cases, the peak transient loads can be significantly greater than the extreme loads experienced during normal operation. Although they comprise only a small fraction of the operating time, they can significantly affect the service life of the gearbox and other components on the wind turbine. Moreover, understanding and controlling these transient loads is a sensible way to extend the life of a wind turbine without affecting its cost or productivity. Table B.1 gives many of the transient wind turbine design conditions that should be considered in developing the torque load spectrum.

B.5.2.2.1 Braking event description

In gearboxes with brakes on the high speed shaft, HSS, the gear teeth, bearings and other components may be subjected to very high transient loads during stopping that significantly exceed the design operating loads. These loads must be included in the load spectrum.

Figure B.2 shows a typical record of rotor shaft torque during a HSS brake stop. This wind turbine has a fixed pitch hub and blades equipped with pitching tips that act as aerodynamic brakes during stopping. At point 1 the generator is producing power. At point 2 the blade tips deploy dragging the generator into negative power. Between points 2 and 4 the drive train oscillates at its torsional natural frequency through one or more backlash/load reversal cycles. The mechanical brake is being applied but is not yet fully engaged. At point 3 the torque rises rapidly as the brake engages and the rotational velocity decreases. At point 4 the HSS comes to a stop. The shafts torsionally wind and unwind as the gear teeth are loaded on both their front and back sides through many backlash cycles, shown at point 5. These rapid torque reversals are repeated many times while the transient vibrations decay. In addition to causing gear tooth impact, torque reversals on helical gears cause reversing thrust loads and impact loads on the gearbox bearings and housing. Gear tooth backlash and bearing endplay should be minimized to limit stress caused by such impacts.



Key
A Operation
B Aero tips deploy
C Generator off-line
D Brake torque rising
E HSS stops
F Torque reversals and teeth impacts

Figure B.2 - Rotor shaft torque during braking event

B.5.2.2.2 Transient event spectra

Each transient event such as tip deployments, rapid pitch events, startup synchronization, shifting between multiple generators, has a unique signature, similar to figure B.2, which should be characterized first in the time domain by the wind turbine designer. These data are then analyzed in a similar way to the method described above for operating loads; by placing measured data or analyzed data into load spectra bins. Extreme caution should be used in taking data from other wind turbines to represent transient events because small changes in the configuration, components, or control strategy can make large differences in the dynamic response of the drive train. A dynamic simulation model of the drive train, which accounts for the rotating inertia and material stiffness of each component, should be used to generate the torque spectrum for the single event. One such model is described in references [18] and [19].

For each transient event type the wind turbine designer should estimate the number of probable occurrences over the life time of the wind turbine and scale the single event load spectrum by this quantity. These numbers should take into account the external effects such as the reliability of the grid. In some cases, a time series description of a typical event should be provided to the gearbox manufacturer to help describe the nature of the loading.

B.5.2.3 Combining torque load spectra

The normal operating load spectrum should be combined with the spectra for each of the transient conditions contributing to the torque load spectrum. This should be done by simply adding the contributions of each spectrum together, bin by bin. Depending on the uncertainty in the data, it may be necessary to apply some margin of safety to the loads. Such multipliers may be different for each design load case, and they may be required by IEC or other standards. The final spectrum and load descriptions should be submitted to the gearbox manufacturer with the other turbine information listed in annex E.

B.5.3 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.

Most wind turbines are designed to withstand a single extreme load event using ultimate strength criteria. This event is described by the IEC-61400-1 wind turbine safety standard, and it is based on a maximum load occurrence. The extreme design load will dictate the strength of many wind turbine components. This design condition is usually calculated with the rotor stopped and the blades oriented flat to the wind. Therefore, most of the load acts perpendicular to the rotor plane. This condition may not impart high or excessive torque loads to the system, but its influence should be considered.

The maximum one-time load event for a gearbox is likely to be a consequence of other events such as an emergency brake stop, generator short circuit fault or utility grid event. The wind turbine designer should determine the likely magnitude and probability of this maximum load and specify it separately to the gearbox designer. Other extreme load cases may apply to various gearbox components, for example bearings or housing, and they should similarly be specified and described.

B.6 Other loading

For integrated gearbox systems it is necessary to specify the load spectrum at all of the critical housing interfaces. An integrated gearbox housing supports the rotor bearings, and it commonly contains interfaces for the generator, pitch mechanism and yaw system. All of these subsystems transmit loads through the housing. Therefore a thorough analysis is required to describe all interface forces and moments.

Low speed shaft loads should include the lateral loading and bending moments resulting from such items as rotor weight, gyroscopic yaw loads, unsteady aerodynamics, wind shear and rotor thrust. The gear housing and housing base should be designed to withstand overturning moments and shear loads due to rotor thrust, as well as the torsional moments reacted through the gearbox and generator. A complete fatigue load spectrum should be developed for each of these interfaces, but the procedure for doing this is beyond the scope of this annex. The proper development of these loads requires a detailed dynamics simulation model [20], [21], [22] and [23] or a rigorous test program.

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Annex C
(informative)
Quality assurance

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ISO 81400-4, *Wind turbines - Part 4: Design and specification of gearboxes*]

C. Purpose

This annex explains the procurement process required to obtain reliable wind turbine gearboxes. It describes the procurement specification, quality assurance plan, quality control tests, quality documentation, and explains responsibilities of purchasers and gearbox manufacturers.

C.1 Definitions

Procurement specification - Specification designed and maintained by the purchaser that defines the application and load spectrum, and specifies minimum requirements for design, manufacturing, quality assurance, testing, and gearbox performance.

Design audit - Systematic and independent examination of engineering drawings and specifications to determine if all components of a gearbox meet requirements of the procurement specification.

Independent examination - Audit performed by a third party auditor or by a purchaser representative. As an alternative to an independent examination, the purchaser and gearbox manufacturer may agree to an appropriately documented internal audit performed by the gearbox manufacturer.

Quality assurance, QA, plan - Manufacturing specification designed and maintained by the gearbox manufacturer that specifies QA inspections, tests, and acceptance criteria for monitoring and controlling the manufacturing process.

Quality audit - Systematic and independent examination of a manufacturer's facilities to determine if equipment and quality procedures comply with requirements of the QA plan, and if QA inspections and tests are implemented effectively and meet requirements of the procurement specification.

Test audit - Systematic and independent examination of the QA plan and witnessing of tests to determine if tests of prototype, first article, and production gearboxes meet requirements of the procurement specification.

Manufacturing schedule - Schedule specifying start and finish dates for significant steps in the manufacturing process, including hold and witness points, for inspection and tests.

Manufacturing audit - Systematic and independent examination to determine if manufacturing, inspection, and testing of manufactured components comply with requirements of the QA plan, and if the components meet requirements of the engineering specifications and the procurement specification.

Prototype gearbox - The first gearbox of new design that is load tested in the field or on a test stand. After testing, the gearbox is completely disassembled and all components are inspected for wear or other distress. Based on the results of the prototype test, changes to design or manufacturing may be necessary. See 4.8.1 for further information.

First article gearboxes - One or more gearboxes manufactured before the start of manufacturing of a production lot of gearboxes. Components for first article gearboxes are 100% inspected for conformance to engineering specifications and requirements of the procurement specification, to ensure manufacturing and the QA plan are adequate. Based on the inspections of first article gearboxes, changes to manufacturing or the QA plan may be necessary. Requirements for run-in and load tests should be negotiated between the purchaser and gearbox manufacturer.

Production gearboxes - Gearboxes manufactured in production lots. Periodic manufacturing audits are conducted to ensure compliance with all specifications. Requirements for run-in and load tests should be negotiated between the purchaser and gearbox manufacturer.

C.2 Responsibilities

The procurement process involves many steps that evolve over time. Therefore, the purchaser should have resources adequate to ensure that each step of the procurement process is properly implemented and all requirements of the procurement specification are met. The purchaser is responsible for:

- procurement plan;
- procurement specification;
- bidding instructions;
- purchase order;
- design audit;
- QA plan audit;
- manufacturing schedule audit;
- manufacturing audit;
- test audit;
- installation, startup, and operation audits.

The gearbox manufacturer should have proven experience and capability necessary to manufacture gearboxes that conform to the procurement specification. The quality management system should provide resources adequate to ensure that each step of the design process, verification activities including testing, and manufacturing process is properly implemented and all requirements of the QA plan are met. The gearbox manufacturer is responsible for:

- proposal data;
- contract data;
- engineering drawings and specifications;
- QA plan;
- manufacturing schedule;
- pricing;
- warranty;
- instruction manuals.

C.3 Procurement plan

The following plan guides the purchaser through all phases of the procurement process from writing the procurement specification to auditing gearbox start-up.

C.3.1 Write procurement specification

A comprehensive procurement specification should be written to ensure a reliable gearbox. The procurement specification should define performance rather than design and should allow the gear manufacturer creative freedom to meet contract requirements by applying their engineering expertise. See the main body of ANSI/AGMA/AWEA 6006-A03 for guidelines for content of the procurement specification.

C.3.2 Solicit bids

The purchaser should solicit bids from gearbox manufacturers who have the experience and capabilities necessary to produce gearboxes that conform to the requirements of the procurement specification. Bid solicitation should include a questionnaire that determines if the gearbox manufacturer understands the procurement specification and has experience and capabilities to produce reliable gearboxes. A formal audit of the questionnaire response including a reference check and capability study is recommended for each major procurement. Bid solicitation should require a preliminary QA plan such as shown in table C.1.

C.3.3 Evaluate proposals and select final bidders

The purchaser should evaluate the gearbox manufacturer's proposals for completeness and conformance to the requirements of the procurement specification. Proposals should be compared and the best proposals should be selected for design review meetings. To determine manufacturer's capabilities, quality audits may be necessary.

C.3.4 Meet for design review meetings

Design review meetings should be conducted with final bidders and evaluations of each proposal should be written. Evaluations should discuss the relative merits of each proposal and provide sufficient information to select a gearbox manufacturer and perform a design audit.

C.3.5 Select gearbox manufacturer, audit design, and award contract

After considering the relative merits of each proposal, the purchaser should select the final gearbox manufacturer and audit the gearbox manufacturer's proposal. The design audit should include the following as a minimum:

- conformance to procurement specification;
- gearbox design;
- gear design;
- bearing design;
- shaft design;
- seal design;
- lubrication system design;
- QA plan for critical components.

Table C.1- Sample QA plan

<table>
<tr><td colspan="8"><BIDDER> <Bidder Name> <Order No.></td></tr>
<tr><td colspan="8"><Signature of quality control manager and date></td></tr>
<tr><td colspan="8">LEGEND:
H = Hold Point – Operation or procedure must be witnessed by purchaser’s representative before moving component to next operation.
W = Witness Point – Operation or procedure may be witnessed by purchaser’s representative if representative is present during manufacture.
D = Document Required – Quality assurance must provide certified copy of inspection or test report to purchaser’s representative.</td></tr>
<tr><td>Procurement Spec No. <Spec No.> Rev. <Rev. No.> Activity</td><td>H</td><td>W</td><td>D</td><td>Proc. Spec. Clause</td><td>Bidder Spec. No.</td><td>Bidder Clause No.</td><td>Bidder Form No.</td></tr>
<tr><td>Gear raw material</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Process</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Form</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Chemistry</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Grain size</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Hardenability</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Cleanliness</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>UT inspect forgings</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Inspection of gear teeth</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Basic geometry</td><td></td><td>X</td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Accuracy</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Root fillets</td><td></td><td>X</td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Grinding stock removal</td><td></td><td>X</td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Surface roughness</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Magnetic particle</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Surface temper</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Surface hardness</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Inspection frequency</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Inspection of coupons</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>General</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Case hardness</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Case depth</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Core hardness</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Case microstructure</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Carbides</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Decarburization</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Carbon content</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Microcracks</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Secondary transform. products</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Intergranular oxidation</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Retained austenite</td><td>X</td><td></td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Core microstructure</td><td></td><td>X</td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Post carburize cold treat</td><td></td><td>X</td><td>X</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Housing accuracy</td><td></td><td>X</td><td>X</td><td></td><td></td><td></td><td></td></tr>
</table>

(continued)

Table C.1 *(concluded)*

Shaft material		X	X				
Shaft hardness		X	X				
Shaft accuracy		X	X				
Shaft magnetic particle		X	X				
Gearbox assembly	X		X				
Tests	X		X				
Contact patterns	X		X				
Load or no-load tests	X		X				
Dykem	X		X				
Lubrication	X		X				
Sound level	X		X				
Vibration level	X		X				
Oil leaks	X		X				
Oil sump temperature	X		X				
Dykem patterns	X		X				
Corrective action	X		X				
Documentation	X		X				
Preparation for shipment	X		X				
NOTE: This is a sample plan and is not inclusive of all items; the important items are shown.							

A report summarizing the design audit should be prepared. Manufacturing should not begin until the purchaser approves the engineering drawings and QA plan, and all agreed to design changes are incorporated into the procurement specification.

C.3.6 Review and approve engineering drawings

The engineering drawings should be reviewed to confirm they meet the requirements of the procurement specification.

C.3.7 Review and approve QA plan

The final QA plan should be reviewed to confirm it meets the requirements of the procurement specification for manufacturing, quality assurance, and testing. The detailed quality assurance plan for the proposed gearbox should be consistent with the manufacturer's overall quality system, such as ISO 9000 or an equivalent. If the gearbox manufacturer wishes to deviate from the procurement specification or QA plan, the alternatives should be described by the following:

- reference to procurement specification clause;
- description of alternative;
- requirements for verification of alternative;
- purchaser's signature of approval.

Table C.1 is a sample QA plan for the prototype gearbox, first article gearboxes, and production gearboxes. It emphasizes gears but includes other items. A similar plan is needed for each load bearing component and the gearbox as an assembly. The QA plan should be detailed enough to uncover any problems during manufacturing of prototype gearboxes. The same QA plan, except as modified based on results of the prototype test, should be used for first article gearboxes to ensure corrective actions have been successful, and the gearboxes meet requirements of the procurement specification before production manufacturing begins. The same QA plan should be used for production gearboxes. However, the frequency of inspections and tests, and schedule of hold and witness points should be negotiated between the purchaser and gearbox manufacturer.

As a minimum, the QA plan should specify control methods for the following:

- gear raw material;
- quality control of subcontracted work and purchased parts;
- quality control of gear metallurgy;
- inspection of gear teeth;
- housing accuracy;

- shaft accuracy;
- quality control in assembly including fitting, cleanliness, serialization and traceability of components;
- gearbox tests, acceptance criteria, and record keeping requirements.

Documentation required by the QA plan should be traceable. All gears and shafts should be serialized and all inspection documents should record serial numbers of components. Serial numbers of all gears and shafts should be traceable from the serial number on the gearbox nameplate and traceable to raw material heat and melt numbers. All test specimens, representative test coupons, heat treatment records, and all significant documents from manufacturing, inspection, testing, and processing should be traceable to components they represent. All measuring instruments and artifacts should be traceably calibrated to national standards and have current calibration certificates. All QA documents generated by the gearbox manufacturer should be maintained for a period agreed to by the purchaser and the gearbox manufacturer, but not less than ten years.

C.3.8 Review and approve manufacturing schedule

The manufacturing schedule should allow sufficient time for corrective action deemed necessary from results of the prototype test, and changes to manufacturing or the QA plan deemed necessary based on inspections of first article gearboxes.

C.3.9 Evaluate performance of prototype gearbox

After the prototype gearbox has been tested and inspected, and all necessary changes to design, manufacturing, and QA plan are made, manufacture of first article gearboxes should be approved.

C.3.10 Inspect first article gearboxes

After the first article gearboxes are inspected, and supporting documentation reviewed, make recommendations for changes to engineering and manufacturing specifications and the QA plan.

C.3.11 Approve manufacture of production gearboxes

After the final engineering and manufacturing specifications and final QA plan have been approved, and first article gearboxes have met requirements of the final QA plan, manufacturing of production gearboxes should be approved.

C.3.12 Audit manufacturing and testing of production gearboxes

Periodic audits of manufacturing and testing should be conducted to resolve problems and ensure compliance with the procurement specification and QA plan.

C.3.13 Audit installation, startup, and operation of production gearboxes

Installation, startup, and operation should be audited to ensure all instructions for installation, operation, and maintenance are properly implemented. Gearbox performance should be closely monitored during startup and the first few weeks after startup, followed by periodic monitoring during operation. Monitoring should include measurements of temperature, sound, and vibration, and lubricant analyses.

C.4 Resolution process

A formal process to allow for resolution of as-built deviations from the manufacturing specifications should be developed. The method to resolve deviations should be agreed to between the purchaser and the gearbox manufacturer as part of the QA plan. For example, the resolution process could be utilized if a grinding notch in a gear tooth occurs.

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Annex D
(informative)
Operation and maintenance

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ISO 81400-4, *Wind turbines - Part 4: Design and specification of gearboxes*]

D. Purpose

The operation and maintenance of a wind turbine gearbox is as important as designing, manufacturing and procuring the gearbox, and it should be fully defined prior to purchasing the gearbox. Startup, operating, and maintenance plans and procedures should be established by the gearbox manufacturer, lubricant manufacturer, and purchaser before the gearbox is placed in service.

D.1 Startup and run-in procedures

The bedplate, mounting surfaces, shims and shaft couplings must be accurately aligned for proper overall performance.

For maximum bearing and coupling life, shaft misalignment should be minimized. Load variations and temperature changes cause the alignment to change. Therefore, it is best to carefully align couplings to ensure that the couplings have adequate capacity to accommodate increases in misalignment. In some cases it is necessary to bias the cold alignment to minimize shaft misalignment at the operating temperature. For shaft mounted gearboxes, the torque arm should be properly located and mounted without excessive clearance in the mounting points to ensure that the gearbox housing is not twisted, and the high speed coupling is not misaligned. Before starting the equipment, these checks should be made:

- oil level and type;
- pipe connections;
- electrical connections;
- torque on mounting and gearbox bolts;
- operation of automatic shutdowns and alarms;
- coupling installation and alignment;
- inspection cover installation;
- heater, cooler, and fan operation.

During initial startup, these procedures are helpful:

- Pre-oil the unit to lubricate gears and bearings;
- For cold environment startups, preheat the lubricant. Load should not be applied until the oil has attained operating temperature;
- Start the gearbox slowly under light load. Check for proper rotation direction. Check system oil pressure;
- After oil circulates, stop the unit, check oil level, and add as necessary;
- Monitor the gearbox for vibration and temperature. If any problems are detected, shut down immediately, and take corrective action;
- If a factory run-in procedure is not performed, operate the first 10 hours at reduced load to run-in tooth surfaces. This will reduce the risk of scuffing and prolong the life of the drive. Not performing such a run-in may lead to premature failure;
- After a field performed run-in, the gearbox should be drained and flushed to remove contaminants, refilled with recommended lubricant and a new oil filter should be installed if applicable;
- Check coupling alignments and re-torque all bolts. Check all piping connections and tighten as necessary.

Finally, to confirm that the requirements of the procurement specification have been met, gearbox performance should be monitored under actual service conditions. Because the first few weeks after startup are critical, data on load, vibration, and performance should be collected. Not every application warrants full telemetry and strain gaging. However, power, temperature and vibration level should be recorded for every gearbox. Also the lubricant should be analyzed for contamination. These actions will check the purchasing process and involve the gearbox manufacturer in corrections as necessary.

D.2 Lubrication

D.2.1 Change interval

The lubricant should be changed based on the gearbox manufacturer's recommendations and the results of the oil monitoring program. Monitoring and testing the lubricant for viscosity, water content, acid number, solid contaminants, and additive depletion will help determine the proper change interval. For a discussion of lubricant monitoring, see annex F.

D.2.2 Recommended analysis limits

Table F.4 gives recommended limits for analysis parameters and typical contaminants for wind turbine gearboxes.

D.2.3 Storage procedures

Drums of spare lubricant should be properly stored to avoid contamination. Drums should be stored indoors, horizontally.

D.3 Gearbox inspection

To assure reliable operation, it is necessary to have a periodic gearbox inspection program. Inspections should commence during or shortly after the run-in period, then progress at regular intervals thereafter.

Detailed records of the inspections should be maintained and may include vibration level, bearing and shaft end play, radial play, bearing and tooth condition, oil condition and overall gearbox condition. If available, WTGS operating hours, energy production and maximum operating temperature should be recorded. When flexible couplings are used, alignment should be inspected as required.

Photographs are an excellent means of tracking gearbox/gear condition. The teeth can be identified and rephotographed during subsequent inspections. Reference [26] provides useful suggestions for such photography.

Inspection personnel should be thoroughly trained to provide the maximum consistency of observations and adjustments.

D.4 Maintainability

This clause addresses specific maintenance areas common to all WTGS gearboxes. The general maintainability of wind turbine gearboxes is a very important issue and should be addressed at the design stage. Technicians must have safe access to equipment that requires maintenance. The work space on top of a wind turbine tower is often very limited and may be hazardous, therefore some planning is needed when choosing the placement of breather, magnetic plugs, drain lines, fill plugs, filters, inspection ports and dip sticks.

The wind turbine operator should consider the following:

- flushing of gearbox prior to adding new lubricant;
- accessibility of filters for changing when using a pressure fed filtered lubrication system;
- use and connections for a portable filtering system;
- methods for draining oil from the gearbox;
- transport of oil to the nacelle;
- filling the gearbox with oil in the nacelle;
- contamination prevention during the filling process.

D.5 Gearbox safety requirements

Safety is a major concern of turbine operators or anyone who is involved with large rotating equipment. Most maintenance for a HAWT is done up-tower in a confined space. Technicians use safety belts and lanyards to attach themselves to the machine to prevent injury from a fall. This may place workers in close contact of a potentially harmful situation if safety procedures are not followed.

The following requirements should be considered to ensure a safe working environment and minimize the probability of a situation potentially injurious to the workers:

Lanyard hook points. There should be an adequate number of appropriately marked hook points on the gearbox or adjacent components to ensure that workers can be attached with at least one lanyard at all times when working in the gearbox area.

Drive lines, shafts and couplings. All drive lines, shafts and couplings should be covered to prevent accidental entanglement by the worker.

Lubrication. Design of the oil system and the maintenance methods of changing the oil should be developed to minimize oil leaks and spills. Oily platforms and ladders are potentially hazardous and may be considered an environmental hazard.

Others. Consideration should be given to adding steps and handholds to the gearbox to provide access to all components. In addition, the housing should have rounded corners.

Annex E
(informative)
Minimum purchaser and gearbox manufacturer ordering data

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ISO 81400-4, *Wind turbines - Part 4: Design and specification of gearboxes*]

E. Purpose

The following technical information is recommended for procurement documents and for offers by potential gearbox manufacturers.

Certain information in the following lists may be intellectual property requiring non-disclosure agreements between the parties involved in the exchange.

E.1 Information to be provided by purchaser

The following items should be provided by the purchaser to define technical and quality assurance requirements.

E.1.1 Applicable standards and specifications

Industry standards such as:

IEC
ISO
AWEA
ANSI/AGMA
ABMA
AWS
ASTM
AMS
SAE
Applicable local/governmental regulations
Project specifications

E.1.2 General description of wind turbine

Manufacturer
Model
Rotor – upwind or downwind
Rotor diameter
Rotor speed
Generator speed
Direction of rotation
Number of blades
Blade pitch - fixed or variable
Hub height
Yaw type - active or passive
Brake type and size
Brake location
Number of generators
Rated power
Rated wind speed
Cut-in wind speed
Cut-out wind speed
Over speed
Gearbox enclosure (nacelle type)

E.1.3 Environment

Various environmental (climatic) conditions other than wind can affect the integrity and safety of the WTGS, by thermal, (photo-) chemical, mechanical, electrical or other physical action. Moreover, combinations of the climatic parameters may increase their effect.

Temperature
Humidity
Air density
Solar radiation
Rain, hail, snow and icing
Chemically active substances
Mechanically active particles
Lightning
Earthquakes
Salinity
Altitude

An offshore environment requires special additional considerations.

E.1.4 Turbine loads

E.1.4.1 Gearbox design torque loads

Fatigue load spectrum

Maximum operational loads

Transient loads

E.1.4.2 Extreme loads

E.1.4.3 Other loads

Critical housing interfaces

External low speed shaft loads

E.1.5 Gearbox design requirements

Gearbox ratio and tolerance

Rating methods for gears and bearings

Required service life and reliability of gears and bearings

Housing components including:

- Inspection covers
- Oil level indicator
- Oil sampling ports
- Magnetic plugs
- Drain valves
- Breathers
- Lifting holes
- Brake mounting pads
- Torque arm mounting lugs

Seal types and materials

Instrumentation/transducers/alarms

Air velocity over gearbox

E.1.6 Lubrication

Lubrication system

Lubricant change interval

Filtration system

Filter change interval

E.1.7 Turbine arrangement drawing

Boundary dimensions and interfaces

Gearbox orientation and mounting position

Shaft orientation

Direction of rotation

Weight limit

Inclination angle of gearbox low speed shaft from horizontal

E.1.8 Accessories

Coupling type, model, size

Coupling misalignment limits

Brake type, model, size

E.1.9 Maintenance components

Spares

Tools

Technical manuals

Maintenance manuals

Drawings

E.2 Information to be submitted or provided access to by gearbox manufacturer

E.2.1 Proposal data

Gearbox manufacturers should submit or provide access to detailed drawings and the data contained in E.2.1.1 through E.2.1.3. The gearbox manufacturer should provide data that shows the gearbox was designed and rated in accordance with the requirements of the procurement specification and the load spectrum. The data should be sufficient to allow the purchaser to audit the gearbox design for conformance to the procurement specification.

E.2.1.1 Outline dimension drawing

An outline dimension drawing of the gearbox that fully describes its overall dimensions and interfaces.

E.2.1.2 Assembly drawing

An assembly drawing of the gearbox that gives the part numbers of all components and the catalog numbers of all bearings. The drawing shall also indicate the rotational direction of gears and shafts.

E.2.1.3 Detailed drawings

Access to detailed drawings of all gears. The following data should be supplied as a minimum:

Number of teeth

Net face width

Outside diameter

Normal module

Normal pressure angle

Helix angle

Hand of helix

Designation of active flank(s)

Operating center distance

Profile shift coefficient

Tooth thickness

Finish stock allowance

Gear cutting method

Gear finishing method

Cutting tool normal tooth thickness

Cutting tool addendum

Cutting tool tip radius

Tip and root relief

Helix modification

Profile modification

Tip chamfer, edge round, end round

Tooth surface roughness

Geometric quality

Metallurgical quality

Material alloy and grade

Heat treatment method

Tooth surface and core hardness

Effective case depth to 50 HRC or 550 HV (for surface hardened members)

E.2.2 Contract data

Gearbox manufacturers should submit data contained in E.2.2.1 through E.2.2.4, and E.2.2.6. Purchasers should approve the drawings, quality assurance plan, and manufacturing schedule before manufacturing begins.

E.2.2.1 Drawings

Drawings should be adequate to achieve the technical requirements of the procurement specification.

The purchaser should state in the inquiry or the purchase order, the number of prints required and the schedule within which they should be submitted by the gearbox manufacturer.

E.2.2.2 Quality assurance plan

The quality assurance, QA, plan should be adequate to achieve quality goals defined by the procurement specification. The QA plan should specify inspections and tests required for monitoring and controlling manufacturing, and identify hold and witness points for audits by the purchaser's representative. See annex C for additional information.

E.2.2.3 Manufacturing schedule

The manufacturing schedule should specify the manufacturing sequence and schedule inspections and tests required by the QA plan.

E.2.2.4 Certified drawings

After the drawings have been approved, the gearbox manufacturer should furnish certified copies in the quantity specified.

E.2.2.5 Purchaser approvals

The gearbox manufacturer should not proceed with manufacturing until drawings, QA plan, and manufacturing schedule are approved by the purchaser in writing. Approval of drawings, QA plan, and manufacturing schedule should not constitute permission to deviate from any requirements in the procurement specification or purchase order unless specifically agreed upon separately in writing.

E.2.2.6 Instruction manuals

The gearbox manufacturer should furnish the required number of instruction manuals for the gearbox. The manuals should include instructions for installation, startup, operating and maintenance. The manuals should also include recommendations for lubrication and storage.

E.3 Contractual data agreed upon by purchaser and gearbox manufacturer

The following items may involve shared responsibilities and should be negotiated by the purchaser and gearbox manufacturer.

E.3.1 Technical requirements

Mass elastic data

System dynamics

Weight

Center of gravity

Structure vibration

Lubricant type and viscosity

Lubricant capacity

Interface dimensions

Run-in requirement

E.3.2 Quality assurance requirements

Responsibilities

Design review

Quality system

Quality assurance plan

Inspection and test schedule

Witnessing agency

Documentation

Traceability

Certification

Prototype tests

First article inspection and tests

Production tests

Inspection and test approval

Corrective action

Preparation for shipment

Preparation for storage

E.3.3 Commercial and other requirements

Guarantee and warranty

Annex F
(informative)
Lubrication selection and condition monitoring

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ISO 81400-4, *Wind turbines - Part 4: Design and specification of gearboxes*]

F. Purpose

Proper selection and condition monitoring of a lubricant is essential to achieving maximum service life for wind turbine gearboxes. This annex provides information on gear lubricant types, additives, filtration and condition monitoring. For further discussion of wind turbine lubricants, see [31], [32] and [41].

F.1 Gear lubrication

The functioning of mechanical components such as gears and bearings involves relative motion of surfaces in contact, under load. Separation of the surfaces by a thin film of fluid is a key factor in achieving smooth operation and good service life. Lubrication is typically classified by regimes, such as hydrodynamic, elastohydrodynamic, and boundary. Gears, bearings and other highly loaded contacts spend the majority of their time in the elastohydrodynamic regime. The theory of elastohydrodynamic lubrication (EHL) is an extension of hydrodynamic lubrication. It was developed to explain the role of the fluid film and to analyze its behavior under various rolling contact operating conditions.

F.1.1 Lubrication regimes

Development and operational influence of the EHL fluid film depends on lubricant characteristics, lubricant inlet conditions, load intensity, operating pitch line velocity, surface conditions, metallurgy and oil cleanliness. These parameters are further discussed in AGMA 925-A03. It is common practice to categorize EHL lubrication into three regimes.

F.1.1.1 Hydrodynamic lubrication

Hydrodynamic lubrication exists when there is complete separation of the sliding and rolling contact surfaces achieved by a continuous lubricant film. Hydrodynamic lubrication films are formed generally under light loads and moderate pitch line velocities, and low or high loads with high pitch line velocities. The surface roughness λ ratio also plays a significant role.

F.1.1.2 Elastohydrodynamic lubrication

EHL exists in highly loaded sliding and rolling contacts when fluid film formation is influenced by elastic deformation of the contacting surfaces. EHL is prevalent in wind turbine gears and bearings, which operate under heavy loads and moderate pitch line velocities.

F.1.1.3 Boundary lubrication

Boundary lubrication exists when the sliding and rolling surfaces are wetted with fluid but the lubricant film thickness is small compared to the sliding and rolling surface roughness. Boundary lubrication occurs under heavy loads and low pitch line velocities.

F.1.2 Damage modes

There are numerous modes of damage associated with rolling element bearing components and gear teeth. The damage modes of gear teeth are listed in ANSI/AGMA 1010–E95. Two of these modes, Hertzian fatigue (macropitting and micropitting) and wear (adhesion, abrasion, polishing and scuffing), are strongly influenced by lubrication. Proper selection, application, and maintenance of lubricants are, therefore, essential in avoiding premature failure. For information on gear failure modes, see [24], [25], [26], [27], [28], [29], [34] and [35].

F.1.3 Lubricant selection

Lubricants recommended in this document will normally produce good results, but verification by analysis of the gear and bearing contact specific film thickness is desirable. Such analysis is beyond the scope of this document. For spur and helical gears, a method is given in AGMA 925-A03. For information on gear lubrication, see [24], [27],[30], [31], [32], and [33].

Viscosity is a key property of the fluid selection process. Tables F.5 through F.8 provide guidelines for selecting an appropriate viscosity grade in the absence of detailed analysis.

F.2 Lubricant base oil types

A key element of the finished fluid is the base oil. The base oil comes primarily from two general sources:

mineral or synthetic. Additionally, natural products such as vegetable-based oils could be used as a base oil especially where biodegradability is an issue. The term mineral usually refers to base oils that have been refined from a crude oil source, whereas synthetics are usually the product of a chemical reaction of one or more selected starting materials. The finished fluids can also contain mixtures of one or more base oil types. Partial synthetic fluids contain mixtures of mineral and synthetic base oils. Full synthetic fluids can also be mixtures of two or more synthetic base oils. As an example, mixtures of polyalphaolefins and esters are commonly used in synthetic formulations today. Mixtures are generally used to tailor the properties of the finished fluid for a specific application or need.

F.2.1 Mineral oil lubricants

Mineral based gear oils have viscosity indices (measurement of viscosity change with temperature variation) that are commonly lower than most, but not all synthetic based gear oils. An advantage of mineral based oils over synthetic based oils is usually their lower initial purchase cost. If a mineral oil is preferred, a mineral oil of higher viscosity can be used to match the viscosity of a synthetic at higher operating temperatures and the lubricant can be preheated for startups if necessary. Mineral based gear oils have been used successfully for many years in properly designed, operated and maintained WTGSs.

F.2.2 Synthetic lubricants

Synthetic oils have special properties that may enhance performance or accommodate severe operating conditions. Many synthetic oils are stable at high operating temperatures and have high viscosity indices (smaller viscosity changes with temperature variations) and low pour points. WTGSs using most commercially available synthetic gear oils can be started without difficulties at a lower bulk oil temperature than those using mineral oils.

Each type of synthetic lubricant has unique characteristics and all have limitations that should be understood. Such things as compatibility with other lubrication systems and mechanical components (seals, sealants, paints, backstops and clutches), behavior in the presence of moisture, lubricating qualities and overall economics should be analyzed carefully for each type of synthetic lubricant under consideration for a given application. In the absence of field experience in similar applications, the use of synthetic oil should be coordinated carefully between the user, the gearbox manufacturer and the lubricant manufacturer.

Synthetic lubricants may improve gearbox efficiency and may operate cooler than mineral oils because of their physical, chemical and thermal properties. Decreasing the operating temperature of a gearbox lubricant is desirable. Lower lubricant temperatures increase the gear and bearing lives by increasing lubricant film thickness, and increase lubricant life by reducing oxidation.

There are several different types of synthetic base oils available today. Their compositions and properties result from the different chemicals that are combined in their manufacture. Some of the major types of synthetic base oils are described in the following clauses. For additional information on synthetics see [37].

F.2.2.1 Polyalphaolefin, PAO, lubricants

Polyalphaolefins, or olefin polymers, are paraffin-like liquid hydrocarbons which can be synthesized to achieve a unique combination of high-temperature viscosity retention, low volatility, very low pour point and a high degree of oxidation resistance along with a structure that can improve equipment efficiency. These characteristics result from the wax-free combination of moderately branched paraffinic hydrocarbon molecules of predetermined chain length.

Compared to conventional mineral oils, PAO lubricants generally have poorer solvency for additives and for sludge that may form as the oil ages. Lubricant manufacturers usually add a higher solvency fluid, such as ester or alkylated aromatic fluids, in order to keep the additives in solution and to prevent sludge from depositing on the gearbox components.

F.2.2.2 Synthetic ester lubricants

Esters are produced from the reaction of an alcohol with an organic acid. There are a wide variety of esters available that can be produced by the numerous combinations of acids and alcohols available. The principle advantage of many esters is their excellent thermal and oxidative stability. A primary weakness for some is poor hydrolytic stability. When in contact with water, esters may deteriorate through a reverse reaction and split back into an alcohol and organic acid. A secondary weakness with some esters is a viscosity index lower than most paraffinic mineral based oils. Some esters do, however, provide a viscosity index higher than mineral or PAO lubricants.

Ester based gear oils that have the appropriate physical and performance characteristics may be used in properly designed, operated and maintained WTGSs where condensate or direct water ingestion is minimal.

Some ester based gear oils may be suitable in water protection areas since they may be biodegradable oils.

F.2.2.2.1 Ester based lubricant compatibility

Ester based gear oils and PAO lubricants with ester additives may adversely affect filters, elastomeric seals, adhesives, sealants, paint, and other surface treatments such as layout lacquer used for contact pattern tests. Therefore, lubricants with esters should be tested for compatibility with all gearbox components before they are used in service.

F.2.2.3 Polyalkyleneglycol, PAG, lubricants

PAG based oils have a chemical structure that is distinctly different from both PAO and ester based oils. PAG based gear oils can have excellent thermal and oxidative stability and most have exceptionally high viscosity indices. PAG lubricants are characterized as being either water miscible or oil miscible. Water miscible PAG should not be combined with mineral oil, PAO, or oil miscible PAG. Consult the lubricant manufacturer for PAG miscible characteristics. Special flushing procedures are required when switching between a PAG and mineral or PAO lubricant. Consult lubricant manufacturers for specific details.

A consideration with PAG lubricants is that they may require different specification for paints, seals, and sealants.

PAG based gear oils have been used successfully in properly designed, operated, and maintained WTGSs since the late 1980’s where condensate or direct water ingestion has been minimal.

F.3 Additive systems

All commercial gear lubricants contain additives that enable them to meet specific performance requirements. Typical additives are:

- rust inhibitor;
- oxidation inhibitor;
- antifoam;
- antiwear; and
- antiscuff.

F.3.1 Standard additives

Most lubricants have rust and oxidation (R&O) inhibitors. Mineral or synthetic lubricants containing all the additives listed in F.3 except antiscuff and possibly antiwear additives are referred to as R&O oils. They are not intended for applications where boundary lubrication is expected. Most wind turbine gearboxes contain gears that encounter the boundary lubrication regime. Therefore, R&O lubricants are not recommended.

F.3.2 Antiscuff additives

Mineral and synthetic lubricants that contain additives to prevent scuffing under boundary lubrication are commonly referred to as extreme pressure (EP) lubricants. Antiscuff additives may contain sulfur, phosphorus or other soluble compounds that form a protective film on the mating component surfaces to prevent scuffing.

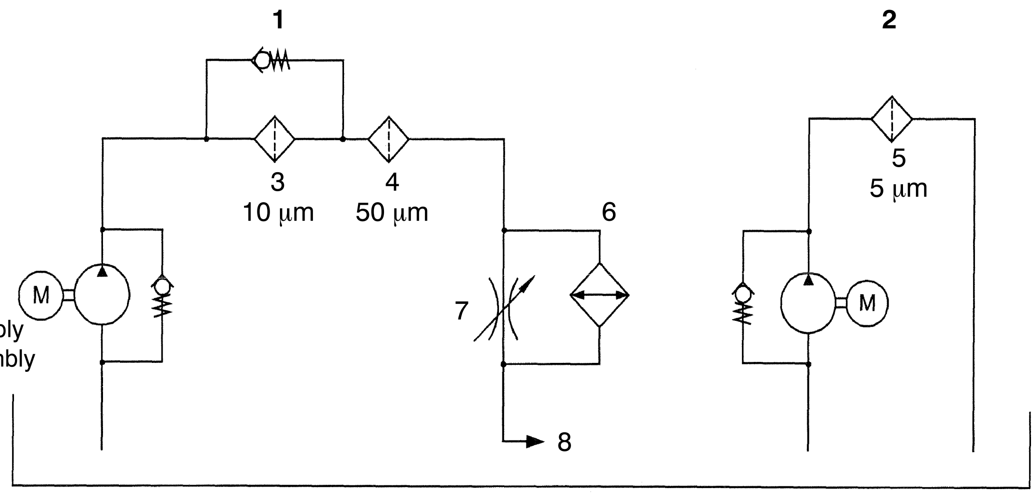
F.4 Maintaining lubricant cleanliness, active lubrication

Properly designed, installed and maintained mechanical filter systems have been found to adequately maintain wind turbine gearbox lubricating oil cleanliness levels as specified in 6.8. Depending on the operating environment, semi-annual or longer servicing intervals can be achieved. For information on oil cleanliness, see [41].

F.4.1 Configuration of filtration systems

Wind turbine gearbox mechanical filtration systems can utilize one or more mechanical filter assemblies. It is useful to combine the filtration system with the cooling system, see figure F.1. Generally, these include:

- an optional suction strainer at the oil entrance end of the pump inlet line to prevent the ingestion of large debris particles into the pump;
- an inline filter assembly adequately sized to minimally restrict the flow of oils up to ISO viscosity grade 460, VG 460, at the volumes required to lubricate the gears, bearings and other components;
- an offline filter assembly (recommended);



Key
1 Inline filter assembly
2 Offline filter assembly
3 Inline filter
4 Pressure strainer
5 Offline filter
6 Cooler
7 Thermo bypass valve
8 To the lubrication sys-

Figure F.1 - Example for circuit design, combination of filtration and cooling system

- a pressure strainer (recommended) in the oil distribution system to capture large particulate matter in the event of inline filter element failure, cold start conditions, and clogged inline filter.

F.4.2 Suction strainer

The suction strainer is a protective device for the oil pump. It is submerged in the bulk oil in the lowest section of the gearbox sump or oil reservoir. It usually consists of a metallic open body and a wire mesh element. The wire mesh should be sufficiently sized dimensionally and in pore size to provide unrestricted flow of oil to the pump suction port. Stainless steel is the preferred material of construction. The suction strainer is an optional device and usually not recommended by system designers. The suction strainer may clog and close off the lubricant supply. Pumps may be damaged by cavitation. For these reasons, a bypass valve in the strainer is recommended as well as designing the system for easy maintenance.

F.4.3 Inline filter

The inline filter assembly, sometimes referred to as an inline filter, should be sized appropriately to:

- provide the minimum restricted flow of the required volume and pressure of oil to the lubricated components during turbine operation;
- prohibit flow restriction on cold startup that may damage or collapse the filter element;
- provide dirt retention capacity capable of achieving seven months of service when semi–annual filter element change cycles are desired, and fourteen months of service when annual filter element change cycles are required.

When choosing the filter micron rating, one should know the cleanliness class requirements (see table 17) and the system properties such as dirt ingression rate, flow rate, and oil properties. A typical value is 10 micrometers with a beta ratio greater than or equal to 200 ($\beta_{10(c)} \geq 200$). The index (c) refers to the certified micron rating per ISO 16889:1999. The filter assembly should include an automatic oil bypass to permit adequate oil flow to protect the lubricated components during cold startup and as the filter media becomes fouled with contaminants. Sensors should be installed to monitor the clogging status and oil flow of the filter assembly. This may include:

- pressure loss across the filter element;
- the status of the bypass device;
- positive oil flow or pressure downstream of the filter element.

F.4.4 Offline filter

The filter element should be sized to:

- prohibit flow restriction during initial operation that is sufficient to damage or collapse the filter element;
- have a contaminant retention capacity capable of providing seven months of service when semi–annual filter element change cycles are desired, and fourteen months of service when annual filter element change cycles are required.

The selected filter micron rating should be appropriate to achieve the cleanliness class requirements and the system properties (for example, dirt ingression rate, flow rate, oil properties). A typical value for the micron rating is 5 micrometers with a beta ratio greater than or equal to 200 ($\beta_{5(c)} \geq 200$). The filter assembly should include an automatic oil bypass. Sensors should be installed to monitor the clogging status of the filter assembly. This may include:

- pressure loss across the filter element;
- the status of the bypass device.

Optionally, offline filtration may be utilized to absorb free water in the oil.

F.4.5 Pressure strainer

Optionally, a metal mesh strainer assembly may be installed in the pressurized oil distribution line down stream of the inline filter assembly. The function of this device is to prevent large particles from entering the oil distribution system in event of a inline filter element mechanical failure, such as collapse. A stainless steel wire mesh element with openings of 50 micrometers is generally acceptable. The strainer housing and element should be dimensionally correct to minimize the restriction of oil flow during cold startup and normal gearbox operation. It is important to prevent damage to the lubrication system due to insufficient flow past a clogged pressure strainer. Suitable measures to avoid such a situation include, but are not limited to, adding a bypass valve around the strainer or adding a pressure or flow switch or transducer to indicate clogging has occurred.

F.4.6 Filter and gear oil compatibility

The filter assemblies shall be sized appropriately and manufactured from materials that are chemically compatible with the lubricating oil. They should not have a detrimental effect on the long term performance of the lubricant or lubrication system. This includes:

- hardening, softening or degradation of the filter media or sealing components;
- removal or alteration of additives such as antifoam agents;
- contributing to electrophoresis (static electricity buildup within the lubrication system).

F.4.7 Coolers and heaters

Coolers and heaters are used to maintain a proper oil temperature for startup and operation of the gearbox. These devices may be installed in the main lubrication system, offline, or a combination of both. Devices may be added to limit the pressure drop across coolers and heaters.

The components, if incorporated into the main lubrication system, must be sized appropriately to provide sufficient oil flow to the gears and bearings. All materials of these devices shall be compatible with the lubricant.

F.5 Condition monitoring

Condition monitoring can provide useful information on the health of gears and bearings, and detect contamination or other adverse changes of the lubricant. After the initial 72-hour and 1000-hour sampling recommended in 6.7, sampling and analysis should be done after each 4000 hours of operation. The results of this analysis will be decisive for determining the frequency of further sampling and analysis. At each evaluation for further use of the oil, the next operation interval, including duration and season, must be considered. The first oil sampling interval can be increased only if long term experience with the specific oil and the specific site is available.

F.5.1 Lubricant sampling

Whenever samples are taken, it is important that the same procedures be followed so that consistent samples are obtained. Once the monitoring program has begun, do not change sampling procedures or sampling points.

F.5.1.1 Sampling techniques

Always use clean, lubricant compatible plastic or glass sample bottles and caps, and keep all sampling equipment scrupulously clean.

Prior to sampling, fill out the label completely and attach it to the sample bottle. Be sure to record the sample point and the date.

Thoroughly clean the area around the sampling port or valve before opening. A ball type sampling valve should be threaded into the port most accessible on the installed WTGS. The unused ports should be closed with appropriate threaded and sealed plugs. A flexible tube (preferably carbon steel or stainless steel) of sufficient length to reach a sample bottle should be attached to the sampling valve outlet port.

A plastic slip on or metal threaded cap should be attached to the open end of the sampling tube when it is not in use.

F.5.1.2 Sampling from oil drums

For monitoring the quality of fresh lubricant, the sample should be taken from the lubricant drum. The procedure described below assumes that the objective is to test the general condition of the oil in the drum. Therefore, the sample is a mixture of oil taken from the top, middle and near the bottom of the drum. If the objective is to test for water contamination, sludge or sedimentation, the sample should be taken from the lowest point of the drum.

- Use a manual suction pump to draw the oil sample into the sample bottle.
- Sample from the top, middle and near the bottom of the drum in order to obtain a representative sample and avoid stratification. Use a sampling rod and attach the sampling tube to the rod with plastic ties. Position the sampling tube a few inches from the bottom of the rod to prevent the end of the tube from touching the sides and bottom of the drum. Take 1/3 of the sample with the rod touching the bottom of the drum. Take another 1/3 of the sample with the rod raised to mid-height of the oil level. Take the final 1/3 of the sample with the rod raised so that the sampling tube is just below the surface of the oil.
- Discard the sampling tube to avoid contaminating subsequent oil samples.

F.5.1.3 Sampling from the gearbox

For monitoring gearbox health, the sample should be taken from the gearbox sampling port while the oil is still warm. Discard any oil in the sampling port that may have been stagnant. Do this by turning on the valve, capturing the oil to be discarded in a separate bottle, and without touching the valve, obtain the sample to be analyzed in a fresh sample bottle.

F.5.2 On-site analysis

There are several simple tests that can be performed on-site, and at low cost, to check for contamination or oxidation of the lubricant. The tests should be performed by the same person each time, since the tests require experience to accurately judge the results. The tests can be run as often as necessary, but they should be done every time samples are taken for laboratory analysis.

F.5.2.1 Appearance test

The simplest test is visual appearance. Often this test will disclose problems such as gross contamination or oxidation.

Look at the lubricant in a clean, clear bottle. A narrow, tall vessel is best. Compare the sample with a sample of new, unused lubricant. The oil should look clear and bright. If the sample looks hazy and cloudy, or has a milky appearance, there may be water present. The color should be similar to that of the new oil sample. A darkened color may indicate oxidation or contamination with fine wear particles. Tilt the bottle and observe whether the used oil appears more or less viscous than the new oil. A change in viscosity may indicate oxidation or contamination. Look for sediment at the bottom of the bottle. If any is present, run a sedimentation test.

F.5.2.2 Odor test

Carefully sniff the oil sample. Compare the smell of the used oil sample with that of new oil. The used oil should smell the same as new oil. Oils that have oxidized have a “burnt” odor, or smell acrid, sour or pungent.

F.5.2.3 Sedimentation test

If any sediment is visible during the appearance test, a simple test for contamination can be performed on site as follows: place a sample of oil in a clean, white cup made from a non-porous material that is compatible with the lubricant, cover and allow it to stand for two days. Carefully pour off all but a few milliliters of oil. If any particles are visible at the bottom of the cup, contaminants are present. Resolution of the unaided eye is about 0.040 millimeters. If particles respond to a magnet under the cup, iron or magnetite wear fragments are present; if they don’t respond to the magnet, and the solids feel gritty between the fingers, they are probably sand. If another liquid phase is visible, or the oil appears milky, water is probably present.

F.5.2.4 Crackle test

If the presence of water is suspected in an oil sample, a simple test can be performed on site as follows: drop a small drop of oil onto a hot plate at 135°C. If the sample bubbles, water is above 0.05%. If the sample bubbles and crackles, water is above 0.1%. When carrying out the crackle test, the rules of health and safety must be taken into consideration, for example, wearing of eye protection.

F.5.2.5. Automatic Particle Counting

Particle counting is a useful tool for on-site analyses (see section about particle counting F.5.3.5).

F.5.3 Laboratory analysis

F.5.3.1 Viscosity

The ASTM D445 (ISO 3104) test is an accurate, widely accepted method for determining kinematic viscosities of lubricants. It measures the time for a fixed volume of oil to flow through a capillary viscometer under accurately reproducible head at closely controlled temperatures. Viscosities are then calculated from the measured flow time and the calibration constant of the viscometer. Units for kinematic viscosity are mm^2/s, but they are commonly referred to as centistokes, cSt.

Viscosity is usually measured at 40°C and 100°C. Viscosities at other temperatures can be determined by plotting the two points on special log paper (ASTM D341). Viscosity index, VI, is a means of expressing the variation of viscosity with temperature. VI is calculated from the measured viscosity at 40°C and 100°C using ASTM Method D2270 (ISO 2909).

An increase in viscosity over that of fresh oil can be caused by oxidation or by contamination with dirt or water. A decrease in viscosity can be caused by contamination with a solvent or fluid of lower viscosity, or from mechanical shearing action on polymeric components that may be used in the formulation.

F.5.3.2 Acid number

The standard test for acid number, AN, is ASTM D664 or ISO 6619:1998. The test uses potassium hydroxide, KOH, to neutralize the acidic constituents in the oil. It yields a single number that represents the amount of KOH used for a given sample of oil in units of milligrams of KOH per gram of oil.

When tested, most new, unused oils will have an acid number because the KOH reacts with additives in the oil. Depending on the additives, the new oil baseline AN can vary widely. Therefore, new oil should be tested to establish a baseline AN. With the new oil baseline AN known, any change in acidity from the new oil baseline can be monitored.

The AN is a measure of the acidity of an oil sample. The higher the AN, the more acidic constituents are present. The acids usually form when high temperatures cause the oil to oxidize. The oxidation may be promoted by contaminants such as water, or wear debris such as iron and copper, that act as catalysts.

Oil oxidation is detrimental because it may increase viscosity, change color and odor, cause residues and sludge, and create acids that promote corrosion.

F.5.3.3 Water content

F.5.3.3.1 Test methods

There are several tests for determining water content in lubricants. They are listed below in order of increasing accuracy:

- Crackle test. A simple test for water contamination is described in F.5.2.4.
- Distillation test, ASTM D95 (ISO 3733). The distillation test is usually used on oils that prove to be positive by the crackle test and require a more accurate determination of water content. The test is a simple distillation of the oil and separation of the water. It detects water at levels of 0.1% (1000 ppm) or greater with reasonable precision.
- Infrared analysis. Infrared spectroscopy is sometimes used when water is present at levels above 0.05% (500 ppm). A baseline new oil reference is recommended for comparisons. The presence of additives depending on their chemical functionality may limit accurate interpretation of the results.
- Karl Fischer test (ASTM D6304). The Karl Fischer titration test determines water content from the chemical reaction between a reagent and the water in the oil sample. It detects water as low as 0.001% (10 ppm). It is commonly used because it is accurate and relatively inexpensive. However, sulfur antiwear or antiscuff additives may interfere with the test and give erroneous values for water content.

F.5.3.3.2 Effects of water contamination

The following potential effects could result with elevated levels of water in mineral and PAO based lubricants. For other lubricant types, such as PAG and esters, one should consult the lubricant manufacturer.

Free and emulsified water are considered to be more harmful than dissolved water. Therefore, moisture levels should be kept below the saturation point, which for many oils is less than 500 ppm depending on oil type and temperature. Water in excess of 500 ppm may contribute to:

Lubricant degradation:

– accelerated additive depletion;

– accelerated oxidation;

– interfere with an active lubricant film formation;

– contributes to foaming.

Degradation of internal components:

– reactions with additives may form residue on critical surfaces and plug filters or clog spray nozzles;

– reaction with the base fluid or additives may promote the hardening of elastomers or the premature failure of internal coatings such as paints.

Corrosion of metallic components:

– reaction with the base fluid where additives may increase acidity;

– direct contact with metal surfaces can produce rust particles that contribute to abrasive wear and act as an oxidation catalyst.

Accelerated metal fatigue:

– corrosion etch pits may initiate fatigue cracks;

– under specific conditions, may lead to hydrogen embrittlement that promotes propagation of fatigue cracks.

F.5.3.4 Spectrochemical analysis

F.5.3.4.1 Description

This test detects microscopic size metal particles in an oil sample. The typical spectrometer is capable of identifying about 20 metals, the source of which may be wear debris, contaminants, or inorganic additives in the lubricant. Knowing the type and quantity of metals can help diagnose wear problems or disclose sources of contamination. For example, a high concentration of iron, chromium, manganese, molybdenum or nickel could indicate wear debris from gear teeth or bearings.

Spectrochemical analysis is rapid and inexpensive. The oil sample is burned and the light emitted is separated by diffraction into distinct wavelengths. Because each metal has its own characteristic wavelength, specific metals in the oil sample can be identified.

Table F.1 is a general list of elements that may be detected by spectrochemical analysis. It lists typical sources for each element.

Table F.1 - Sources of metallic elements

Element	Symbol	Typical source
Aluminum	Al	Dirt, labyrinth seals
Antimony	Sb	Journal bearings, grease, additives
Arsenic	As	Journal bearings
Barium	Ba	Water, grease, additives
Bismuth	Bi	Journal bearings
Boron	B	Additives
Cadmium	Cd	Journal bearings, plating
Calcium	Ca	Additives, water, grease, dirt
Chromium	Cr	Gears, bearings, shafts
Cobalt	Co	--
Copper	Cu	Bearings, coolers, additives
Indium	In	Solder
Iron	Fe	Gears, shafts, bearings, rust
Lead	Pb	Journal bearings, solder, grease, paint
Magnesium	Mg	Dirt, additives
Manganese	Mn	Gears, bearings, shafts
Molybdenum	Mo	Gears, bearings, shafts, additives
Nickel	Ni	Gears, shafts, bearings
Phosphorus	P	Additives
Potassium	K	Dirt
Silicon	Si	Additives, dirt, sealants
Sodium	Na	Additives, dirt
Tin	Sn	Bearings, solder, coolers
Titanium	Ti	Paint, dirt
Vanadium	V	--
Zinc	Zn	Additives, coolers, brass components

F.5.3.4.2 Limitations

Emission spectroscopy works well for detecting metal particles up a few micrometers. Low results are obtained for particles greater than a few micrometers because it is incapable of completely and consistently burning large particles. Therefore, it readily detects particles from mild adhesive wear, polishing and micropitting because wear debris from these wear modes are within the detectable range. However, failure modes such as severe abrasion, macropitting or scuffing usually generate particles that are larger than 10 micrometers. In such situations, ferrography, particle counting or analysis with a ferrous debris analyzer may be superior monitoring techniques.

Emission spectroscopy does not distinguish between particles of free metal and particles of metal oxides or other compounds of metal. For example, rust particles may show up as increased iron content, but emission spectroscopy cannot identify whether the iron is in the form of wear debris, iron oxide, or iron sulfide. High concentrations of silicon or aluminum may indicate contaminants such as sand, dust or dirt. However, there are other sources for silicon, such as silicone antifoam additives or silicone gasket sealants. It is therefore important to analyze samples of fresh oil from new oil drums to establish a baseline level of silicon to help distinguish between contaminants and lubricant additives.

Some spectroscopy does not detect sulfur and certain other elements. Specific elements of interest should be requested from the laboratory processing the test.

It is helpful to plot the results of spectrochemical analyses over time. The graphs will indicate the normal test variability and will help one follow any trends in test results. An accelerating wear problem is most easily predicted from a trend line that is increasing.

F.5.3.5 Automatic particle counting

Particle counters are a common method used to determine lubricant cleanliness. They monitor the number of particles of a given size range in a given volume of oil sample or a given oil flow. A common method for determining lubricant cleanliness, particle counting, detects all particles regardless of their composition and is capable of detecting particle sizes in the range of 0.5 to 100 micrometers, or greater.

Particle counters may use light-interruption, laser-scanning, induction sensors or conductivity measurements. These methods provide analyses that are rapid and inexpensive. Light-interruption and laser-scanning techniques detect all particles regardless of their composition, whereas induction and conductivity sensors allow discrimination between ferrous and non-ferrous particles. Theoretically, all methods are capable of detecting particles in the range of 0.5 to 100 micrometers, or greater. In practical applications the sensitivity, however, may be reduced due to high oil viscosity, oil discoloration (additives, oxidation), opaqueness (water) or air bubbles.

With light-interruption particle counters, the lubricant flows through a small passage while a light beam scans the oil through a window. Particles in the oil that are within a set size range momentarily interrupt the light beam. The output from a detector that senses the interruption of the light beam is related to the time of interruption and hence the size of the particle.

Laser-scanning particle counters operate on a principle that is similar to that of the light-interruption type, except the oil sample remains stationary in a clear glass container while it is scanned by a revolving laser beam. The particle size range is selectable, as it is with the light-interruption method.

F.5.3.5.1 Limitations of automatic particle counting

While particle counting detects all particles, it gives no information on the shape or composition of the particles. It is susceptible to incorrect particle counts caused by including bubbles of air or water. The oil sample must not be opaque. For accurate results, the oil sample should be well agitated to produce a uniform suspension of particulates, and the concentration of particles should be low enough to avoid counting two or more particles as one.

F.5.3.5.2 ISO solid contamination code

The International Organization for Standardization, ISO, Solid Contamination Code, ISO 4406 has been universally accepted as the simplest and best means for expressing cleanliness levels. It has been adopted by the Society of Automotive Engineers (SAE J1165).

The ISO 4406 code changed in 1999. The revised system uses three code numbers, corresponding to concentrations of particles larger than 4, 6, and 14 micrometers. The new 6 and 14 micrometer sizes were chosen so code numbers would not change significantly from the older system based on 5 and 15 micrometer sizes. For higher viscosity oils such as gear oils, the number of particles lesser than or equal to 4 micrometers is generally not reported, the value being substituted with a "–", for example –/15/12. Some companies use a three digit form of the ISO 4406:1987 code representing 2, 5, and 15 micrometer size particles. This three digit code can be upgraded to the new ISO 4406:1999 system by increasing the first digit by one while keeping the last two digits the same. For example, 17/15/12 under

ISO 4406:1987 becomes 18/15/12 under ISO 4406:1999.

There are two ways for assigning the ISO code. In the first method, the range numbers are selected from a table of range numbers versus particle concentration (see table F.2) for the number of particles greater than 4, 6 and 14 micrometers per milliliter. If a particle count falls between adjacent particle concentrations, the ISO range number is found opposite the higher concentration. In the second method, the particle counts are plotted on graph paper and the range numbers are determined where the line crosses the 4, 6 and 14 micrometers vertical lines.

Table F.2 - What the ISO codes mean

ISO number	Number of particles per milliliter of fluid
25	160 000 to 320 000
24	80 000 to 160 000
23	40 000 to 80 000
22	20 000 to 40 000
21	10 000 to 20 000
20	5000 to 10 000
19	2500 to 5000
18	1300 to 2500
17	640 to 1300
16	320 to 640
15	160 to 320
14	80 to 160
13	40 to 80
12	20 to 40
11	10 to 20
10	5 to 10
9	2.5 to 5
8	1.3 to 2.5

NOTES:
ISO code examples:
- –/21/18 Dirty system
- –/17/14 New oil
- –/16/13 Average system (inline filter)
- –/13/10 Clean system (offline filtration)

As an example, suppose the following particle counts are obtained:

Particle size, μm(c)	Particles per milliliter
> 4	1617
> 6	78
> 14	17
> 50	3
> 100	0

Since there are 1617 particles per milliliter greater than 4 micrometers, the first range number given by table F.2 is 18. The 78 particles per milliliter greater than 6 micrometers gives a range number of 13. The 17 particles greater than 14 micrometers give a range number of 11. Therefore, the ISO code number for the sample is 18/13/11.

F.5.3.6 Ferrographic analysis

Ferrographic analysis separates wear debris and contaminants from a lubricant sample by magnetic precipitation. It is capable of precipitating particles that range from less than 1 to 100 micrometers, or greater. Ferrography provides two types of analysis, a relatively inexpensive direct-reading, DR, ferrograph, and a more expensive analytical ferrograph.

In the DR ferrograph, a diluted sample of lubricant is siphoned through a precipitation tube that resides in a powerful magnetic field. The combination of magnetic force and the viscous forces exerted by the lubricant causes the particles to be separated according to size. The large particles, greater than 5 micrometers, are deposited first, near the entry of the tube, then the smaller particles, 1 to 2 micrometers, deposit farther down the tube. Two light beams pass through the precipitation tube, one at the entry deposit and one several millimeters farther down the tube where small particles deposit. Light attenuation at the two locations along the tube is used to quantify the relative amount of large to small particles. The results are reported as two scalar readings, “direct large”, DL, and “direct small”, DS.

In the analytical ferrograph, a diluted sample of lubricant is pumped across a microscope slide that is mounted at an angle above a magnet so that the field varies along the length of the slide. The particles are subjected to a continuously increasing magnetic force as they flow along the slide. Consequently the particles precipitate, distributed according to size, along a narrow band about 50 millimeters long. Ferrous particles line up in strings that follow the magnetic lines. Nonferrous particles and contaminants travel down the field in a random pattern. The slide is washed with a fixative that washes away the oil, locks the particles in place, and floats away other material.

The ferrogram (slide upon which particles have been deposited) is examined in a bichromatic microscope equipped with a camera. The microscope uses both transmitted green light projected from the bottom of the ferrogram and red light reflected from the top of the ferrogram to distinguish the size, shape, texture, and composition of both metallic and nonmetallic particles. The particles have characteristics that help determine the wear mechanism and identify the source of the particles. Table F.3 classifies the types of particles.

F.5.3.7 Wear particle analyzer

In the ferrous debris analyzer, a diluted sample of lubricant is drawn through a filter that is in a strong magnetic field. The filter has a matrix of fine, ferromagnetic fibers that become magnetized in the magnetic field. The fibers capture small particles magnetically, and physically capture particles larger than the spacing between the fibers.

A flux sensor determines the change in the magnetic field due to the presence of the particles, and displays the magnetic equivalent of the captured particles in micrograms of iron metal. The reading, known as the magnetic iron content, MIC, is independent of particle size.

The ferrous debris analyzer is capable of capturing particles 1 micrometer and larger with an efficiency of 95% or greater.

The filter can be back washed with solvent to recover the particles for microscopic examination and other diagnostic analyses.

Table F.3 - Characteristics of particles

Wear particles	
Rubbing	Flat platelets <15μm long, <1μm thick. Found in lubricants of all machines, and are indicators of normal wear.
Sliding	Generally >15μm with length to thickness ratio between 5 and 30. Surfaces have parallel striations and may have temper colors. They are evidence of severe sliding.
Cutting	Long, curled chips resembling lathe cuttings. They are evidence of abrasive wear. May be caused by contamination of lubricant by hard, sharp edged particles.
Fatigue	Generally >5μm with length to thickness ratio <5. Surfaces are rough and particles are shaped like chunks of coal, rather than platelets.
Laminar	Thin, bright particles often with holes in their surfaces and edges that are split. Length to thickness ratio >30. Typical of gear and rolling bearing wear particles that pass between contacting surfaces.
Spherical	Generally <5μm diameter with smooth surfaces. They are a precursor to rolling bearing fatigue failure.
Ferrous oxides	
Red oxide	Red/orange particles magnetically aligned. Appear thick, rounded and translucent. They are severe sliding wear particles that have oxidized.
Dark metallo-oxide	Resemble red oxide sliding wear particles, except they are not translucent. Often show flecks of free metal on their surfaces. They are caused by heat and may be evidence of lubricant starvation or severe wear.
Black oxide	Dark gray/black particles magnetically aligned. Shaped like pebbles. They are evidence of inadequate lubrication and represent a more severe condition than red oxide particles.
Corrosion	Fine deposit of <1μm size particles at the exit of ferrogram. They are formed by corrosive attack of metal surfaces and depletion of lubricant additives.
Contaminants	
Friction polymer	Amorphous, translucent material of no particular size. Indicates lubricant polymerization under extreme conditions.
Sand, Dirt	Generally >5μm crystalline particles, not magnetically aligned.
Fibers	Translucent, fibrous particles not magnetically aligned. Typical fibers include hair, cotton, wood, glass, minerals, nylon and cellulose.
Spheres	Generally >5μm with rough surfaces. May be contaminants from grinding, welding or shot blasting.
Other	Contaminants such as paper, paint, varnish, glue, gasket or seal materials, or lubricant additives such as molybdenum disulfide or graphite.

F.5.4 Recommended analysis limits

Table F.4 gives recommended limits for analysis parameters and typical contaminants for wind turbine gearboxes. In the absence of guidelines from the lubricant manufacturer, the guidelines in table F.4 should be used.

When borderline limits are reached, sampling frequency should be increased to determine if corrective action is required.

If unsatisfactory limits are reached, the cause should be determined and the situation corrected.

F.5.5 Micropitting resistance test

Because of the sensitivity of this application to micropitting, it is recommended that the lubricant be evaluated for micropitting resistance. Although not a standardized method, the FVA 54/I test (see FVA information sheet 54/7) is one recognized method to evaluate this phenomenon. Typically, a failure stage equal to 10 is considered acceptable performance.

Table F.4 - Analysis limits for wind turbine gearbox lubricants

Analysis parameter	Borderline	Unsatisfactory
Water (Karl Fischer)[1)]	0.05%	>0.10%
Sediment (see F.5.2.3)	-	visible
AN increase over fresh oil	40%[2)]	>75%[2)]
Viscosity change from ISO VG limits	10%	>20%
Iron (Fe), ppm	75-100	>200
Copper (Cu), ppm	50-75	>75
Silicon (Si) increase over fresh oil, ppm	15-20	>20
ISO 4406:1999 cleanliness (Acceptable is -/16/13)	-/17/14	-/18/15

NOTES:
1) For limits of water miscible PAG oils, consult lubricant manufacturer.
2) Values to be advised by the lubricant manufacturer.

Table F.5 - Viscosity grade[1)] at bulk oil operating temperature for oils having a viscosity index of 90[2)]

Temp °C	Pitch line velocity, m/s[3)]							
	1.0 - 2.5	**2.5**	**5.0**	**10.0**	**15.0**	**20.0**	**25.0**	**30.0**
10	32							
15	46	32						
20	68	46	32					
25	68	46	32					
30	100	68	46	32				
35	100	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	32
50	320	220	150	100	46	46	46	32
55	460	220	150	100	68	68	68	46
60	460	320	220	150	68	68	68	46
65	680	460	320	220	150	100	100	68
70	1000	680	320	220	150	100	100	68
75	1500	680	460	320	220	150	150	100
80	2200	1000	680	460	220	220	220	150
85	3200	1500	1000	460	320	220	220	150
90	3200	2200	1000	680	460	320	320	220
95		3200	1500	1000	460	460	320	220
100		3200	2200	1000	680	460	460	320

NOTES:

1) (light shaded) Consult gearbox, bearing and lubricant manufacturers if a viscosity grade of less than 32 or greater than 3200 is indicated.
(dark shaded) Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
(white) Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
(double-lined) Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

2) This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

3) Determine pitch line velocity of all gearsets. Select viscosity grade for critical gearset taking into account cold startup conditions.

Table F.6 - Viscosity grade[1)] at bulk oil operating temperature for oils having a viscosity index of 120[2)]

Temp °C	Pitch line velocity, m/s[3)]							
	1.0 - 2.5	2.5	5.0	10.0	15.0	20.0	25.0	30.0
10	32							
15	46	32						
20	68	46	32					
25	68	46	32	32				
30	100	68	46	32				
35	150	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	32
50	320	220	100	100	68	46	46	46
55	320	220	150	100	68	68	46	46
60	460	320	220	150	68	68	68	46
65	680	460	320	150	100	100	100	68
70	1000	460	320	220	150	150	100	68
75	1000	680	460	220	150	150	150	100
80	1500	1000	460	320	220	220	150	100
85	2200	1000	680	460	220	220	220	100
90	2200	1500	1000	460	320	320	220	150
95	3200	2200	1000	680	320	320	320	220
100		2200	1500	680	460	460	320	220

NOTES:

1) Consult gearbox, bearing and lubricant manufacturers if a viscosity grade of less than 32 or greater than 3200 is indicated.
Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

2) This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

3) Determine pitch line velocity of all gearsets. Select viscosity grade for critical gearset taking into account cold startup conditions.

Table F.7 - Viscosity grade[1)] at bulk oil operating temperature for oils having a viscosity index of 160[2)]

Temp °C	Pitch line velocity, m/s[3)]							
	1.0 - 2.5	2.5	5.0	10.0	15.0	20.0	25.0	30.0
10	32	32						
15	46	32	32					
20	68	46	32					
25	68	46	32	32				
30	100	68	46	32				
35	150	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	
50	220	150	100	68	46	46	46	32
55	320	220	150	100	68	68	46	32
60	460	220	150	100	68	68	68	46
65	460	320	220	150	100	100	68	46
70	680	460	220	150	100	100	100	68
75	680	460	320	220	150	150	100	68
80	1000	680	320	220	150	150	150	100
85	1500	680	460	320	220	220	150	100
90	1500	1000	680	320	220	220	220	150
95	2200	1500	680	460	320	220	220	150
100	3200	1500	1000	460	320	320	220	150

NOTES:

1) Consult gearbox, bearing and lubricant manufacturers if a viscosity grade of less than 32 or greater than 3200 is indicated.
Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

2) This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

3) Determine pitch line velocity of all gearsets. Select viscosity grade for critical gearset taking into account cold startup conditions.

Table F.8 - Viscosity grade[1] at bulk oil operating temperature for oils having a viscosity index of 240[2]

Temp °C	Pitch line velocity, m/s[3]							
	1.0 - 2.5	2.5	5.0	10.0	15.0	20.0	25.0	30.0
10	46	46						
15	68	46	32					
20	68	68	32	32				
25	100	68	32	32				
30	100	68	32	32	32			
35	150	68	68	46	32	32		
40	150	100	68	46	32	32	32	
45	220	100	100	68	46	32	32	
50	220	100	100	68	46	46	46	32
55	320	150	150	68	68	46	46	32
60	320	150	150	100	68	68	46	46
65	460	220	150	100	100	68	68	46
70	460	320	220	150	100	68	68	46
75	680	320	220	150	100	100	68	68
80	680	460	220	150	100	100	100	68
85	1000	460	320	220	150	100	100	68
90	1000	680	320	220	150	150	100	100
95	1000	680	460	320	150	150	150	100
100	1500	1000	460	320	220	150	150	100

NOTES:

1) Consult gearbox, bearing and lubricant manufacturers if a viscosity grade of less than 32 or greater than 3200 is indicated.
Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

2) This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

3) Determine pitch line velocity of all gearsets. Select viscosity grade for critical gearset taking into account cold startup conditions.

Annex G
(informative)
General gear information

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ISO 81400-4, *Wind turbines - Part 4: Design and specification of gearboxes*]

G. Purpose

This annex has additional information about gear design and gear rating.

G.1 Gear types

There are three types of gears commonly used in wind turbine gearboxes: spur, single helical and double helical.

Depending on the application, the purchaser may want to specify the type of gear.

G.2 Gear design

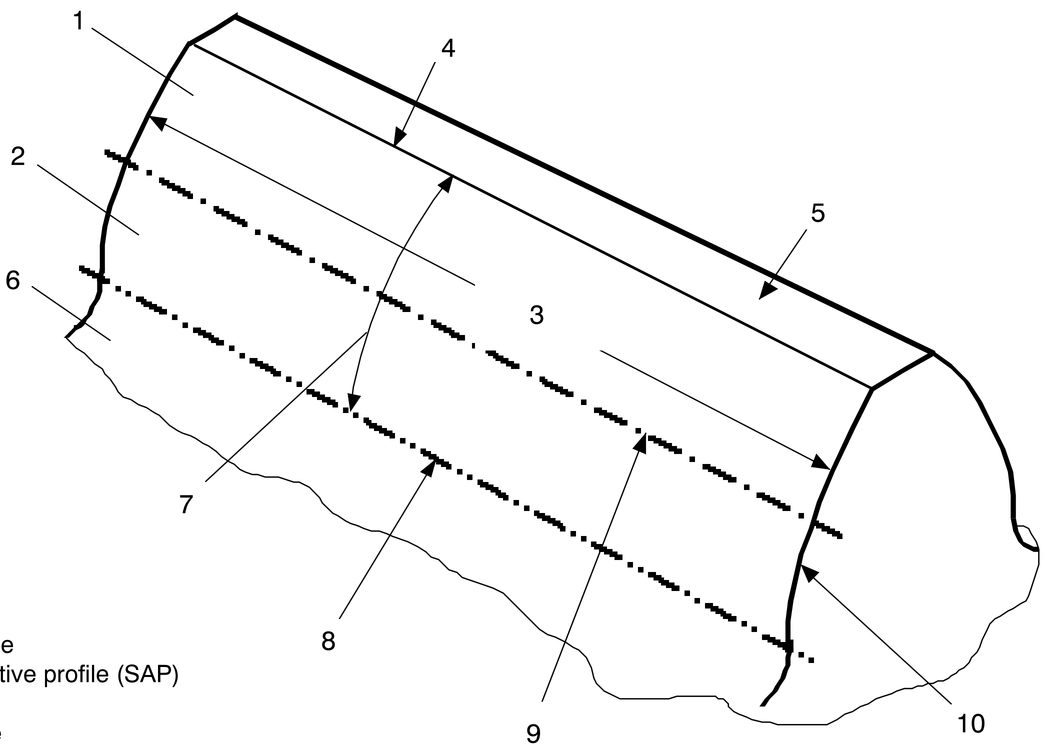
The gear geometry influences both the load capacity and the sound level of a gearbox. When properly designed and manufactured, gears operate smoothly and quietly, and have adequate load capacity to transmit the required power for the design life. See ANSI/AGMA 1012-F90 and figure G.1 for gear nomenclature.

G.3 Relative hardness

For maximum micropitting resistance, pinions should be at least 2 HRC points harder than gears. This is especially important for sun pinions. See AGMA 925-A03 for further information.

G.4 Consideration of manufacturing variation in load distribution

The load distribution along the face width of a gear mesh is a complex combination of systematic and stochastic deviations from the ideal contact geometry. The stochastic deviations result from manufacturing deviations of gears, shafts, bearings, and housing. If sufficient statistical data on different production processes are available, the resulting mesh misalignment, f_{ma}, can be determined from production variation on shaft parallelism and tooth alignment of pinion and gear.



Key
1 Addendum
2 Dedendum
3 Face width
4 Tooth tip
5 Topland
6 Root fillet
7 Active profile
8 Start-of-active profile (SAP)
9 Pitchline
10 Tooth edge

Figure G.1 - Gear nomenclature

If no statistical process control is implemented, these stochastic influences should be considered in the rating analysis by conservatively superimposing the elastic deformation with a virtual flank line end gap, $\pm f_{ma}$. The value of f_{ma} should be derived from the following simplification that approximates twice the standard deviation on the stochastic influences, hence covering about 95% of all possible tolerance combination.

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

where

$f_{\Sigma\gamma}$ is the extreme (worst-case) total shaft misalignment deviation projected into the plane of action, derived from the RMS value of worst-case initial housing alignment tolerances, $f_{\Sigma\delta}$ and $f_{\Sigma\beta}$, according to DIN 3964, fit up tolerances, and tolerances on bearing clearance. To be in line with the definition of $f_{H\beta}$, $f_{\Sigma\gamma}$ shall represent the difference between the actual alignment and zero, hence half the manufacturing tolerance zone;

$f_{H\beta,1}$ is the extreme (worst-case) tooth alignment error of the pinion according to ANSI/AGMA ISO 1328-1;

$f_{H\beta,2}$ is the extreme (worst-case) tooth alignment error of the gear according to ANSI/AGMA ISO 1328-1.

Annex H
(informative)
Determination of the application factor, K_A, from a given load spectrum using the equivalent torque, T_{eq}

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ISO 81400-4, *Wind turbines - Part 4: Design and specification of gearboxes*]

H. Purpose

The following method is useful for a preliminary gear design, where the gear geometry is not fixed. If ISO 6336-1:1996 is used, use 90% of the value of c_γ determined in the standard.

H.1 Application factor, K_A

The application factor, K_A, is defined as the ratio between the equivalent torque and the nominal torque.

$$K_A = \frac{T_{eq}}{T_n} \qquad \text{(H.1)}$$

where

T_{eq} is equivalent torque;

T_n is nominal torque corresponding to rated power and nominal speed.

The application factor, K_A, has to be determined for tooth root breakage and pitting resistance, both for pinion and wheel. The highest of these four values has to be used for a gear rating according to ISO 6336:1996.

The equivalent torque is defined by the following equation:

$$T_{eq} = \left(\frac{n_1 T_1^p + n_2 T_2^p + ...}{n_1 + n_2 + ...} \right)^{\frac{1}{p}} \qquad \text{(H.2)}$$

where

n_i is number of load cycles in bin i;

T_i is torque in bin i;

p is slope of the Wöhler-damage line.

Because equation H.2 does not clearly define how many bins of the load spectrum have to be used for the determination of T_{eq}, the procedure described in H.2 shall be used instead of equation H.2.

H.2 Determination of the equivalent torque, T_{eq}

For this procedure, the load spectrum, the slopes of the Wöhler-damage lines, p, and the number of load cycles, $N_{L\,ref}$, at the reference point must be known.

H.2.1 Methodology

The following method applies for a design case where the Wöhler-damage line is simplified by ignoring all damage that occurs at stresses below some limit stress. It is based upon the fact that while the position of the endurance limit in terms of stress is not known in relation to the gear until the design is available, the position of that endurance limit in terms of cycles does not change as the gear design changes.

Furthermore, a torque T_i in the bin i can be replaced by a torque T_j in a new bin j in such a way that the damage caused by the torque T_i is the same as that caused by the torque T_j. This is shown in figure H.1 and can be expressed by equation H.3.

$$T_i^p \, n_i = T_j^p \, n_j \qquad \text{(H.3)}$$

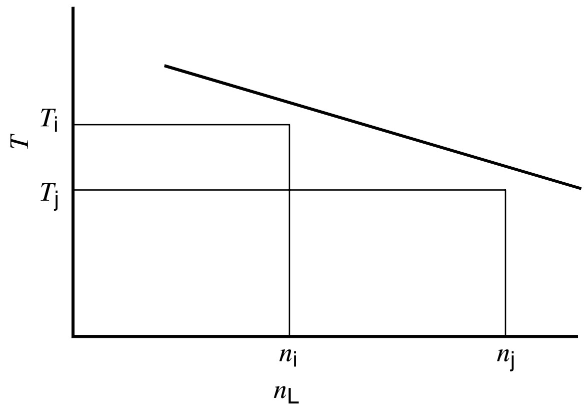


Figure H.1 - Load bins with equal damage behaviour according to equation H.3

H.2.2 Calculation procedure

The load bins have to be denoted as (T_i, n_i) and numbered in descending order of torque, where T_1 is the highest torque.

Then the cycles n_1 at torque T_1 are equivalent in terms of damage to a larger number of cycles n_{1a}, at lower torque T_2, where according to equation H.3,

$$n_{1a} = n_1 \left(\frac{T_1}{T_2}\right)^p \tag{H.4}$$

If $n_{2e} = n_2 + n_{1a}$, then bins 1 and 2 can be replaced by a single bin (T_2, n_{2e}), see figure H.2.

Similarly, the cycles n_{2e} at torque T_2 are equivalent to n_{2a} at T_3, where

$$n_{2a} = n_{2e} \left(\frac{T_2}{T_3}\right)^p \tag{H.5}$$

Writing $n_{3e} = n_3 + n_{2a}$, then bins 1, 2 and 3 can be replaced by a single bin (T_3, n_{3e}).

This procedure has to be stopped, when n_{ie} reaches the endurance limit cycles $N_{L\ ref}$.

The required equivalent torque T_{eq} is now bracketed:

$$T_i < T_{eq} < T_{i-1} \tag{H.6}$$

or

$$\frac{T_i}{T_n} < K_A < \frac{T_{i-1}}{T_n} \tag{H.7}$$

and can be found by linear interpolation on a log-log basis.

The slope exponent, p, and the endurance limit cycles, N_L, are a function of the heat treatment. Values to be used in equations H.4 and H.5 are shown in table H.1. The value of the slope exponent, p, to be used in equations H.4 and H.5 for pitting damage is half that tabulated in table H.1 to adjust the slope of the Wöhler damage line versus torque, rather than versus stress, as tabulated in ISO 6336–2:1996, table 2.

Table H.1 - Exponent p and number of load cycles $N_{L\ ref}$

Heat treatment	Pitting		Tooth root	
	p 1)	$N_{L\ ref}$	p	$N_{L\ ref}$
Case carburized, Induction hardened	6.610	5×10^7	8.738	3×10^6
Through hardened	6.610	5×10^7	6.225	3×10^6
Nitrided	5.709	2×10^6	17.035	3×10^6
Nitro-carburized	15.715	2×10^6	84.003	3×10^6

Note:
1) Values p for pitting are given for torque, to convert for stress, these values are to be doubled.

H.3 Example

An example is shown in figure H.3 and the corresponding table H.2. In the right hand column of the table a switch is shown, which indicates when the endurance limit has been reached. In this example the application factor, K_A, is between 1.16 and 1.18. From the fact that on row 12 the value of n_{ie} is very close to the endurance limit, the interpolation will give K_A = 1.18.

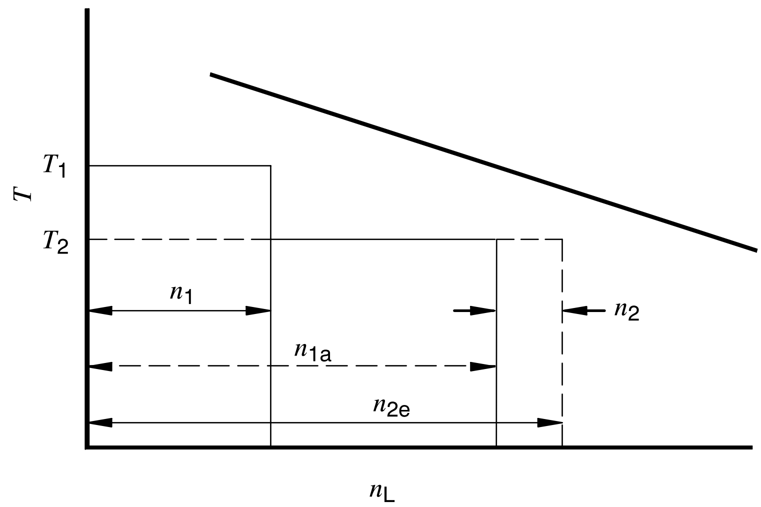


Figure H.2 - Bins (T_1, n_1) and (T_2, n_2) replaced by bin (T_2, n_{2e})

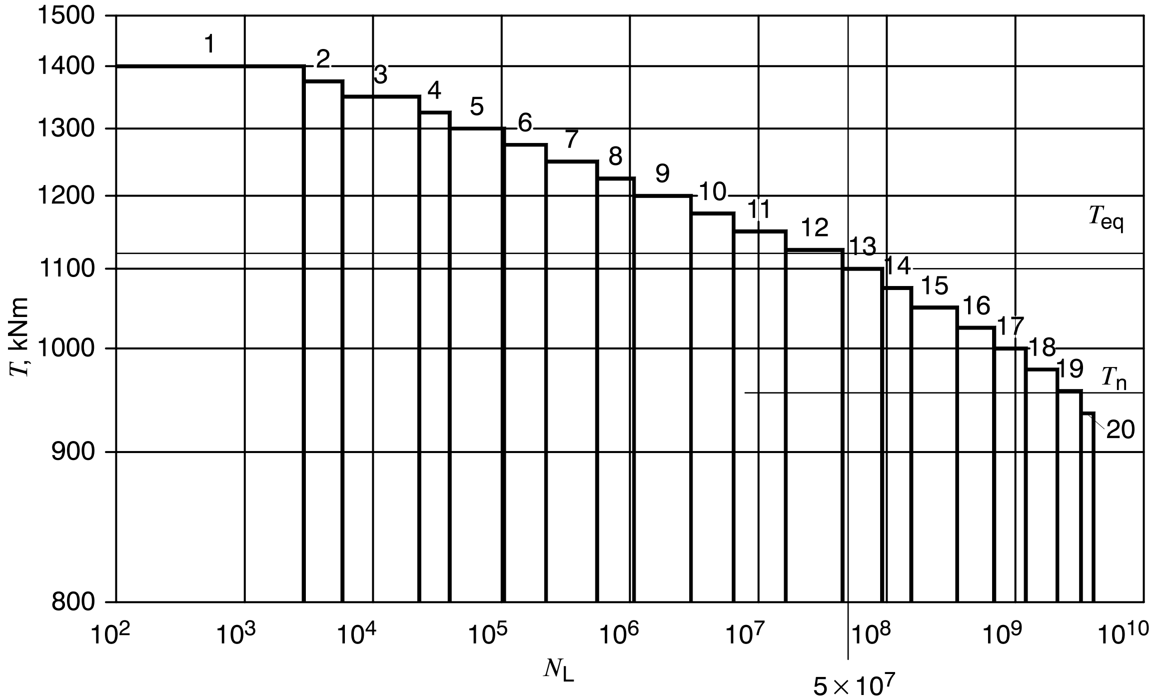


Figure H.3 - Load spectrum with corresponding equivalent torque, T_{eq}

Table H.2 - Example for calculation of K_A from a load spectrum

Cumulative damage/calculation of K_A							
Flank ratio to this gear u = 75 contacts per rev. = 1 speed = 1500 stress cycles/min				nominal torque T_n = 950 kNm blade speed n_b = 20 rpm slope exponent p = 6.6 endurance limit cycles $N_{L\ ref}$ = 5.00e+07			
	Blade torque	Torque ratio	Hours	Cycles	Equivalent from row above	Total	Switch
i	T_i	T_i/T_n	L	n_i	n_{ia}	n_{ie}	-
1	1400	1.47	0.032	2.88e+03		2.88e+03	0
2	1375	1.45	0.032	2.88e+03	3.24e+03	6.12e+03	0
3	1350	1.42	0.19	1.71e+04	6.91e+03	2.40e+04	0
4	1325	1.39	0.183	1.65e+04	2.72e+04	4.36e+04	0
5	1300	1.37	0.708	6.37e+04	4.95e+04	1.13e+05	0
6	1275	1.34	1.3	1.17e+05	1.29e+05	2.46e+05	0
7	1250	1.32	3.7	3.33e+05	2.80e+05	6.13e+05	0
8	1225	1.29	5.8	5.22e+05	7.00e+05	1.22e+06	0
9	1200	1.26	21	1.89e+06	1.40e+06	3.29e+06	0
10	1175	1.24	38	3.42e+06	3.78e+06	7.20e+06	0
11	1150	1.21	110	9.90e+06	8.30e+06	1.82e+07	0
12	1125	1.18	320	2.88e+07	2.10e+07	4.98e+07	0
13	1100	1.16	520	4.68e+07	5.78e+07	1.05e+08	1
14	1075	1.13	700	6.30e+07	1.22e+08	1.85e+08	1
15	1050	1.11	2200	1.98e+08	2.16e+08	4.14e+08	1
16	1025	1.08	3700	3.33e+08	4.85e+08	8.18e+08	1
17	1000	1.05	5800	5.22e+08	9.63e+08	1.48e+09	1
18	975	1.03	10200	9.18e+08	1.76e+09	2.67e+09	1
19	950	1.00	12400	1.12e+09	3.17e+09	4.29e+09	1
20	925	0.97	9100	8.19e+08	5.11e+09	5.93e+09	1

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Annex I
(informative)
Bearing stress calculation

[The foreword, footnotes and annexes, if any, are provided for informational purposes only and should not be construed as a part of ISO 81400-4, *Wind turbines - Part 4: Design and specification of gearboxes*]

I. Purpose

This annex provides a simplified method to estimate the maximum contact pressure in spherical, cylindrical and tapered roller bearings (SRB,CRB,TRB) operating without preload. This method is not intended to replace more advanced bearing life calculations, but rather to create transparency for advanced methods that may be proprietary to the bearing manufacturer. Precise methods for calculation of contact stress are described in [38], [39] and [40].

Text variables and calculation flow are consistent with the Mathcad® worksheet included at the end of this annex.

I.1 References

Except as noted, the methods and nomenclature are consistent with reference [42].

I.2 Influence factors

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

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

I.3 Procedure

I.3.1 Equivalent bearing load

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

$$P_o = X_o F_r + Y_o F_a \tag{I.1}$$

where

- F_r is maximum radial load, N;
- F_a is maximum axial load, N;
- X_o is radial factor, see table I.1;
- Y_o is axial factor, see table I.1.

I.3.2 Maximum rolling element load

The combined external force, P_o, is distributed over a number of rolling elements creating a statically indeterminate support system. Based on methods in reference [42] from reference [43], the load distribution may be determined from a mutual displacement of the rings and calculating the associated deformation of the rolling elements.

Single roller maximum load for a clearance free bearing:

$$Q = \frac{P_o}{Z \cos \alpha_o} k \tag{I.2}$$

Table I.1 - Factors for statically stressed radial bearings

Type of radial bearing	Single row[1)]		Double row	
	X_o	Y_o	X_o	Y_o
Spherical roller bearing	0.5	0.22 cot α_o	1	0.44 cot α_0
Cylindrical roller bearing	1	0	1	0
Tapered roller bearing	0.5	0.22 cot α_o	1	0.44 cot α_0

NOTE:
1) Equivalent load P_o must always be greater than or equal to radial load F_r.

Which yields the ratio:

$$\frac{Z\,Q}{F_r} = k \tag{I.3}$$

and k equal to 4.4, with zero internal clearance and a 180° load zone. This condition occurs when a bearing operates centered, due to the thrust to radial ratio F_a/F_r exceeding the bearing's published thrust factor e:

$$k = 4.4 \text{ if } \left(\frac{F_a}{F_r}\right) > e \tag{I.4}$$

In figure I.1, k is given for various levels of internal radial clearance G_r. k varies by the ratio:

$$\frac{F_r}{C_{\delta L}\left(\frac{G_r}{2}\right)^{1.08} Z} \tag{I.5}$$

where

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

$$C_{\delta L} = 26200\,(L_{we})^{0.92} \tag{I.6}$$

where

L_{we} is effective roller length, mm;

Z is total number of rolling elements;

G_r is internal radial clearance, mm.

The variable G_r must be greater than 0.0 to prevent numerical instability. A practical lower boundary is G_r equal to 0.0005.

For bearings operating with clearance, the following is a curve fit for k:

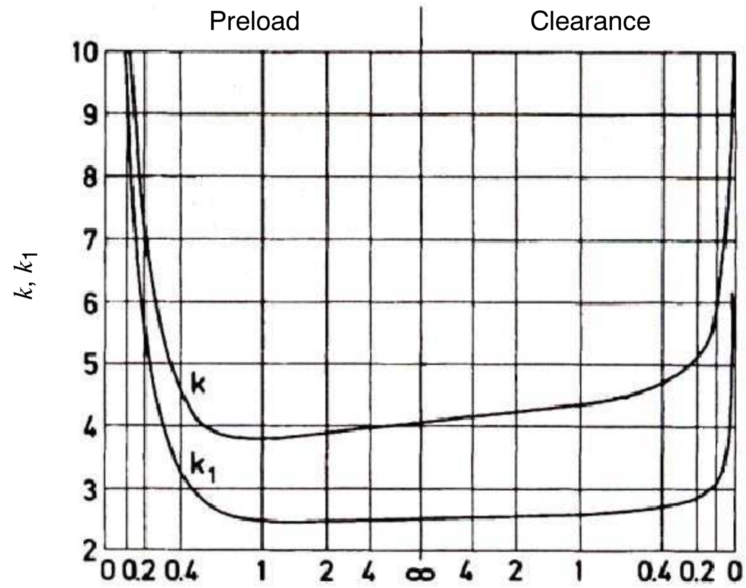
$$k = 4.05 + 0.3209\left[\frac{F_r}{C_{\delta L}\left(\frac{G_r}{2}\right)^{1.08} Z}\right]^{-0.7911} \tag{I.7}$$

The graph in figure I.1 is based on nominal contact angle α_o equal to 0.0. Introducing α_o to the equation and replacing F_r with P_o provides the general case:

$$k = 4.05 + 0.3209\left[\frac{P_o}{C_{\delta L}\left(\frac{G_r}{2}\right)^{1.08} Z\cos(\alpha_o)}\right]^{-0.7911} \tag{I.8}$$

With the elastic constant $C_{\delta L}$ and k factor known, we compute the load on the maximum loaded rolling element:

$$Q = \frac{P_o}{Z\,\cos(\alpha_o)}k \tag{I.9}$$

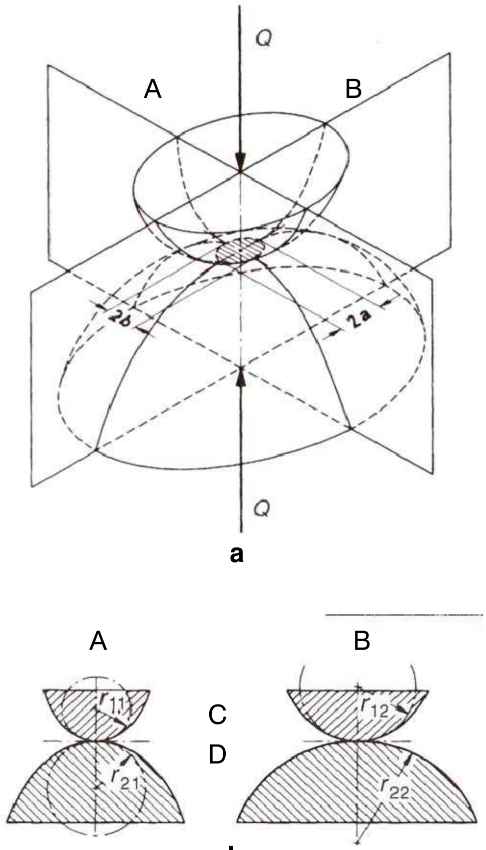


$$\frac{F_r}{C_{\delta L}\left|\frac{G_r}{2}\right|^{1.08} Z} \longrightarrow \qquad \longleftarrow \frac{F_r}{C_{\delta L}\left|\frac{G_r}{2}\right|^{1.08} Z}$$

Figure I.1 - Effects of clearance and preload on the pressure distribution in radial roller bearings

I.3.3 Determining the contact pressure factors

Figure I.2a depicts the planes under consideration, and 2b gives the nomenclature for the equation subscripts. Note that convex curvatures are positive, and concave curvatures are negative. The osculation, S, is the ratio of the radius of curvature of the inner ring to the radius of curvature of the roller, both in the principal plane 2 through the axis of the roller, see figure 2a. The value of S is equal to r_{22}/r_{12}, and greater than or equal to 1.0. Usually, S is between 1.01 and 1.03. To prevent numerical instability, the minimum value of S is set equal to 1.001.



Key
A Principal plane 1
B Principal plane 2
C Body 1
D Body 2

Figure I.2 - Nomenclature of bearing curvature

The factors are derived from the following relationships:

$$\rho_{11} = \frac{2}{D_w} \tag{I.10}$$

where

D_w is ball or roller diameter, mm;

for CRB and TRB

$$\rho_{12} = 0 \tag{I.11}$$

for SRB

$$\rho_{12} = -\rho_{22} S \tag{I.12}$$

$$\rho_{21} = \frac{2}{\dfrac{D_{pw}}{\cos(\alpha_o)} - D_w} \tag{I.13}$$

where

S is osculation at inner ring contact;

D_{pw} is pitch diameter of ball or roller set, mm;

α_o is nominal contact angle of the bearing, degrees.

for CRB and TRB

$$\rho_{22} = 0 \tag{I.14}$$

for SRB

$$\rho_{22} = \frac{-2}{\dfrac{D_{pw}}{\cos(\alpha_o)} + D_w} \tag{I.15}$$

The factors as summed as follows:

$$\Sigma\rho_{point} = \rho_{11} + \rho_{12} + \rho_{21} + \rho_{22} \tag{I.16}$$

$$\Sigma\rho_{line} = \rho_{11} + \rho_{21} \tag{I.17}$$

With the curvature factors known, along with curvature sums $\Sigma\rho_{point}$ and $\Sigma\rho_{line}$, we can determine Hertzian coefficients required to compute stress. The values of μ and ν are given in table 2.3 of reference [42], based on cos τ.

Values for μ and ν have been curve fitted using the following equations:

if cos τ > 0.87

$$\mu = 1.396748 \cos\tau^{0.665242} (1 - \cos\tau)^{-0.37399} \tag{I.18}$$

otherwise

$$\mu = \left(5.6864 \cos\tau^4 - 6.0607 \cos\tau^3 + 2.7985 \cos\tau^2 + 0.35289 \cos\tau + 1.005\right) \tag{I.19}$$

if cos τ > 0.87

$$\nu = 0.683241 \cos\tau^{0.4} (1 - \cos\tau)^{0.189343} \tag{I.20}$$

otherwise

$$\nu = -0.30365\cos\tau^3 + 0.373719\cos\tau^2 - 0.67694\cos\tau + 1.0014 \quad (I.21)$$

With μ and ν determined, the Hertz contact ellipse dimensions a and b are computed:

$$a = \frac{0.0472\,\mu\sqrt[3]{\dfrac{Q}{\Sigma\rho_{\text{point}}}}}{2} \quad (I.22)$$

$$b = \frac{0.0472\,\nu\sqrt[3]{\dfrac{Q}{\Sigma\rho_{\text{point}}}}}{2} \quad (I.23)$$

These variables are represented in figure I.3.

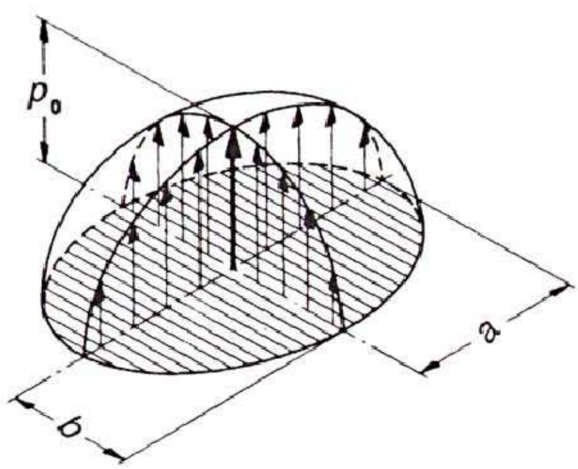


Figure I.3 - Stress distribution over the elliptical contact area

I.3.4 Contact pressure

Maximum contact pressure for SRB is calculated for elliptical contact considering osculation. Maximum contact pressure for CRB and TRB is calculated for line contact modified by an empirical factor that accounts for axial crowning of rollers and inner raceway.

From reference [42], the unadjusted contact pressures are:

$$p_{\text{line}} = 270\sqrt{\frac{1}{2}\left(\frac{Q}{L_{\text{we}}}\right)\Sigma\rho_{\text{line}}} \quad (I.24)$$

$$p_0 = \frac{858}{\mu\,\nu}\sqrt[3]{Q\left(\Sigma\rho_{\text{point}}\right)^2} \quad (I.25)$$

I.3.5 Misalignment factor

For cylindrical and taper roller bearings, the contact pressure is influenced by misalignment of the raceways, represented by shaft slope angle θ_L. K_m is set to unity for spherical bearings that freely accommodate misalignment:

if $m_a < 1.3$

$$K_m = 0.00105\,\theta_L^2 + 0.00406\,\theta_L + 0.9976 \quad (I.26)$$

otherwise

$$K_m = 0.0042\,\theta_L^2 - 0.0092\,\theta_L + 1.013 \quad (I.27)$$

Set K_m equal to 1.0 for spherical type, not less than 1 for others.

$$x = K_m \quad (I.28)$$

if Type = 1

$$K_m = 1 \quad (I.29)$$

otherwise: if $x < 1$

$K_m = 1$

or

$K_m = x$

where

m_a is the ratio of roller length to diameter;

$$m_a = \frac{L_{\text{we}}}{d_1} \quad (I.30)$$

θ_L is misalignment slope of shaft, arc minutes.

I.3.6 Truncation factor

If the calculated length of the contact, $2a$, is greater than the effective roller length, L_{we}, the calculated maximum stress is multiplied by a truncation factor, C_T.

An approximation of this factor is given by the following equation:

if $2a > L_{we}$

$$C_T = \left[1 + \frac{4\left(\frac{b}{a^3}\right)\sqrt{a^2 - \left(\frac{L_{we}}{2}\right)^2}\left[\frac{64}{105}a\left(a - \frac{L_{we}}{2}\right)^2 - \frac{40}{189}\left(a - \frac{L_{we}}{2}\right)^3\right]}{\pi ab - \frac{8}{3}(a - L_{we})\frac{b}{a}\sqrt{a^2 - \left(\frac{L_{we}}{2}\right)^2}} \right] \quad (I.31)$$

otherwise

$$C_T = 1 \quad (I.32)$$

I.3.7 Ratio of maximum to nominal line contact pressure

Based on test data calculated with DIN ISO 281 Bbl.4:2003, with bearing inner diameters ranging from 80 to 500 millimeters, and contact stresses from 800 to 2,500 megapascals, the following curve fit is given for CRB and TRB's:

$$K_{lc} = 1 + 3185(p_{line})^{-1.3633} \quad (I.33)$$

For spherical bearings, the maximum stress can be calculated using the formula for elliptical contact. In this case, K_{lc} is defined as the ratio between p_T and p_{line}:

if Type =1

$$K_{lc} = \frac{p_T}{p_{line}} \quad (I.34)$$

where

$$p_T = C_T\, p_0 \quad (I.36)$$

I.3.8 Calculating the stress

Maximum stress is the previously calculated contact pressure for line contact, p_{line}, adjusted with factors described above:

$$p_{max} = K_{lc}\, K_m\, p_{line} \quad (I.37)$$

where

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.

Mathcad® worksheet example:

Input variables

$Type := 1$	Input type, [1]SRB, [2]CRB or [3]TRB
$Xo := 1$	Static factor, radial, from table 3.2 in reference [42]
$Yo := 2.5$	Static factor, axial, from table 3.2 in referance [42]
$Fr := 7997$	Radial load, N
$Fa := 8297$	Axial load, N
$i := 2$	Number of rolling element rows
$z_{row} := 18$	Number of rolling elements per row
$L_{we} := 19.3$	Roller length, effective, mm
$\alpha_{o\_deg} := 9.45$	Nominal contact angle, degrees
$D_w := 25$	Diameter of rolling element, mm
$D_{pw} := 155$	Roller pitch diameter, mm
$e := .25$	Bearing radial to thrust [e] factor
$Gr := .04$	Radial clearance (absolute), mm
$S := 1.0618$	Osculation ratio
$\theta_L := 0$	Shaft tilt angle, arc minutes

Set bounds for S and Gr

$$temp := \begin{vmatrix} 1.001 \text{ if } S < 1.001 \\ S \text{ otherwise} \end{vmatrix} \qquad S := temp \qquad S = 1.0618$$

$$temp := \begin{vmatrix} .0005 \text{ if } Gr \le 0 \\ Gr \text{ otherwise} \end{vmatrix} \qquad Gr := temp \qquad Gr = 0.04$$

Set total number of rolling elements

$$z := z_{row} \cdot i \qquad z = 36$$

Change contact angle to radians

$$\alpha_o := \alpha_{o\_deg} \cdot \frac{\pi}{180} \qquad \alpha_o = 0.1649 \qquad \text{radians}$$

Equivalent combined load:

$$P_o := Xo \cdot Fr + Yo \cdot Fa \qquad P_o = 28739.5 \quad (N)$$

$$P_o := \begin{vmatrix} P_o \text{ if } P_o > F_r \\ F_r \text{ otherwise} \end{vmatrix}$$

Elastic constant

$$C_{\delta L} := 26200 \cdot L_{we}^{.92} \qquad C_{\delta L} = 399037.78 \qquad (N/mm^{1.08})$$

Calculate the k factor;

$$k := \begin{cases} 4.4 & \text{if } \dfrac{Fa}{Fr} > e \\ 4.05 + .3209 \cdot \left[\dfrac{Po}{C_{\delta L} \cdot \left(\dfrac{Gr}{2}\right)^{1.08} \cdot z \cdot \cos(\alpha_o)} \right]^{-.7911} & \text{otherwise} \end{cases}$$

$$k = 4.4$$

Calculate the maximum rolling element load:

$$Q := \frac{Po}{z \cdot \cos(\alpha_o)} \cdot k \qquad Q = 3560.9 \quad (N)$$

Calculate the curvature factors;

$$\rho_{11} := \frac{2}{D_W} \qquad \rho_{11} = 0.08$$

$$\rho_{21} := \frac{2}{\dfrac{D_{pw}}{\cos(\alpha_o)} - D_W} \qquad \rho_{21} = 0.0151$$

$$\rho_{22} := \frac{-2}{\dfrac{D_{pw}}{\cos(\alpha_o)} + D_W} \qquad \rho_{22} = -0.011$$

$$\rho_{12} := -\rho_{22} \cdot S \qquad \rho_{12} = 0.0117$$

Sum curvature factors for point and line contact

$$\Sigma\rho_{point} := \rho_{11} + \rho_{12} + \rho_{21} + \rho_{22} \qquad \Sigma\rho_{point} = 0.0958 \quad \left(mm^{-1}\right)$$

$$\Sigma\rho_{line} := \rho_{11} + \rho_{21} \qquad \Sigma\rho_{line} = 0.0951 \quad \left(mm^{-1}\right)$$

Calculate cosine of tau

$$\cos\tau := \frac{(\rho_{11} - \rho_{12} + \rho_{21} - \rho_{22})}{\Sigma\rho_{point}} \qquad \cos\tau = 0.9858 \qquad \tau := \operatorname{acos}(\cos\tau)$$

$$\tau = 0.1685 \quad \text{radians}$$

Determine Hertz coefficents, from cos (τ) with curve fits

$$\mu := \begin{cases} 1.396748\cos\tau^{.665242} \cdot (1-\cos\tau)^{-.37399} & \text{if } \cos\tau > .87 \\ \left(5.6864\cos\tau^4 - 6.0607\cos\tau^3 + 2.7985\cos\tau^2 + .35289\cos\tau + 1.005\right) & \text{otherwise} \end{cases}$$

$\mu = 6.7986$

$$\nu := \begin{cases} .683241\cos\tau^{.4} \cdot (1-\cos\tau)^{.189343} & \text{if } \cos\tau > .87 \\ -.30365\cos\tau^3 + .373719\cos\tau^2 - .67694\cos\tau + 1.0014 & \text{otherwise} \end{cases}$$

$\nu = 0.3034$

Calculate the contact dimensions a and b

$$a := \frac{.0472\mu \cdot \sqrt[3]{\dfrac{Q}{\Sigma\rho_{point}}}}{2} \qquad a = 5.3544 \quad (mm)$$

$$b := \frac{.0472\nu \cdot \sqrt[3]{\dfrac{Q}{\Sigma\rho_{point}}}}{2} \qquad b = 0.239 \quad (mm)$$

Calculate contact pressure, line

$$p_{line} := 270 \cdot \sqrt{\frac{1}{2}\left(\frac{Q}{L_{we}}\right) \cdot \Sigma\rho_{line}} \qquad p_{line} = 799.8804 \quad \text{N/mm^2}$$

Contact pressure for point contact

$$p_0 := \frac{858}{\mu \cdot \nu} \cdot \sqrt[3]{Q\left(\Sigma\rho_{point}\right)^2} \qquad p_0 = 1329.9499 \quad \text{N/mm^2}$$

Calculate factor for misalignment

$$m_a := \frac{L_{we}}{D_W} \qquad m_a = 0.772$$

$$Km := \begin{cases} .00105 \cdot \theta_L^2 + .00406 \cdot \theta_L + .9976 & \text{if } m_a < 1.3 \\ .0042 \cdot \theta_L^2 - .0092 \cdot \theta_L + 1.013 & \text{otherwise} \end{cases} \qquad Km = 0.9976$$

Set Km to 1.0 for spherical type, not less than 1 for others

$x := Km$

$$Km := \begin{cases} 1 & \text{if } Type = 1 \\ \text{otherwise} & \\ \quad \begin{cases} 1 & \text{if } x < 1 \\ x & \text{otherwise} \end{cases} & \end{cases} \qquad Km = 1$$

Truncation factor

$$C_T := \begin{cases} \left[1 + \dfrac{4 \cdot \left(\dfrac{b}{a^3}\right) \cdot \sqrt{a^2 - \left(\dfrac{L_{we}}{2}\right)^2} \cdot \left[\dfrac{64}{105} \cdot a \cdot \left(a - \dfrac{L_{we}}{2}\right)^2 - \dfrac{40}{189} \cdot \left(a - \dfrac{L_{we}}{2}\right)^3 \right]}{\pi \cdot a \cdot b - \dfrac{8}{3} \cdot \left(a - L_{we}\right) \cdot \dfrac{b}{a} \cdot \sqrt{a^2 - \left(\dfrac{L_{we}}{2}\right)^2}} \right] & \text{if } 2 \cdot a > L_{we} \\ 1 & \text{otherwise} \end{cases}$$

$C_T = 1.00000$

$p_T := C_T \cdot p_0$ $\qquad$ $p_T = 1329.9499$

$$K_{lc} := \begin{cases} \dfrac{p_T}{p_{line}} & \text{if } Type = 1 \\ 1 + 3185 \cdot \left(p_{line}\right)^{-1.3633} & \text{otherwise} \end{cases} \qquad K_{lc} = 1.6627$$

$p_{max} := K_{lc} \cdot Km \cdot p_{line}$ $\qquad$ $p_{max} = 1329.95$ N/mm^2

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