How reliable is a reliability calculation?

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Management summary

Main gearbox bearing failures are of great concern to operators of wind turbines and by extension to the gearbox supplier. They have a strong interest in predicting this risk of failure or the reliability of bearings and the bearing subsystem. The bearing subsystem reliability number for a required lifetime is the fraction of the total gearbox population (provided it is a large population) that will survive the required lifetime without any bearing failing.

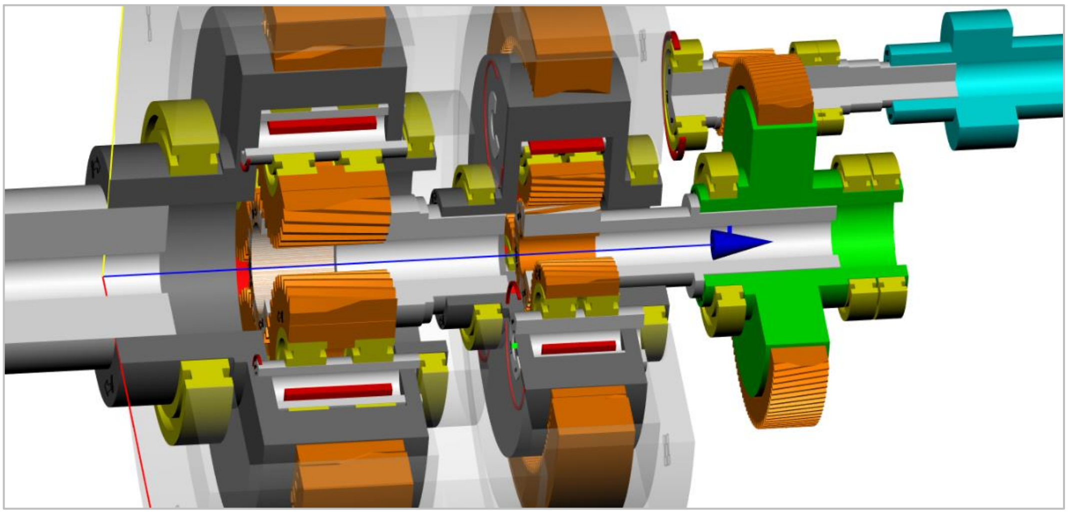


Figure 0-1 Typical wind turbine main gearbox, bearings in yellow.

This reliability number has obviously a major influence on the total cost of ownership of a gearbox and the product in which it is used [23]. A higher, initial investment, e.g., in a safe design, will result in a higher reliability and hence lower costs for repairs and from downtime. A lower initial investment may result in a lower reliability and hence higher costs for e.g. spare parts. It is therefore of great interest to predict the reliability of a gearbox or its bearing subsystem. Such predictions are based on the bearing life calculation and a model for the variation of the component life, if a large number of components is considered.

Here, a particular effect contributing to the margin of error of such predictions is investigated. The reliability number from a reference calculation is compared to reliability numbers of calculations where different parameters or assumptions used in the calculations are varied slightly. These variations occur e.g., if two parties (manufacturer and user of the gearbox, manufacturer and certification body, two potential suppliers, …) perform such a calculation using the same bearing data and the same loads. It is found that even small variations have an effect in the range of +-10 %-points on the calculated reliability number for required life, $R(H_{req})$. This means that – when comparing two calculated reliability numbers – it is of utmost importance that the calculations are done with exactly the same assumptions, calculation methods, loads, using the same tools and preferably by a single entity. This basically means that a purchaser of gearboxes is advised against comparing two reliability numbers submitted by two potential gearbox suppliers, but he should calculate these numbers by himself. It is too simple to "tune" such calculated reliability numbers!

Introduction

Objective of the study

Bearing failures are a major concern in wind turbine main gearboxes. Predicting the risk of a bearing failure (to calculate the total cost of ownership TCO and to predict the need for spare parts) or a comparison between several designs to find the one with the lowest risk (e.g. when selecting suppliers and their designs) is of interest to all stakeholders.
A qualitative assessment of the risk of failure is achieved by calculating the bearing life L and therefrom the bearing failure probability $F(H_{req})$ for a required life H_{req}. If this is done for all bearings in a gearbox, then, the bearing subsystem reliability for the required life $R(H_{req})$ may be calculated. The bearing subsystem reliability $R(H_{req})$ may then be used to calculate the bearing subsystem failure probability $F(H_{req}) = 1\text{-}R(H_{req})$. The bearing subsystem failure probability for a required life, $F(H_{req})$ is then the fraction of a gearbox fleet (assuming the fleet is a high number of gearboxes) that will experience a failure of any bearing in the gearbox by design. This means it is an "in-built" and expected fatigue failure.
The theory that is typically applied for such a calculation is well established and has been widely used e.g. in the design process for e.g. aircraft gearboxes even in the 1970s, [11], [12], [13], [14], [15], [16], [17]. It is well documented, e.g. in [9] from where it may easily be implemented in a computer code. Commercial software is widely available for system level reliability calculations as part of a design failure mode and effect analysis (DFMEA) process, e.g. Reliasoft, [27] or within gear and gearbox design tools, e.g. KISSsoft [28]. It is nowadays applied to wind gearboxes by turbine OEMs, operators, gearbox suppliers, [10] or academia, [25].

All stakeholders will be aware that a calculated reliability or rather risk of failure will not necessarily reflect the failure rate experienced in the field. Reason for this is for example the fact that only certain types of bearing failures are open to a predictive reliability calculation. Basically, only the bearing rolling contact fatigue (surface or subsurface initiated) is available for a probabilistic assessment, see section 0 below. Also, tuning the calculations to field observations to take into account e.g. site-specific loads is a necessity.

In this paper, another, simpler and therefore often overlooked aspect influencing the accuracy of such calculations is addressed. The calculated reliability as a function of time t, R(t) of a bearing subsystem (or any other subsystem, e.g. the gears) will change with any small variation of input parameters or calculation method. Such variations may be due to different parties performing the calculations (this can be two engineers in the same company or two suppliers of gearboxes to one wind turbine manufacturer, or a gearbox supplier and a certification agency) using different assumptions with respect to those input parameters. Practical experience in the industry confirms this likely to be the case whenever more than one calculation is done, in particular, if the several calculations are done by different persons. The question then is: how much will the calculated reliability change if somebody, by mistake or by intention, changes or "tunes" those input parameters? How reliable are such reliability calculations?

Bearing failures in wind gearboxes

The 2014 wind turbine gearbox damage distribution based on NREL gearbox reliability database as shown below, [23], is based on approximately 320 gearbox damage records. The majority of the damage occurs to bearings (65 %), followed by gears (25 %).

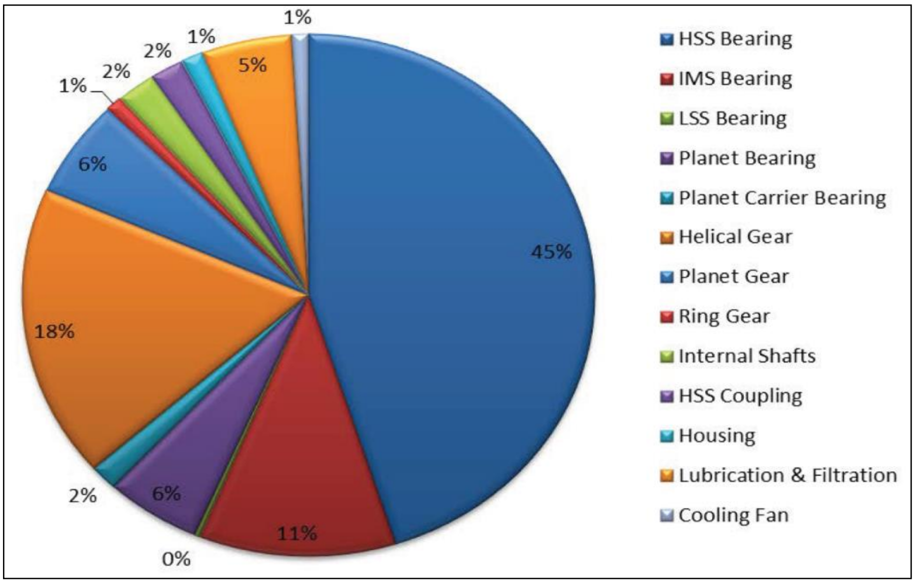


Figure 0-1 Gearbox damage distribution, [23].

On the other hand, a quantitative assessment of the bearings is based on a static rating, typically along ISO 76 [6] a life rating along ISO 281 [5] and ISO/TS 16281 [7]. Furthermore, contact stresses may be compared to recommended levels given in industry specific standards and guidelines [1], [2], [3], [4]. However, these calculated lifetimes and stress levels are covering only one or two (if we consider the static rating) possible failure modes of bearing, that is, fatigue failure (due to surface or sub-surface-initiated fatigue). Other failure modes, in particular white etching cracks or axial cracks, are not yet open to a quantitative assessment. Failure due to poor handling or assembly technique is also not considered. This means that the calculations done during the design phase are not addressing all potential future failure modes and therefore under predict the risk of failure or over predict the reliability for a required lifetime (or overpredict the lifetime for a required reliability).

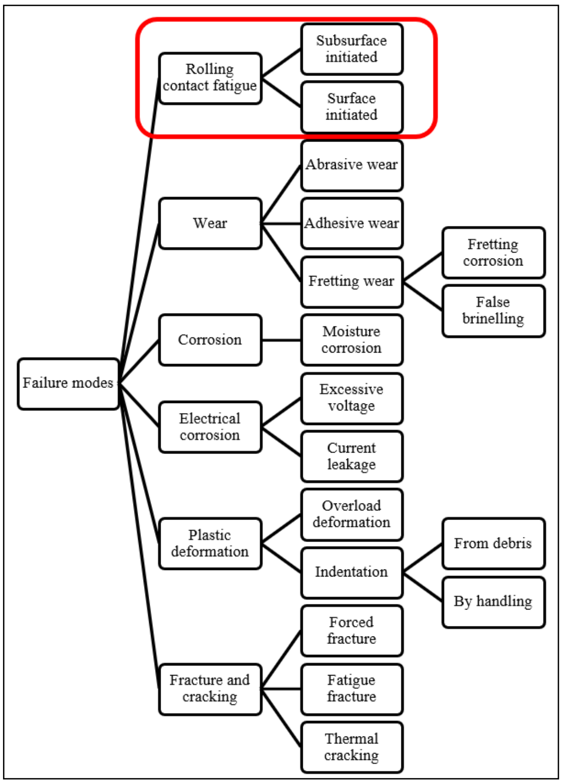


Figure 0-2 Bearing failure modes along ISO 15243 [24].

Reliability as a design requirement for wind turbine gearboxes

ISO 81400-4, [1], states "… 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."

In this statement, attention is directed to the underlying reliability of the material data or load capacity numbers associated with the life calculation of components. For the bearing rating, applicable and typical design rules [1], [2], [3], [4] and others usually stipulate the use of a bearing failure probability of 10 %. Wind turbine OEMs however often change this requirement to a lower failure probability of e.g. 5 %, giving an a1 factor a1 = 0.64 along ISO 281, [5]. Correspondingly, the modified rating life is 0.64 times the basic rating life.

For gears, values for the permissible root and flank stress, σFlim and σHlim respectively, are given for 1 % probability of damage, corresponding to a reliability level of 99 % for the gear rating against bending and pitting. However, ISO 6336 gear rating does include data for other levels of reliability. A more detailed approach is available with AGMA 2101 where a reliability factor Y_Z is used. Said factor is equal to unity for one failure in 100 and increases to $Y_Z = 1.50$ for fewer than one failure in 10'000. For shaft rating, DIN 743 again uses a different material reliability level of 97.5 %.

This reliability requirement is valid only for the single component, or rather, the rating of a single component against a single type of failure. E.g. if the gear rating is performed against bending with a reliability of 99 % and the against pitting again with a reliability of 99 %, then, the reliability of the gear against bending or pitting is less than 99 %.

Correspondingly, if in a gearbox each bearing individually just reaches the required life H_{req} (e.g. 175'200 h = 20 y as typically required in wind industry) at a probability of failure of F = 10% or a reliability of R = 90 %, then, the reliability of the bearing subsystem for the required life is less than 90 %. The reliability decreases with the increasing number z of bearings. To quantify the reduction, the reliability of the bearing subsystem (= all z bearings together) $R(H_{req})$ for a certain time H_{req} is calculated from the reliability of each bearing individually, $R_b(H_{req})$ as follows:

$$R(H_{req}) = R_{b1}(H_{req}) * R_{b1}(H_{req}) * \dots * R_{bz}(H_{req}) = \prod_{i=1}^{z} R_{bi}(H_{req})$$

The underlying principle for above formula to be applicable is that the bearing subsystem is considered as a system with a strictly serial structure. This means that if one bearing fails, there is no back-up and the bearing subsystem is considered as failed. Other conditions are that the components and the subsystem is either "functional" or "failed" and that the components are independent, their respective state "functional" or "failed" do not depend on another components state.

Bearing reliability calculation

In 1939, W. Weibull developed a method and an equation for statistically evaluating the fracture strength of materials based upon small population sizes, [33]. This method can be and has been applied to analyze, determine, and predict the cumulative statistical distribution of fatigue failure or any other phenomenon or physical characteristic that manifests a statistical distribution. The dispersion in life for a group of homogeneous test specimens can be expressed by:

$$\ln\left(\ln\left(\frac{1}{R}\right)\right) = \beta * \ln\left(\frac{L-\gamma}{L_\beta - \gamma}\right)$$

Where:
R = Probability of survival or reliability, $0 < R \leq 1$
β = Shape parameter or Weibull slope
L = life or number of cycles
γ = location parameter or the time below which no failure occurs
L_β = Characteristic life, time at which 63.2 % of a population will fail (further below, T will be used for characteristic life)

The format of above equation is referred to as a three parameter (3P) Weibull function. For most – if not all – failure phenomenon, there is a finite time period under operating conditions when no failure will occur. In other words, there is zero probability of failure, or a 100 % probability of survival, for a period of time during which the probability density function is non-negative. This value is represented by the location parameter γ. Without a significantly large database, this value is difficult to determine with reasonable engineering or statistical certainty. As a result, γ was taken as zero by Lundberg and Palmgren and above equation was used in below, re-written form for the prediction of bearing life vs. reliability, [33]:

$$\ln\left(\ln\left(\frac{1}{R}\right)\right) = \beta * \ln\left(\frac{L}{L_\beta}\right)$$

This format is referred to as the two-parameter (2P) Weibull function. The estimated values of the Weibull slope β and L_β for the two-parameter Weibull analysis may not be equal to those of the three-parameter analysis, [12]. This concept is also the basis for the calculation of the a1 factor for the modified rating along ISO 281. Note that the calculation for a1 factor has been changed in ISO 281:2007 (three parametric Weibull distribution is used with a failure free time of $\gamma = 0.05*L$; or $C\gamma = 0.05$) compared to ISO 281:1990 (where a two parametric Weibull distribution is used and $\gamma = C\gamma = 0$), [33]. In the latest revision of AGMA 6006, [3], for each bearing i, the reliability $R_{bi}(t)$ as a function of time t is calculated. Furthermore, for each bearing i, the reliability $R_{bi}(H_{req})$ for the required component life H_{req} is calculated. Note that the bearing life is taken as the modified reference rating life, $L = L_{nmrh}$ with n = 10% or R0 = 90%. For the calculation of the bearing reliability, a three parametric Weibull distribution as shown below is used:

$$R(t) = e^{-\left(\frac{t-\gamma}{\eta}\right)^{\beta}}$$

$$\eta = \frac{L-\gamma}{\sqrt[\beta]{-\ln(R_0)}}$$

Where
L = calculated bearing life as modified reference rating L_{nmrh}
γ = Location parameter = $C\gamma$ * L; $C\gamma$=0.05
β = Shape parameter = 1.500
η = Scale parameter
R_0 = Reference reliability, reliability used to calculate L_{nmrh}, R_0 = 90%

The shape parameter β is in line with the value used in ISO 281, [5] and ISO / TR 1281, [31]. For the shape parameter β, values in literature differ considerably, e.g. Harris [32], states "… for modern, ultraclean, vacuum remelted steels, values of e in the range 0.7-3.5 have been found…".

The location parameter $\gamma = C\gamma*L = 0.05*L$ is also in line with above references. It is conservative when compared with other values published, e.g. γ = 0.053 in [30] or f_{tB} = 0.1…0.3 in [9]. Note however that the failure free time (or the location parameter) is defined in [31] with respect to the basic rating life L_{10} while in AGMA 6006, [3], the calculated life L is the modified reference rating life L_{10mr}.

Influence of the shape of the Weibull distribution

AGMA 6006 uses a three parametric Weibull distribution for bearings and it defines the shape and location parameter to be used, see above. Let us compare different Weibull distributions according to relevant sources including FAG Wälzlagerpraxis, [34], AGMA 6006 [3] (based on ISO 281:2007 [5]), Bertsche [9] (using parameters as listed in section 7, mean values), ISO 281:1977 [35]. Below, the respective parameters are listed:

	Type of Weibull distribution	Shape parameter β	Location parameter γ
FAG	2 parametric	1.11 for ball, 1.125 for roller bearings	0*L
AGMA 6006	3 parametric	1.5 for ball, 1.5 for roller bearings	0.05*L
Bertsche	3 parametric	1.1 for ball, 1.35 for roller bearings	0.20*L

Table 0-1 Shape and location parameter for different bearing types according to different sources.

Using the above parameters for roller bearings, we find the below curves for reliability and failure probability over time (where the time t is normalized by the characteristic time T). While all curves obviously yield identical results for t = T, the results are quite different for a required reliability of R = 90% (horizontal solid line in red). There, we find (for roller bearings) that the calculated life along FAG is t = 0.135*T (left vertical solid line in grey) whereas along Bertsche, we find t = 0.355*T (right vertical dashed line in grey). Value for AGMA rating is t = 0.265*T (middle vertical dotted line in grey). Basically, we observe that for low lifetimes (as typically required in a design) the choice of the parameters β (shape parameter) and γ (location parameter) is critical. A comparison between two designs or two calculations for one design must therefore quite obviously use the same distribution (Weibull distribution) and the same parameters β and γ. Not discussed here is the influence if distribution functions other than the Weibull distribution (e.g. normal distribution, log-normal distribution, …) although Thoma, [10], reports that field data (of wind turbine gearboxes, considering gears and bearings failures) may be matched more closely if a log-normal distribution is used.

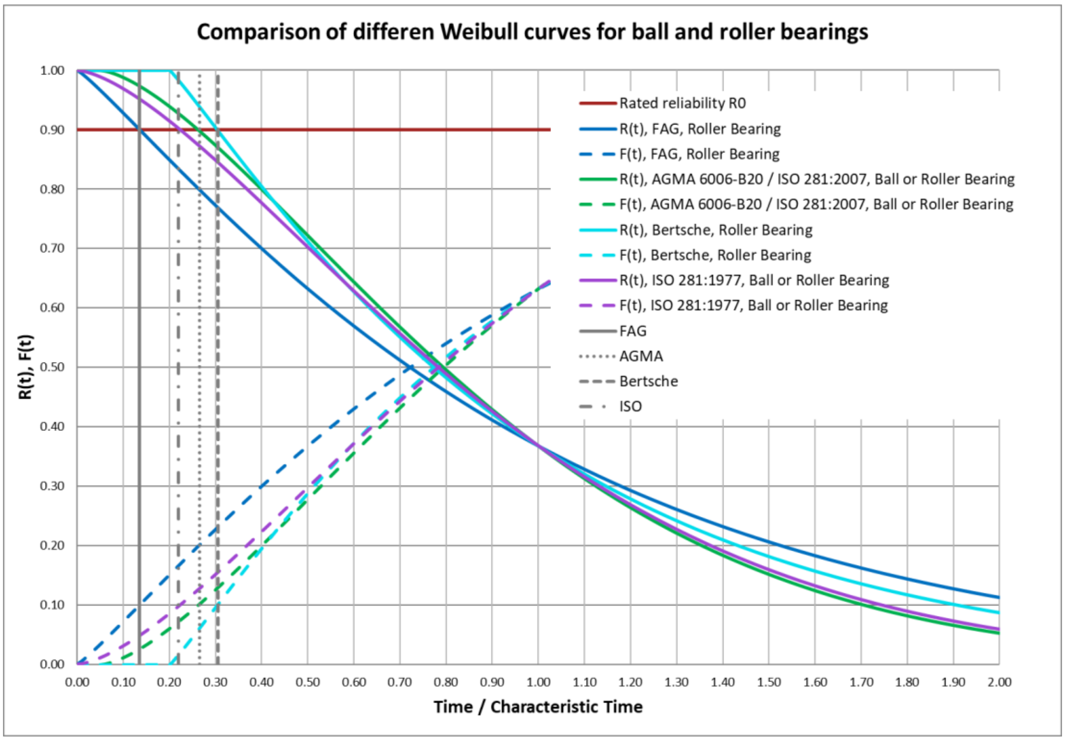


Figure 0-1 Bearing reliability as a function of normalized time for different bearing types (shown for roller bearings only) and different shape and location parameters along different sources (FAG, AGMA 6006 / ISO 281:2007, Bertsche and ISO 281:1997).

Bearing subsystem life and reliability

Base line model and reference conditions

In this paper, the bearing subsystem of a 3MW class wind turbine main gearbox (MGB) as shown below is investigated. The gearbox uses four planets with four rows of CRB each in the low-speed stage LSS, three planets with two rows of cylindrical roller bearings CRB in the intermediate speed stage ISS and a CRB and two taper roller bearings TRB on the high-speed stage HSS shafts. Together with the planet carrier bearings, a total of 32 individual bearing rows are employed. For each bearing, the modified reference rating life $L = L_{10nmrh}$ is calculated along ISO/TS 16281 using a KISSsoft, [28], model as shown below. Note that the gearbox is used with a main shaft supported by two main bearings, hence, the loads on the planet carrier bearings are low and the corresponding lifetimes very high.

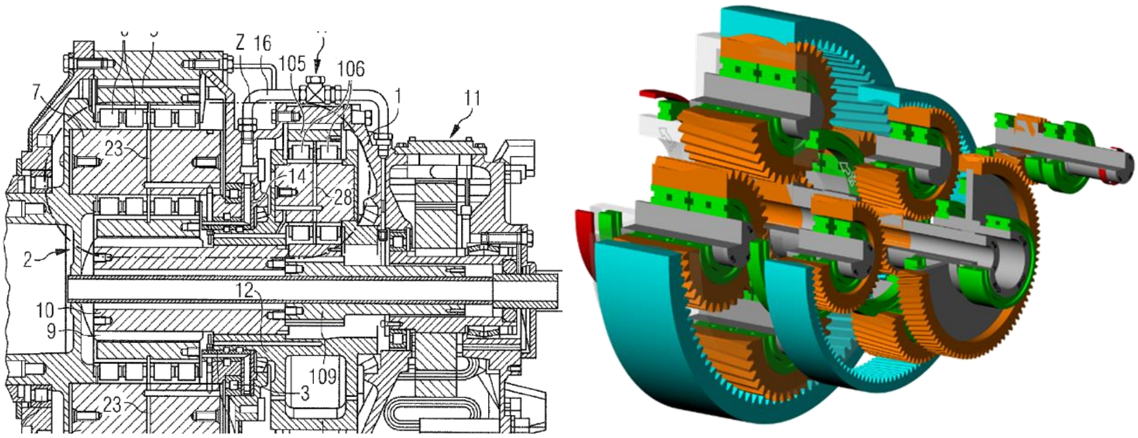


Figure 0-1 Left: Bearing arrangement considered, note that HSS shaft is not shown, two CRB and a four point ball bearing FPBB is used there, [21]. Right: KISSsoft model used for calculation of the modified reference rating life, the operating clearance, non-linear bearing stiffness, load and stress distribution for all bearings. Planetary carrier ISS not shown, housing not shown.

For this paper, different effects will be considered in the calculation. From a reference condition, small changes will be introduced to study how these changes affect the resulting reliability of the bearing subsystem. In the be low table, the effects considered and their reference conditions are listed.

Property	Reference value	Affects	Comments
Bearing clearance, position within the tolerance field	Mean tolerance value for bearing clearance, shaft and housing tolerance	Bearing operating clearance and hence load distribution in bearings	The load distribution then affects the rated life
Shaft and housing / inner and outer race temperature	Temperature differences are set as per Table 4 of ISO 81400-4, [1]	Bearing operating clearance and hence load distribution in bearings	The load distribution then affects the rated life
Load application position with respect to gear face width	Load is assumed to act in center of gear face width	Load distribution on planet bearings LSS and ISS	No recommendations are given in standards or certification guidelines
Planet load distribution ($K\gamma$)	$K\gamma = 1.10$ for LSS and $K\gamma = 1.05$ is assumed for bearing load calculation	Load level on the planet bearings LSS and ISS	$K\gamma$ is a factor typically used in gear rating but uneven load distribution among the planets also affects bearing load

Nominal torque on gear-box input	100% nominal load is used	Load level on all bear-ings except carrier bear-ings	To consider site specific loads
Lubricant temperature	Selected as 65C° for all bearings	Lubricant viscosity and aISO factor	
Lubricant contamination	Selected as - / 17 / 14, beta25 = 75 along ISO 4406	Life rating, through ec value and aISO factor	
Bearing clearance varia-tion in planet bearings	All bearings have same bearing clearance	Load distribution among the bearings, for LSS and ISS planet bearings	Bearings assumed to be perfectly matched
Pressure angle, gears	Operating pressure an-gle αwt is used to calcu-lated gear forces	Bearing forces	Often, bearing supplier uses nominal pressure angle, not operating pressure angle

Table 0-1 Reference conditions for the calculation of the bearing life.

Results, reference calculation

Using calculation settings in line with DNV GL guideline [4] and those listed in above table, the calculations per-formed using KISSsoft software [28] resulted in the following individual bearing lifetime. Note that the modified reference rating life in hours, L_{10mrh} is calculated, this is in line with requirements of AGMA 6006 [3]. The life of the planet carrier bearings is not a calculated but an assumed value, such that they will not noticeably influence the reliability rating.

Stage	Position	Shaft	L10mrh in hours
LSS	RS-RS	Planet, same result for all four planets	314'820
LSS	RS-GS	Planet, same result for all four planets	9'889'777
LSS	GS-RS	Planet, same result for all four planets	10'504'562
LSS	GS-GS	Planet, same result for all four planets	345'866
LSS	RS	Carrier	9'999'999
LSS	GS	Carrier	9'999'999
ISS	RS	Planet, same result for all two planets	646'878
ISS	GS	Planet, same result for all two planets	675'973
ISS	RS	Carrier	9'999'999
ISS	GS	Carrier	9'999'999
HSS	RS	Driving shaft	344'247
HSS	GS-RS	Driving shaft	9'999'999
HSS	GS-GS	Driving shaft	622'918
HSS	RS	Driven shaft	286'648
HSS	GS-RS	Driven shaft	677'429
HSS	GS-GS	Driven shaft	382'756

Table 0-1 Calculated bearing life, reference values. For carrier bearings, a very high life is assumed.

For each of the 32 bearings, the reliability function R(t) and the failure probability function F(t) is plotted using a three parametric Weibull distribution with shape and location parameter along AGMA 6006, $\beta = 1.50$, $\gamma = 0.05*L_{10mrh}$. See grey lines in below figure. Then, the bearing subsystem reliability function (pink line in below figure) and the failure probability function (cyan line in below figure) is calculated as the product of all functions of the individual curves.

The intersection of the line representing the required subsystem life H_{req} at 175'200 hours (vertical, red) and the line representing the time dependent subsystem reliability (blue) results in a subsystem reliability value of about 0.59 (blue, dashed, horizontal line). This means that in a large population of gearboxes, we expect that for 100%-59%=41% of the gearboxes, a bearing fatigue failure is to be expected by design. If the required reliability of R_{req} = 90% should also apply for the bearing subsystem reliability (and not only for the individual bearing), then, the rated life drops from 175'200 hours to about 72'000 hours (intersection of horizontal, solid cyan line and blue curve, from there, vertical, solid pink line).

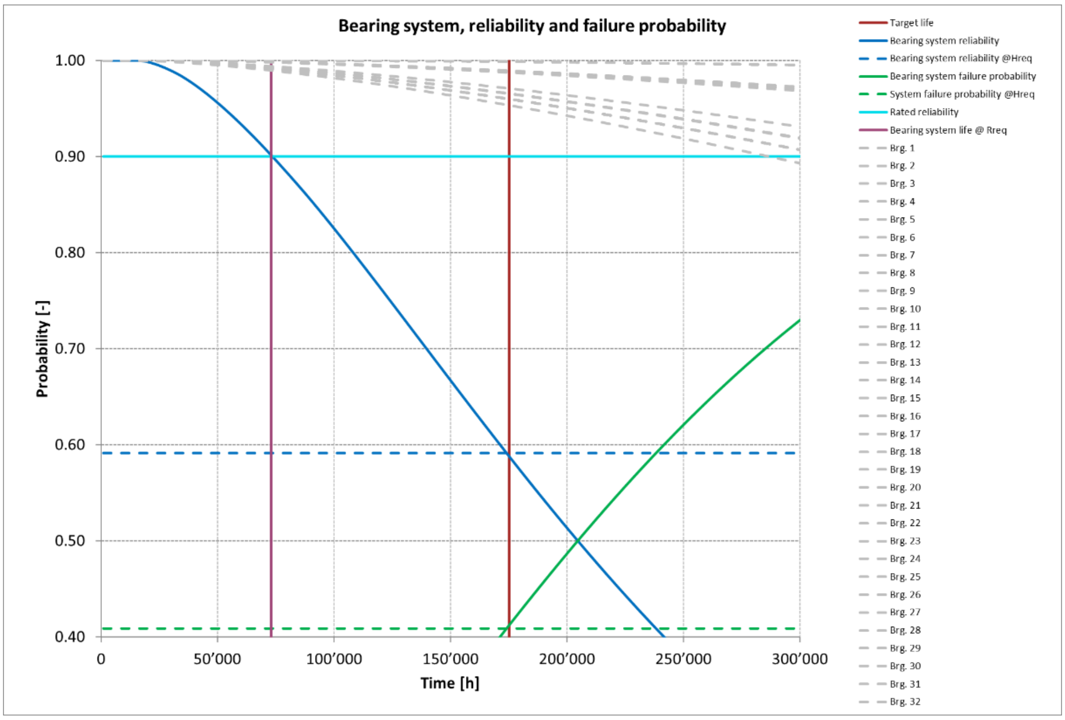


Figure 0-1 Single bearings reliability curves (grey), bearing subsystems reliability (pink), bearing subsystems failure probability (cyan), required subsystems life H_{req} (vertical, solid blue), rated reliability (horizontal, solid orange) and subsystem reliability at required life (horizontal, dashed orange) and resulting subsystem life at rated reliability (vertical, dashed blue).

If we now investigate the rated reliability (the reliability for the required lifetime H_{req}) by grouping the values per stage (LSS planet bearings in orange, carrier bearings in blue, ISS planet bearings in pink, HSS bearings in cyan), we observe in the below graphic that

- The outer planet bearings (RS-RS and GS-GS position) drive the reliability of the LSS
- The ISS has fairly high reliability values
- Bearing 30 on the output shaft has the lowest reliability

If we multiply the reliability values, we find in the second graphic below that the bearing subsystem reliability will be determined by the LSS planet bearings reliability and to some extend by the HSS bearings. The results are in line with field experience and expectations.

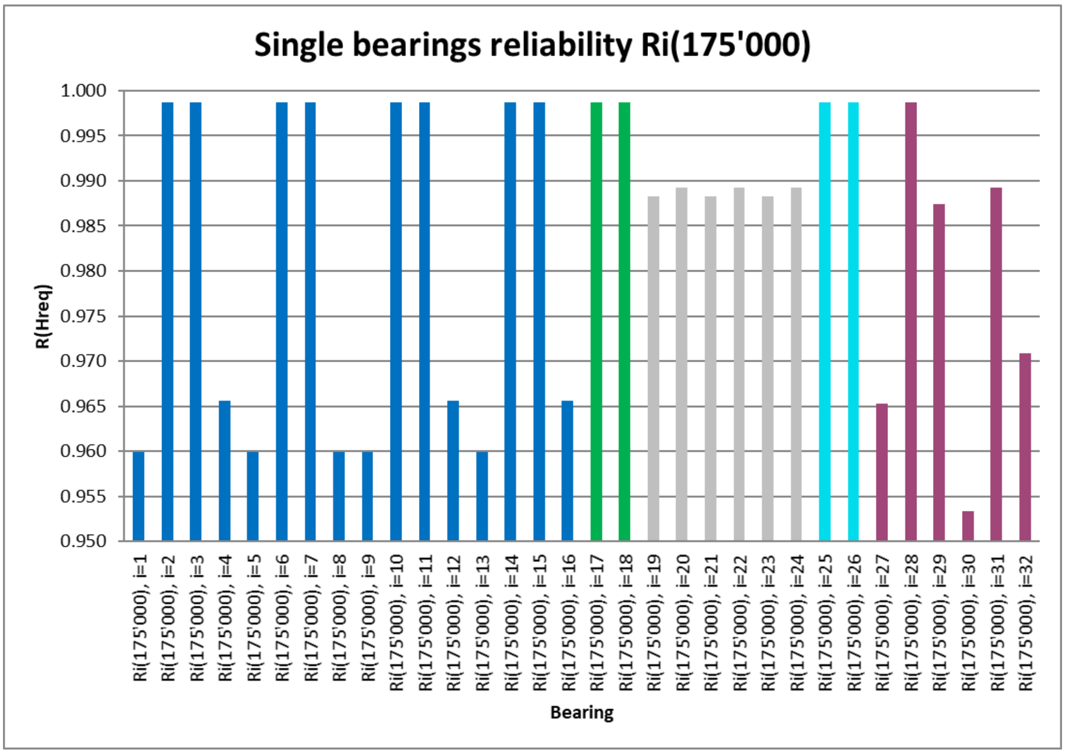


Figure 0-2 Reliability for required life for all bearings, grouped per stage (blue: LSS planet bearings, green: LSS carrier bearings, grey: ISS planet bearings, cyan: ISS carrier bearings, pink: HSS bearings).

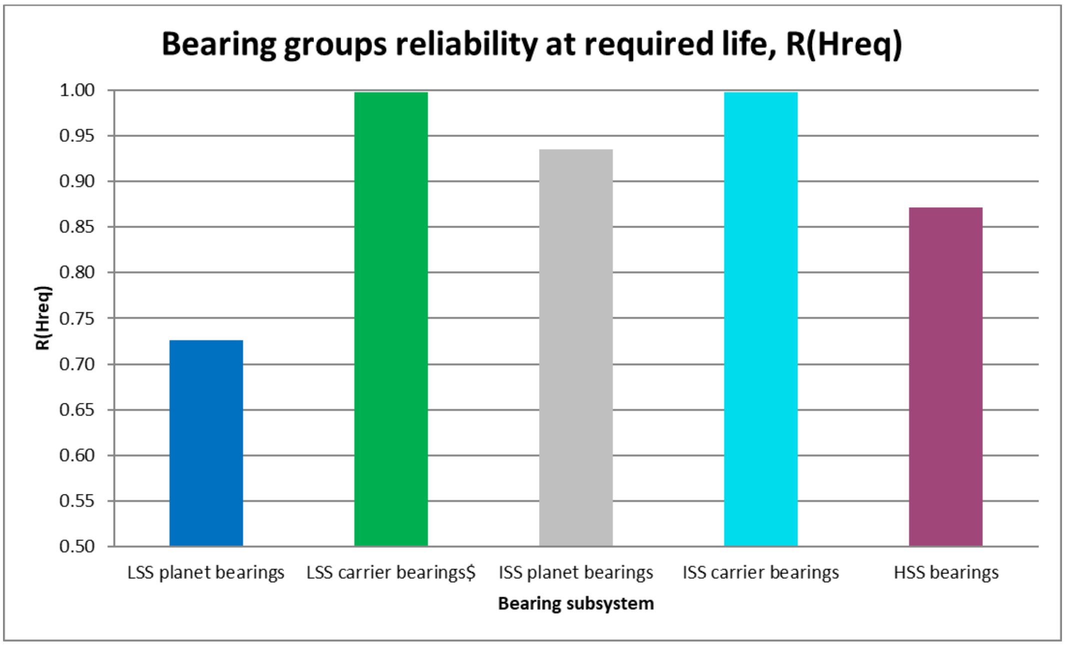


Figure 0-3 Bearing group reliability per stage / group of bearings.

Variation of calculation settings

Parameters varied

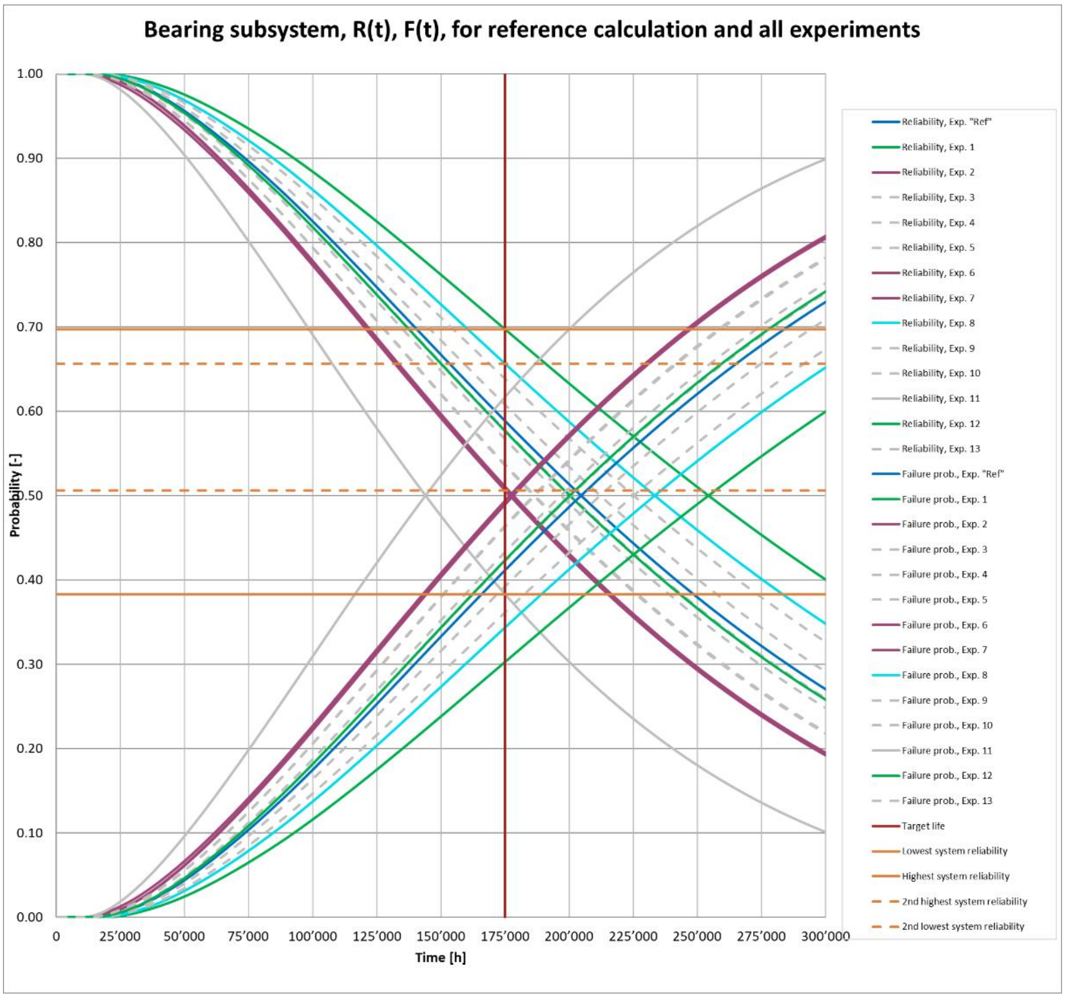
Several experiments are set up. In each experiment, only one parameter is changed compared to the reference calculation. This is a non-conservative approach.

Experi-ment	Parameter varied	Variation	Comments
1	Bearing clearance, position within the tolerance field	Lower position in tolerance field of resulting bearing clear-ance is used	Considers bearing clearance, shaft and housing / planet bore tolerance
2	Bearing clearance, position within the tolerance field	Upper position in tolerance field of resulting bearing clear-ance is used	Considers bearing clearance, shaft and housing / planet bore tolerance
3	Shaft and housing / inner and outer race temperature	Temperature difference be-tween races reduced by 5K	Influences bearing operating clearance but not oil viscosity
4	Shaft and housing / inner and outer race temperature	Temp. difference between races increased by 5K	Influences bearing operating clearance but not oil viscosity
5	Load application position with respect to gear face width	Offset of 1 cm from center	
6	Load application position with respect to gear face width	Offset of 2 cm from center	
7	Planet load distribution ($K\gamma$)	$K\gamma$ values increased by 0.05 in both LSS and ISS	
8	Planet load distribution ($K\gamma$)	$K\gamma$ values decreased by 0.05 in both LSS and ISS	
9	Nominal torque on gearbox in-put	Nominal load decreased by 2.5 %	To consider e.g. site-specific loads
10	Nominal torque on gearbox in-put	Nominal load increased by 2.5 %	To consider e.g. site-specific loads
11	Lubricant contamination	One class worse, - / 19 / 16, beta25 = 7	Compared to reference class - / 17 / 14, beta25 = 75
11a	Lubricant contamination	One class better, -/15/12, beta12=200	Compared to reference class - / 17 / 14, beta25 = 75
12	Pressure angle, gears	Nominal instead of operating pressure angle	
13	Bearing clearance variation in planet bearings	Outer bearing rows, LSS plan-ets, clearance reduced by 10 μm, inner have a clearance in-creased by 10 μm	To simulate that bearings are not perfectly matched

Table 0-1 Set up of the 14 experiments

Resulting reliability curves

For each experiment described above, the bearing lifetime for all 32 bearings were calculated (for the four planet carrier bearings, a value of L = 9’999’999 h was used throughout). For each bearing in each experiment, the reliability curve was calculated, and the bearing subsystem reliability and failure probability curve is plotted in the below figure. This gives a cluster of 13 curves corresponding to the 13 experiments. The resulting reliability values for the required life H_{req} are determined as intersection of the reliability curves with the vertical line at x = H_{req}. The resulting values are shown in the second figure below, again for all experiments. In both graphics, the highest (green), second highest (cyan), lowest (grey) and second lowest (magenta) curve are highlighted. Also, the reference calculation result is shown in blue. Experiment 1 and experiment 12 gave highest reliability (same result was achieved when result is rounded to two digits), highlighted (green). Experiment 8 gave second highest reliability (cyan). Experiment 2, experiment 6 and experiment 7 gave second lowest reliability (same result was achieved when result is rounded to two digits), (pink). Experiment 11 gave lowest reliability (grey, solid line).



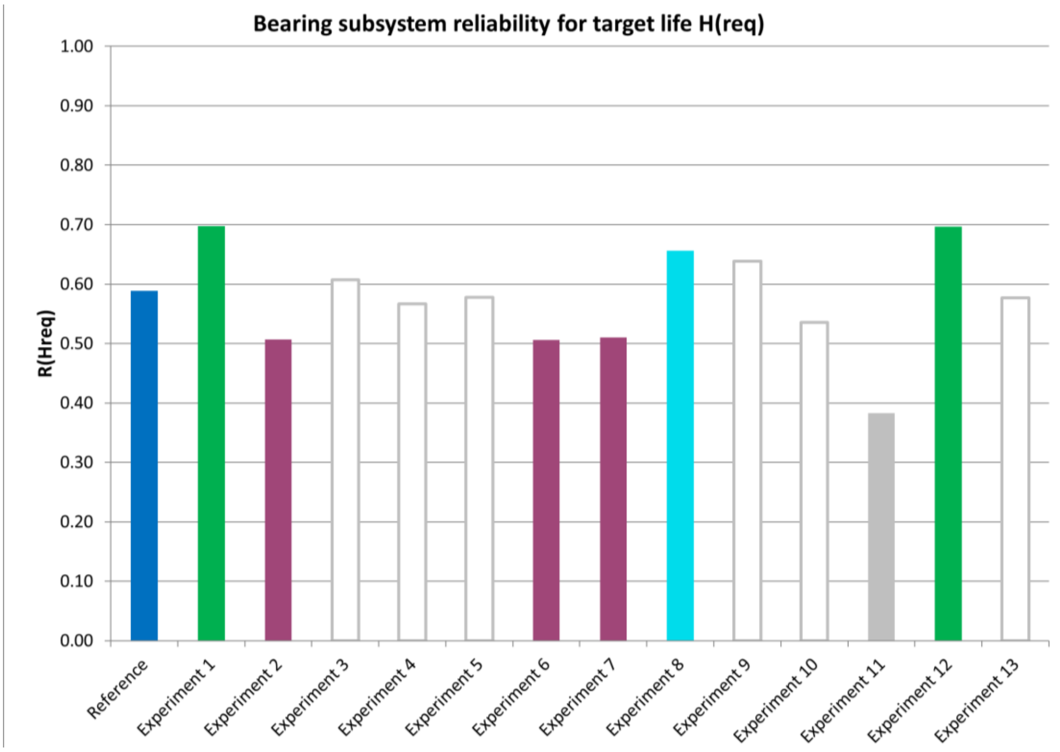


Figure 0-1 Bearing subsystem reliability for different experiments (blue: reference calculation, green: highest result, pink: second lowest result, grey: lowest result, cyan: second highest results, white: other results)

Calculated reliability values $R(H_{req})$ range from 38 % to 70 % or by 32 %-points. If we accept that experiment 11, where a lower lubricant purity is used, is somewhat extreme (recommended purity levels to be used in calculation are clearly documented e.g., in certification guidelines) and omit it from the assessment, we find as lower limit the value of $R(H_{req})$ = 51%, giving a range of 19%-points. The standard deviation for above values is σ = 8%-points if highest and lowest values are considered. If experiment 11 is not considered, then, standard deviation reduces to σ = 6(.4) %-points. Highest deviation from reference calculation is +11 %-points and -21 %-points (-8 %-points if experiment 11 is not considered). Note that in the experiments, only a single parameter is varied compared to the reference calculation while in a real-life scenario, it is likely that several parameters may deviate. From the above, we clearly see that a +-10%-point error margin or more is to be expected if calculation settings are not controlled with utmost care.

Conclusion

Governing effects

In line with prior experience, we find that the reliability of the bearing subsystem is governed by the outer bearing rows in the LSS planet bearings and the output shaft bearings.
From above sensitivity study, we find that the most dramatic influence was observed if the lubricant cleanliness level is changed. This is easy to understand as the lubricant cleanliness level itself affects the rated life of the LSS planet bearings the most, since those have the lowest lubricant film thickness and therefore a low aISO factor. This means that for the LSS planet bearings, a low reliability at reference condition and a high variation of the life and hence reliability if the lubricant cleanliness level changes are combined.
For the LSS planet bearings, other parameters are also difficult to assess. These include the load distribution among the planets, in particular for systems with number of planets higher than three. Also, due to the high face width of planets and due to system deformation, the gear forces typically do not act in the middle of the face width, but the exact location is hardly known and changes over one rotation. Finally, since there are four bearing rows in a single planet, there will be a variation in the bearing clearance even if they are matched. This again leads to an uneven load distribution among the bearing rows.
On the HSS bearings, a major influence is the pre-tension of paired TRBs and the influence of the bearing raceway temperature. On the output shaft bearings, additional loads due to generator misalignment will play a role but is not investigated here. This is in line with experience that system alignment procedure, bearing assembly and cooling is crucial for the reliability of the HSS bearings.
In this respect, the above calculations and the results gained are well in line with practical experience and underline the importance of sound bearing design, careful quality control in assembly to get the required clearances and lubrication (in terms of temperature management and cleanliness).

How reliable is the reliability calculation?

From above analysis, we find that a calculated bearing subsystem reliability of a typical wind turbine gearbox has a reproducibility error of +-10 %-points. This means that if two parties calculate the same gearbox, we must expect that their results will differ by the above error. This also means that if for two competing designs, the difference in the calculated reliability (assuming the calculations are done by the two respective suppliers) is not more than 20 %-points, then, it is not clear whether the difference is accidental. The high level of reproducibility error had to be expected as any life calculation is highly sensitive to the assumptions and process.

As a purchaser of a gearbox, interested in comparing the total cost of ownership for several competing designs, based on gearbox reliability numbers supplied by the suppliers, this level of variation is disappointing. The obvious solution is that all calculations are done strictly with identical assumptions and methods with respect to all parameters affecting them.

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