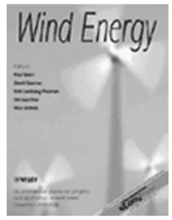

Misalignment in wind turbine drivetrains: improving bearing reliability

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Complete List of Authors:	Whittle, Matthew; Durham University, School of Engineering and Computing Sciences Trevelyan, Jon; Durham University, School of Engineering and Computing Sciences Shin, Won; Romax Technology Ltd, Tavner, Peter; Durham University, School of Engineering and Computing Sciences
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Misalignment in wind turbine drivetrains: improving bearing reliability

Matthew Whittle[1], Jon Trevelyan[1], Won Shin[2], Peter Tavner[1]

1. School of Engineering and Computing Sciences, Durham University, Durham, DH1 3LE, UK
2. Romax Technology Ltd., Rutherford House, Nottingham Science and Technology Park, Nottingham, NG7 2PZ, UK

Abstract

Improving the reliability of wind turbines (WT) is an essential component in the bid to minimise the cost of energy, especially for offshore wind due to the difficulties associated with access for maintenance. Numerous studies have shown that WT gearbox and generator failure rates are unacceptably high, particularly given the long downtime incurred per failure. There is evidence that bearing failures of the gearbox high speed stage (HSS) and generator account for a significant proportion of these failures. However, the root causes of these failure data are not known and there is, therefore, a need for fundamental computational studies to support the valuable ‘top down’ reliability analyses. In this paper a real (proprietary) 2 *MW* geared WT was modelled in order to compute the gearbox-generator misalignment and predict the impact of this misalignment upon the gearbox HSS and generator bearings. At rated torque misalignment between the gearbox and generator of 8500 μm was seen. For the 2 *MW* WT analysed the computational data show that the L_{10} fatigue lives of the gearbox HSS bearings were not significantly affected by this misalignment but that the L_{10} fatigue lives of the generator bearings, particularly the drive-end bearing, could be significantly reduced. It is proposed to apply a nominal offset to the generator in order to reduce the misalignment under operation thereby reducing the loading on the gearbox HSS and generator bearings. The value of performing integrated systems analyses has been demonstrated and a robust methodology has been outlined.

Keywords: wind turbine, misalignment, gearbox, generator, bearings, fatigue, reliability, availability

1. Introduction

In order to minimise the cost of energy (COE) from wind it is necessary to increase reliability, and reduce unplanned downtime. The link between reliability and availability is particularly acute in the offshore wind energy sector, where even minor reliability issues can severely reduce availability due to the difficulties associated with access [1]. Unfortunately many drivetrains are suffering premature bearing failures. These bearing failures are very costly, not least because of the associated unplanned downtime.

Wind turbines, unlike traditional generation plant, experience highly stochastic loading, due to the fluctuating nature of the wind, which can be difficult to characterise. There is evidence of a positive correlation between wind speed, the standard deviation of wind speed (which is used as a proxy for turbulence) and WT failures [2-4]. Computational techniques, such as those presented in this paper, may be employed to understand the root causes of these correlations and identify potential causality.

Moreover, WTs are often exposed to extreme ambient conditions, and can be located in very remote regions on shore and offshore. This creates design challenges and also logistical challenges for operation and maintenance (O&M); reliability is, therefore, closely linked to COE [5].

Numerous studies [6-10] have been undertaken to obtain the distribution of failures by assembly in WTs. It has been found that the gearbox and generator failure rates are unacceptably high. Furthermore, the downtime for these failures is amongst the highest of all WT assemblies, because often the entire gearbox or generator needs to be replaced which requires the deployment of a large crane. These cranes are costly, and may take some days or weeks to deploy. Offshore it is not unusual for rough weather to prevent access for O&M for weeks at a time. In 2010 Chen and Alewine [11] published the findings of a survey of over 800 failed WT generators which showed that the dominant source of failures in muti-megawatt WT generators is the bearings (see Figure 1). To date, understandably, much attention has been given to blade reliability, but it is now clear that drivetrain bearing failures can have a significant impact upon the COE [12].

In practice, a full understanding of the behaviour of a WT drivetrain can only be gained by undertaking a sophisticated analysis including the 3D dynamic response. For example, Heege *et al.* [13] demonstrate that for large turbines, various sources of excitation can give rise to periodic dynamic 3D orbits of the misalignment between the gearbox and the generator. In 2011 Helsen *et al.* used advanced multi-body modelling techniques in order to describe the complex modal behaviour of large WT gearboxes [14]. However, there are significant complexities involved in undertaking a full dynamic simulation and there is recognition, in the industry, of the need for experimental validation of these multi-body simulations capable of capturing the dynamic behaviour of the turbine [15]. The output of these dynamic simulations must also be linked to a fatigue calculation and here some compromises must be made in order to minimise the computational effort of the operation. Moreover, the system dynamics will be strongly dependent on the specification of individual drivetrain components and their couplings. For these reasons, it is currently common practice to make a quasi-static assumption in the design process and, for example, in understanding drivetrain loading utilising operational data such as those obtained from SCADA [16,17].

In this paper a potential failure mechanism that does not appear to have been addressed in the literature is investigated, that of WT bearing failure due to misalignment between the gearbox and the generator. Quasi-static analyses of a complete WT drivetrain have been undertaken to calculate the gearbox-generator misalignment and then to compute the impact upon bearing loading and fatigue life. Because of the dependence of the results on drivetrain component specifications, a single (typical) configuration has been analysed and it is shown that the misalignment can have an important effect on bearing fatigue life. A mitigating strategy is presented. The methodology used provides a generalised insight from the analysis of one specific case. However, the influence of misalignment on bearing fatigue life, and the optimisation of the mitigation, will vary greatly from turbine to turbine, and will also be affected by enhancing the analysis by including dynamic effects. Thus for practical application, a full analysis is recommended for each turbine considered.

2. The drivetrain

Although many different drivetrain concepts are used in multi-megawatt WTs the most common is that shown in Figure 2. It consists of a low speed shaft supported on one or two bearings and a three stage gearbox which drives a high speed generator via a flexible coupling.

Typically the low speed shaft operates at 15-23 rpm while the high speed output of the gearbox is at 1200-1800 rpm. To achieve this gear ratio (approximately 1:80) a three stage gearbox with one planetary, and two helical, stages is commonly employed (see Figure 3).

The gearbox is mounted to the bedplate by the torque arm mounts; however the purpose of this mounting is to prevent torque roll rather than to support the weight of the gearbox which is instead cantilevered from the main bearings. Conventional WT drivetrains are supported on rolling element bearings, with those in the gearbox being oil lubricated and those in the generator greased.

3. Misalignment

Misalignment is a very common problem for rotating machinery; estimates suggest that misalignment may be the root cause of 20-30% of downtime [19]. In 2011 Whittle *et al.* presented the results of a computational parametric analysis which was undertaken to evaluate the sensitivity of the generator and gearbox HSS bearing fatigue lives to parallel misalignment [20]. This was motivated by the high failure rate of the gearbox HSS and generator bearings coupled with the insight that the rubber bushings of the gearbox torque arm mounts have a relatively low stiffness ($k_y \approx 40$ *kN/mm*, using the coordinate system shown in Figure 2). The current practice is to use relatively compliant rubber elements in the torque arm mounting arrangement to control noise and vibration. The low stiffness of these elements means that they undergo large strains under rated torque. Because in a typical three stage WT gearbox the output shaft is off-centre this torque induced rotation of the gearbox causes misalignment between the gearbox and the generator. This is illustrated in Figure 4, in which it can be seen that the rotation of the gearbox causes a linear displacement, denoted δ, in the output shaft; the misalignment, δ, is also shown in diagrammatic form in the yz plane in Figure 3.

Misalignment between the generator and the gearbox is accommodated by means of a flexible coupling. The restoring force of the coupling must be reacted at the generator and gearbox HSS bearings, as illustrated in Figure 3. The bearing types usually used for the gearbox HSS and generator are given in Table 1 [21]. In [20] the preliminary results of a computational investigation into WT gearbox-generator misalignment were outlined, but in this paper a more complete analysis is presented.

4. Bearing fundamentals

Wind turbine drivetrains are supported on rolling element bearings. In most applications as long as the bearing is correctly specified, installed and appropriately lubricated it is possible to predict the life of the bearing with some degree of accuracy.

4.1. Rolling contact fatigue

Classical fatigue damage in rolling element bearings is characterised by subsurface crack growth, as explained by Hertzian contact mechanics [22]. These subsurface cracks propagate towards the surface and eventually cause small particles to break free forming pitting damage. These particles hasten the onset of failure by causing abrasion of the raceways and rolling elements.

A range of bearing life prediction methods exist, but they are mostly based upon the assumption that in the end the bearing fails by classical rolling contact fatigue due to the cyclic nature of the loading under rolling contact [23]. In this paper the bearing fatigue lives have been calculated according to ISO 281:2007 [24] which is based upon the Lundberg-Palmgren equation [25,26].

The L_{10} life, defined as the number of revolutions (in millions) 90 % of bearings would survive before the first manifestation of fatigue damage in one of the raceways or rolling elements, is given by

$$L_{10} = a_{iso}\left(\frac{C_r}{P_r}\right)^e \qquad (1)$$

where a_{iso} is a factor which accounts for the existence of a fatigue load limit; e is a dimensionless exponent which is a function of bearing type ($e = 3$ for ball bearings and $e = 10/3$ for roller bearings); C_r is the basic dynamic radial load rating and P_r is the dynamic equivalent radial load. More details of the ISO 281:2007 bearing fatigue damage calculations are given in the Appendix, including formulae for the determination of the parameters that appear in Equation 1.

4.2. Low load requirements

In a very lightly loaded bearing the force normal to the contact is insufficient for the elastohydrodynamic tractive force to overcome the cage drag and churning losses. Therefore, the actual cage frequency is a little below the theoretical cage frequency and the raceway will skid past the rolling elements [27,28]. Whereas in Hertzian contact the maximum shear stress is below the surface, the presence of skidding can result in a larger shear stress at the surface – this renders the bearing fatigue life calculations outlined in Section 4.1 invalid. Rolling element bearing skidding is not a well understood phenomenon and consequently there is a range of different low load requirements used in industry. For deep groove ball bearings (such as those used in WT generators), one bearing manufacturer has developed the relationship

$$F_{rm} = k_r \left(\frac{\upsilon n}{1000}\right)^{2/3} \left(\frac{d_m}{100}\right)^2 \qquad (2)$$

where F_{rm} is the minimum radial load required (*kN*) to avoid skidding; k_r is the minimum load factor (this dimensionless parameter is a function of the bearing geometry and is obtained from the bearing datasheets); υ is the oil kinematic viscosity (mm^2/s) at the operating temperature; n is the number of shaft revolutions per second and d_m is the bearing mean diameter (*mm*) [29]. Other manufacturers opt for simpler and more conservative minimum load requirements of either 1% [30], or 2% [31] of the dynamic load rating of the bearing.

5. Methodology

A series of computational simulations was performed in order to:

1. Compute the gearbox-generator misalignment due to displacement of the gearbox under load
2. Predict the impact of this misalignment upon the fatigue lives of the gearbox HSS bearings and generator bearings.

A typical 2 *MW* geared WT drivetrain was modelled in RomaxWIND, a proprietary drivetrain simulation environment [32]. A system model of the WT was used to generate a twenty year time series of load data. This model included an aeroelastic description of the blades and simplified drivetrain dynamics (torsional stiffness only) to derive a set of load data that is considered representative of WT load spectra. These data were then binned according to torque, speed and the non-torsional loads (forces F_x,F_y,F_z and moments M_x,M_y) to generate forty load cases (see Figure 5), each of which could be solved quasi-statically to compute the stresses and strains in the system. The coordinate system used throughout is that shown in Figure 2.

The bearing fatigue damage for the gearbox HSS and generator bearings for each load case was computed according to ISO 281:2007 and the damage contribution for each load case was summed according to Miner's principle (see Figure 6).

In order to determine the most efficient computational model that would sufficiently accurately capture the stiffness characteristics of this complex problem, four different representations of the same 2 *MW* drivetrain were considered. In the first, the benchmark, the drivetrain was rigidly mounted; the second included the gearbox torque arm mount compliance, but neglected the compliance of the housing and bedplate; the third model included both the mount compliance and the housing compliance; the fourth model was the most detailed with the compliance of the mount, housing and bedplate included in the model. These four cases are summarised in Table 2.

In the generator model the rotor was supported on two deep groove ball bearings which were directly grounded (i.e. the generator housing and bedplate compliance were neglected), since the misalignment between the generator and gearbox is likely to be dominated by the gearbox displacements under operational loads.

The final, but important, parameter in the model described in this section is the tilt stiffness prescribed for the flexible couplings. There is a range of flexible couplings on the market; the type used in WT drivetrains typically gain their flexibility by means of two arrangements of links (one at each end). The tilt stiffness of the flexible coupling link set adopted for the numerical model of the drivetrain was 5 *kNm/rad*, i.e. that of the Centalink 71 coupling [33].

5.1. Assessing mitigation strategies

Two strategies for mitigating the effects of gearbox-generator misalignment were considered:

1. Over-rating the generator bearings. Three different ratings were considered for the generator bearing, varying from a lean design (bearing designation 61930) to a conservative approach (bearing designation 6330); see Table 3 for details.
2. Imposing a nominal offset on the generator to reduce operational misalignment (i.e. apply static misalignment to the generator).

6. Results

6.1. Displacement

The magnitude of the misalignment between the gearbox HSS and the generator shaft in the xy plane (axial displacement is neglected) for the four different modelling cases, presented in Table 2, is shown in Figure 7. In case 1 (rigidly mounted) a misalignment of approximately 1000 μm is observed under rated torque; this misalignment is due to the compliance of the bearings and the shaft under significant radial loading at the HSS gear set. By comparison of the magnitude of misalignment for the other cases it may be seen that the housing and bedplate compliances have a small influence upon the gearbox-generator misalignment (<300 μm). In the analysis that follows the full model is used (case 4) but such a detailed model is not essential. The gearbox housing and bedplate compliances may be neglected where these data are not available, or where computational or time constraints are severe, when computing gearbox-generator misalignment since, as may be seen from Figure 7, it is mainly a function of the torque arm mount stiffness.

6.2. The bearing loads

Having seen that significant misalignments – 8500 μm at rated torque – between the gearbox and generator are caused by rotation of the gearbox under torsional loading this section considers the effect of the misalignment upon the loads transmitted through the bearings and thence on their fatigue lives. Figure 8 shows the radial and axial loads on the gearbox HSS bearings with no generator nominal offset (0 μm) and a large nominal offset of the generator (10,000 μm). Applying a nominal generator offset of 8500 μm aligns the drivetrain for rated torque; the bearing reaction forces for this case lie in between the two graphs (0 μm and 10,000 μm) shown in Figure 8, but for clarity these data were not graphed.

For the gearbox HSS bearings the loading is a linear function of the absolute torque. These bearing forces are reacting the contact forces in the HSS helical gear set. At rated torque inclusion of misalignment in the computation changes the radial load on HSS A and HSS B by 20% and 15% respectively (refer to Figure 3 for the definition of HSS A and HSS B); i.e. at rated torque the HSS bearing loading is primarily due to the gear contact.

For a 2 *MW* DFIG the generator bearings typically have a load of 9.8 *kN* to 12.3 *kN*. The radial loading on the generator bearings is only about 33% of that on the upwind HSS bearing (HSS A), or 19% of the force on the HSS downwind bearing (HSS B). Thus, whilst the restoring forces of the coupling under misalignment have a small influence upon the gearbox HSS bearing loads the same cannot be said for the loading of the generator bearings, as shown in Figure 9. It may be seen that the generator drive-end (DE) bearing loading is, as expected, much more sensitive to misalignment than the non-drive end (NDE) bearing. When the system was statically aligned and operating at rated torque the generator DE bearing load was 160% that of the NDE bearing, i.e. the system was very imbalanced.

The graphs in Figure 9 show the generator bearing reaction forces as a function of torque for a range of nominal generator offsets (0 – 10,000 μm) in the y direction. The low load requirements for the conservative design choice (bearing designation 6330, see Table 3) are also plotted on Figure 9 – under all but the most severe conditions the low load requirements are met. However, low load phenomena are not well understood, therefore a greater factor of safety, with regards to this requirement, may be preferred by some OEMs.

6.3. Mitigation

6.3.1. Over-rating the generator bearings

A simple approach to mitigate misalignment is to increase the bearing L_{10} fatigue life by increasing the bearing rating. This option is attractive for two reasons:

- The generator deep groove ball bearings are relatively inexpensive, so if costly downtime can be avoided by using larger, more expensive, bearings the cost-benefit analysis is likely to be favourable
- It is a simple solution requiring very little detailed system knowledge

Three bearing selections were considered for the 2 *MW* generator in this test case. For the twenty year simulation the accumulated fatigue damage was calculated for each bearing choice and these data are shown in Table 4.

It may be seen that the smallest bearing selected has very imbalanced fatigue damage for the DE and NDE bearings due to the effect of misalignment. The second option, bearing 6230, still betrays the presence of misalignment with the DE bearing having twice the fatigue damage of the NDE bearing. The most conservative design choice, bearing 6330, has a near infinite life.

However, although over-rating the bearing will give a longer L_{10} fatigue life, it may actually reduce the serviceable life of the bearing if the load is insufficient to prevent the bearing from skidding [34]. This is particularly problematic in WT generators because the bearings must be able to withstand significant thermal growth. Upon start-up the bearing temperature will increase more rapidly than the surrounding housing, which has a large thermal inertia, and this leads to a loss of operating clearance [35]. The bearings must have a large enough internal clearance to accommodate this thermal expansion but this makes the rolling elements more prone to skidding. The necessity of ensuring sufficient operating clearance under thermal instability therefore makes WT bearings more likely to suffer from skidding and so increasing the probability of skidding still further by over-rating bearings may be inadvisable.

6.3.2. Nominally offset generator

An alternative strategy is to apply a nominal offset to the generator such that the system moves into alignment under the application of a load. Such a system might be designed to become fully aligned under rated torque. However, here the prospect is considered that the optimum fatigue life may arise for a system that has a somewhat different nominal offset.

The effect of applying a nominal offset to the generator upon its bearing fatigue lives is shown in Figures 10 and 11 which compare the fatigue damage accumulation in the forty simulated load cases for the drivetrain statically aligned (offset = 0 μm) and when it is aligned for the rated torque (offset = 8500 μm). Aligning the drivetrain for the rated torque not only reduces the fatigue damage on the DE bearing at high torque, but also reduces it at low torque levels. This is because the misalignment at low torque unloads the DE bearing so it comes at the cost of slightly increased damage to the NDE bearing.

The influence of the nominal generator offset upon the bearing fatigue damage accrued over the twenty year simulation, for the gearbox HSS and generator (for bearing 61930 and 6230), is shown in Figure 12. The gearbox HSS bearings are not significantly affected by misalignment because the coupling reaction forces are small compared to the gear loading; the generator bearings, particularly the DE bearing, are sensitive to misalignment. It is interesting to note that for the leanest generator specification (bearing 61930) nominally offsetting the generator is sufficient to make the difference between meeting the target L_{10} bearing fatigue life or not.

The more conservative design, using a 6230 type bearing, should not encounter fatigue problems even when statically aligned. However, it may still be better to offset the generator to reduce vibration and reduce the probability of coupling failure.

The optimal generator offset is a function of the specific drivetrain configuration, and a similar analysis to that presented here would have to be undertaken for each drivetrain. In particular it is important that the torque arm mount and flexible coupling stiffnesses are correctly specified since the model is sensitive to these parameters. Moreover, the optimal generator offset is also a function of the wind conditions for the particular WT location. Different wind conditions will give a different shape to the load data histogram (Figure 5) – for a given WT high wind speed sites will, therefore, require larger generator offsets than low wind speed sites. In fact, it is not a simple matter to define the optimal generator offset even for a given WT subjected to known loading. By reference to Figure 12 it may be seen that alignment for the rated torque gives imbalanced generator bearing fatigue lives. This is because the variable nature of the wind means that the WT is not operating at rated power for much of the time. Therefore, a better solution may be to align the turbine for 70% of rated torque which yields balanced fatigue damage for the generator bearings, albeit resulting in imbalanced loading at rated operation.

7. Conclusions

For a typical 2 *MW* geared WT significant misalignment between the gearbox and the generator results from the compliance of the gearbox torque arm mounts. Depending upon whether this misalignment loads, or unloads, the generator DE bearing it could decrease the bearing L_{10} fatigue life, or increase the probability of failure due to skidding. The NDE bearing could also be affected by misalignment, but gearbox bearings are less likely to be affected by radial misalignment as they must be rated for significantly greater loading from the HSS helical gear set.

In modelling global gearbox displacement the gearbox torque arm mount stiffness is critical. The gearbox housing and bedplate compliances may be neglected for the present purpose if necessitated by computational constraints or unavailability of data. The need for integrated system analyses of WT drivetrains has been demonstrated, and this requires OEMs to collaborate in order to understand the interactions between WT subassemblies.

The present quasi-static analysis showed that the interactions between drivetrain assemblies are important, and that large gearbox-generator misalignments can occur which may cause increased fatigue damage to the generator bearings. Promising advances are being made in the application of multi-body simulation techniques capable of describing the dynamic behaviour of WT drivetrains. In the future the analysis could be enhanced by incorporating dynamic effects in order to quantify the influence of dynamic loading upon the gearbox HSS and generator bearings. Of particular interest may be the contribution that extreme load events, such as emergency stops, make to the gearbox HSS and generator bearing fatigue lives.

The practice of over-rating bearings to increase their L_{10} fatigue life is not recommended for WT generators. These bearings need large internal clearances to accommodate the thermal instability inherent in this application; this makes these bearings prone to skidding which is one possible cause of premature failure. Therefore, it is proposed that gearbox-generator misalignment be mitigated by means of a nominal generator offset rather than over-rating the generator bearings. This requires a careful analysis of the drivetrain, but the potential to increase availability, thereby reducing the risk to investors, may reduce the COE enough to justify the additional capital cost involved.

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8. Appendix: Calculating bearing fatigue lives with ISO 281:2007

8.1. Radial ball bearing ratings

The basic dynamic radial load rating, C_r, (in Newtons) for a radial ball bearing is

$$C_r = b_m f_c (i \cos \alpha)^{0.7} Z^{2/3} D_w^{1.8} \qquad (3)$$

if $D_w \leq 25.4$ mm and

$$C_r = 3.647 b_m f_c (i \cos \alpha)^{0.7} Z^{2/3} D_w^{1.4} \qquad (4)$$

if $D_w > 25.4$ mm, where b_m is the rating factor (dimensionless parameter, varies with bearing type and design), D_w is the nominal ball diameter (*mm*), f_c is a factor which depends on the bearing geometry (dimensionless), i is the number of rows of rolling elements, Z is the number of rolling elements per row and α is the nominal contact angle (degrees).

The dynamic equivalent radial load, Pr, is defined by the empirical relationship

$$P_r = XF_r + YF_a \qquad (5)$$

where F_r and F_a are the actual radial and axial bearing loads (N) and X and Y are dimensionless factors the values of which are dependent upon the bearing type and geometry.

8.2. Radial roller bearing ratings

For radial roller bearings the approach is the same, but the equations are modified for line contact. The basic dynamic radial load rating becomes

$$C_r = b_m f_c (i \, \mathrm{L}_{we} \cos \alpha)^{7/9} Z^{3/4} D_{we}^{29/27} \qquad (6)$$

the new parameter, L_{we}, is the effective roller length (*mm*) which is the theoretical maximum length of the contact between a roller and whichever raceway has the shortest contact. The dynamic equivalent radial load is calculated according to Equation 6 unless $\alpha = 0^{o}$ in which case it becomes

$$P_r = F_r \qquad (7)$$

8.3. Calculating the fatigue life

The L_{10} life, defined as the number of revolutions (in millions) 90 % of bearings would survive before the first manifestation of fatigue damage in one of the raceways or rolling elements, is defined as

$$L_{10} = a_{iso} \left(\frac{C_r}{P_r}\right)^e \qquad (8)$$

where e is dimensionless exponent which is a function of bearing type; $e = 3$ for ball bearings and $e = 10/3$ for roller bearings. The factor a_{iso} is incorporated as part of the 2007 amendment to ISO 281 to account for the existence of a fatigue load limit thereby yielding more accurate fatigue life calculations at low loads. a_{iso} captures the S-N characteristic of the raceway and rolling element metals so it can be said that it is a function of the fatigue stress limit, σ_u, and the actual stress, σ, experienced by the raceway

$$a_{iso} = f\left(\frac{\sigma_u}{\sigma}\right) \qquad (9)$$

For practical purposes it is easier to equate this to the loads on the bearing by introducing the fatigue load limit, C_u, thus

$$a_{iso} = f\left(\frac{C_u}{P}\right) \qquad (10)$$

where C_u is defined as the load at which the fatigue stress limit is just reached at the most heavily loaded raceway contact. The value of C_u is a function of the internal geometry of the bearing, the manufacturing quality and the fatigue limit of the raceway steel. Reference [24] should be consulted for a full account of the bearing life calculations including more details on the calculation of a_{iso}.

Tables

Bearing	Type
HSS A	Cylindrical or tapered cylindrical rolling element bearing
HSS B	A single row or back-to-back tapered cylindrical rolling element bearing(s)
Gen DE	Deep groove ball bearing
Gen NDE	Deep grove ball bearing

Table 1 Gearbox HSS and generator bearing types

Case	Mount	Housing	Bedplate
1	Rigid	Rigid	Rigid
2	Compliant	Rigid	Rigid
3	Compliant	Compliant	Rigid
4	Compliant	Compliant	Compliant

Table 2 Drivetrain models

Bearing	C (kN)	C_0 (kN)	C_u (kN)	0.01C (kN)	0.02C (kN)
61930	88.4	93	2.9	0.884	1.768
6230	174	166	4.9	1.74	3.48
6330	276	285	7.8	2.76	5.52

Table 3 Ball bearing data used in generator model (C is the dynamic rating, C_o is the static rating, C_u is the fatigue load limit)

Bearing	Fatigue damage (%) (zero offset)	
	Gen DE	Gen NDE
61930	250	50
6230	20	10
6330	3	2

Table 4 Generator bearing fatigue damage for three different bearing selections over simulated twenty year design life

Figures

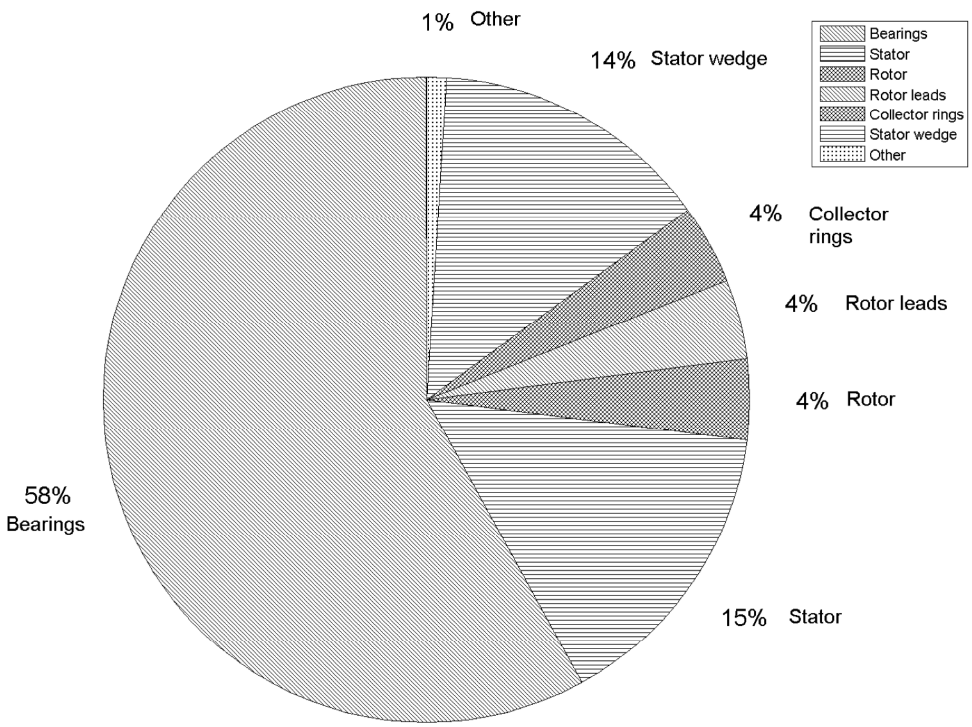

Figure 1 Distribution of failures in WT generators rated > 2 MW (taken from data published in [12])
380x281mm (300 x 300 DPI)

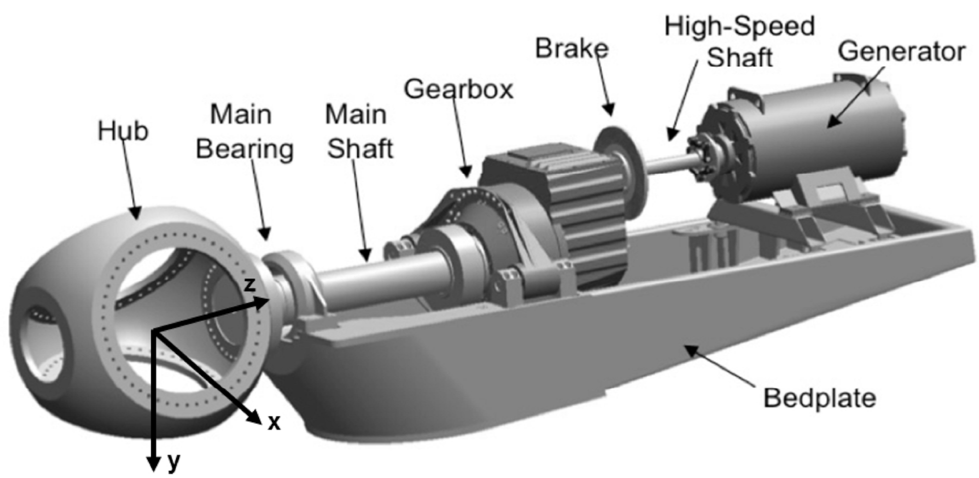

Figure 2 Typical multi-MW WT drivetain (reproduced from [18], the coordinate system has been added)
190x143mm (300 x 300 DPI)

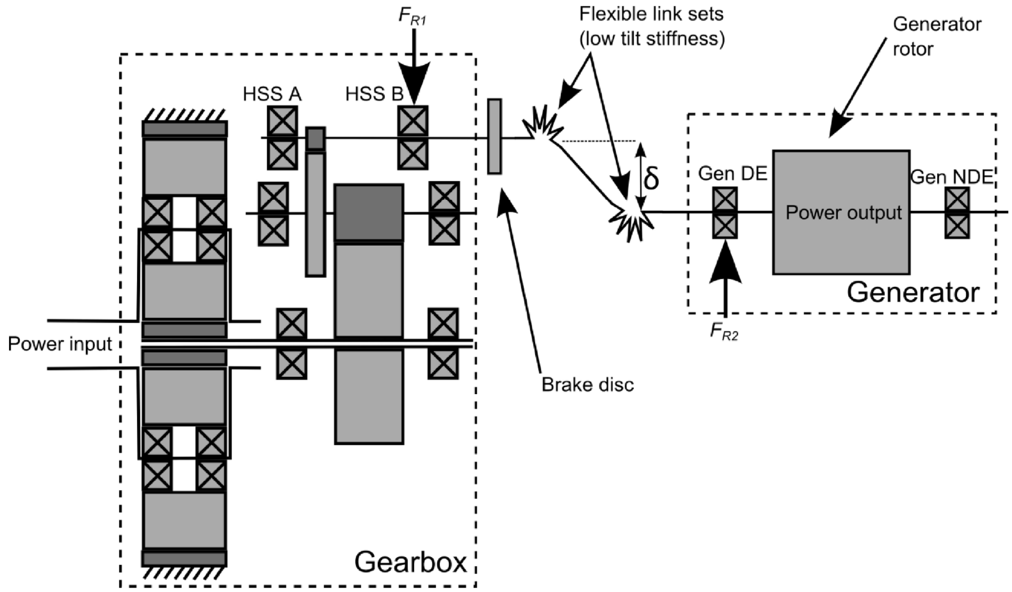

Figure 3 Three stage WT gearbox flexibly coupled to the generator (F_{R1} and F_{R2} indicate reaction forces caused by misalignment; δ denotes the misalignment)
345x201mm (300 x 300 DPI)

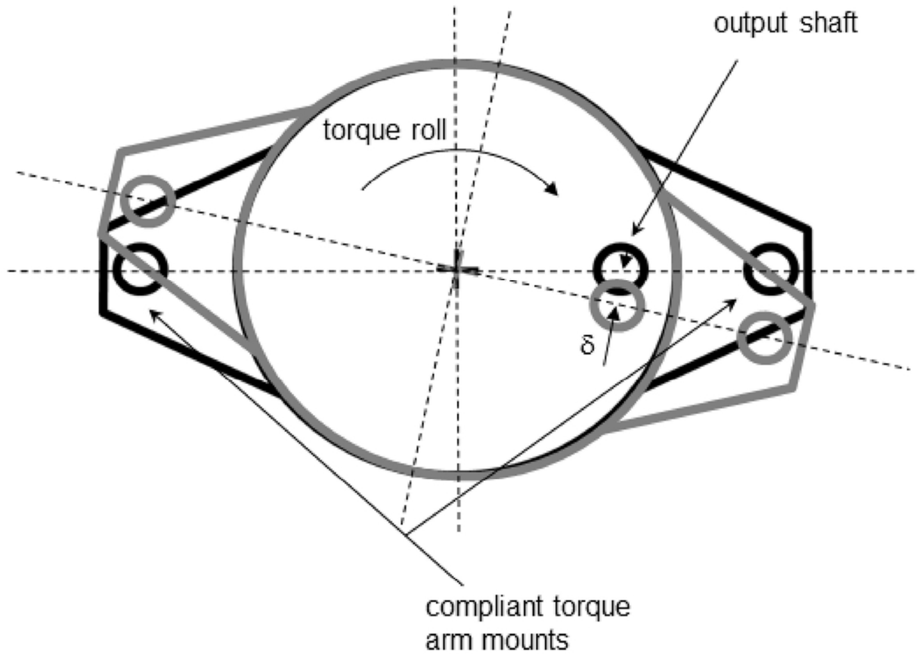

Figure 4 Displacement of gearbox under load induces gearbox-generator misalignment
104x72mm (300 x 300 DPI)

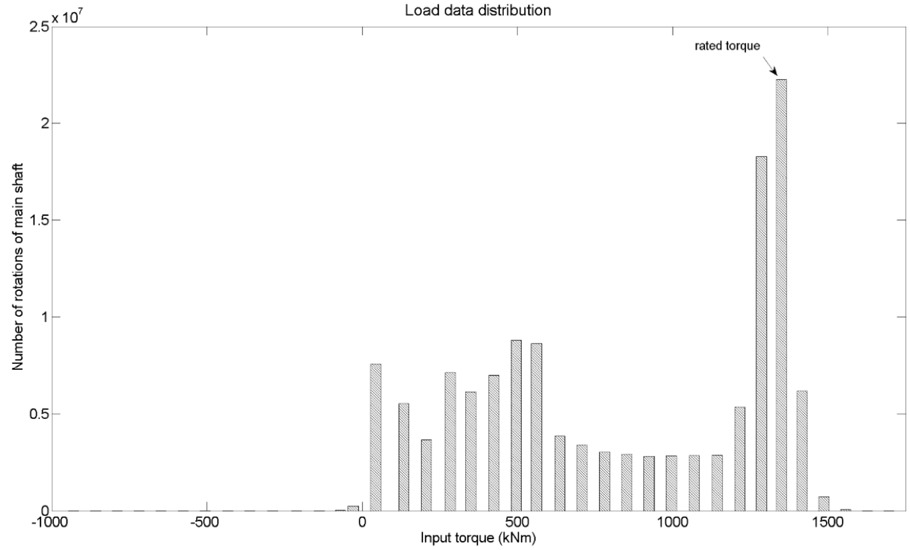

Figure 5 Twenty years of simulated load data binned according to torque, speed and non-torsional loads
305x174mm (300 x 300 DPI)

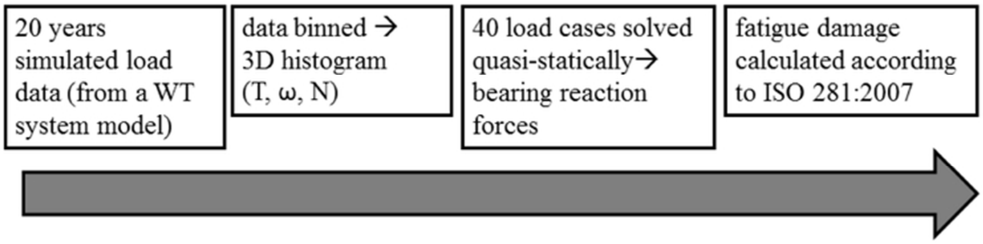

Figure 6 Computational strategy
63x17mm (300 x 300 DPI)

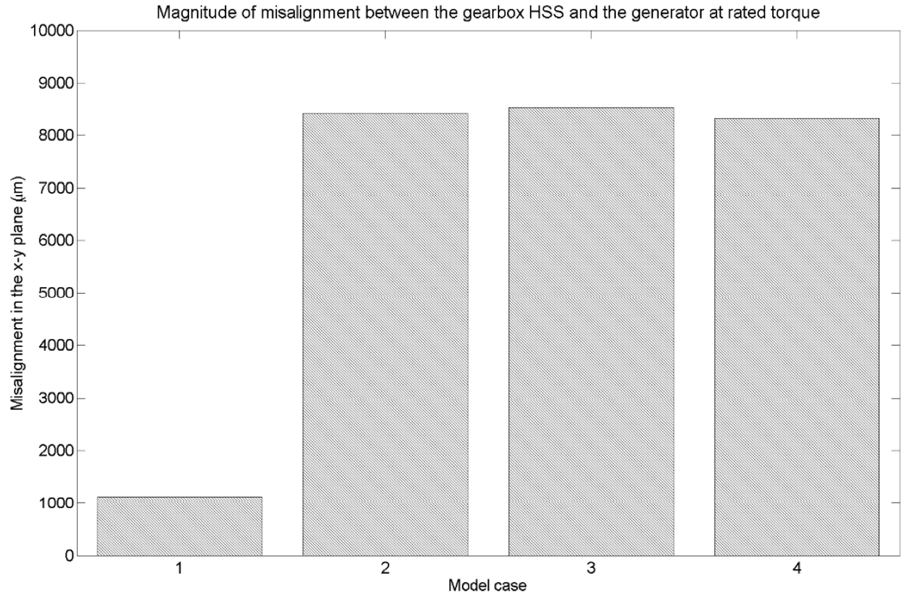

Figure 7 Impact of component compliances
302x184mm (300 x 300 DPI)

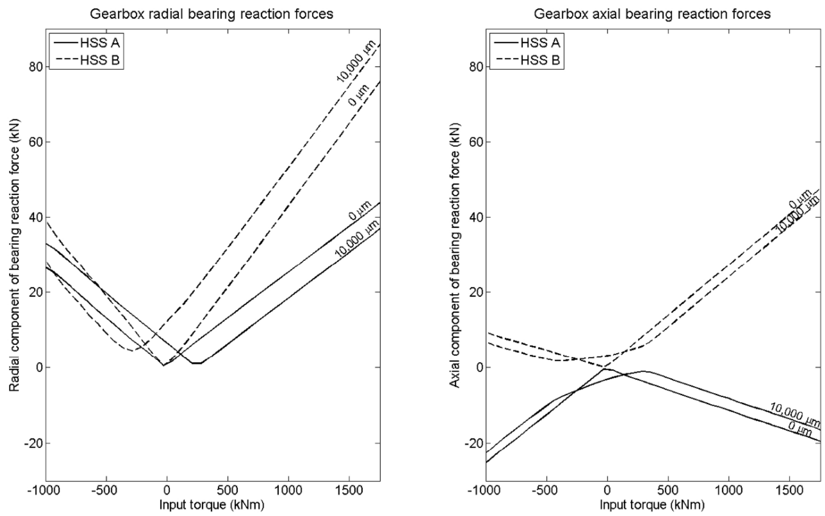

Figure 8 Gearbox HSS bearing reaction force as a function of torque
247x137mm (300 x 300 DPI)

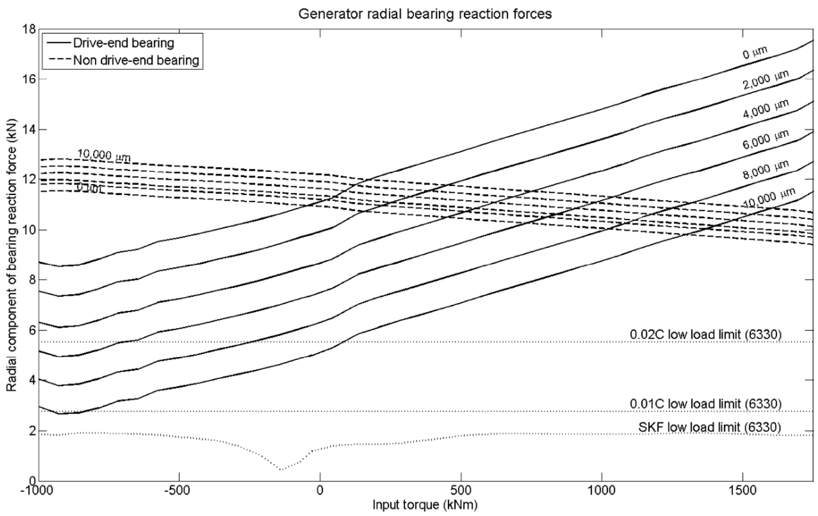

Figure 9 Generator radial bearing reaction forces
247x137mm (300 x 300 DPI)

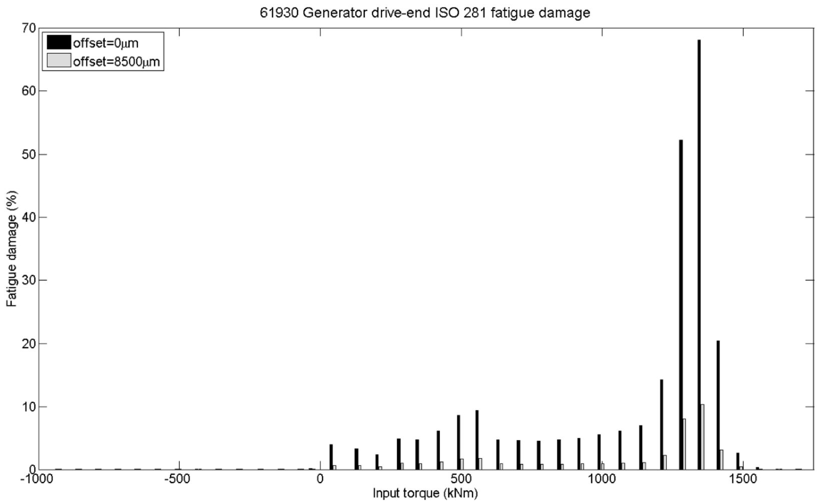

Figure 10 61930 generator drive-end ISO 281 fatigue damage
240x130mm (300 x 300 DPI)

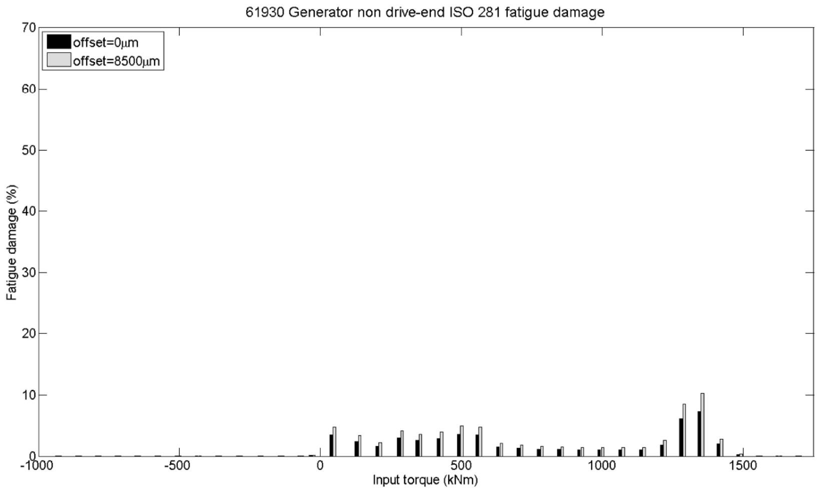

Figure 11 61930 generator non drive-end ISO 281 fatigue damage
234x124mm (300 x 300 DPI)

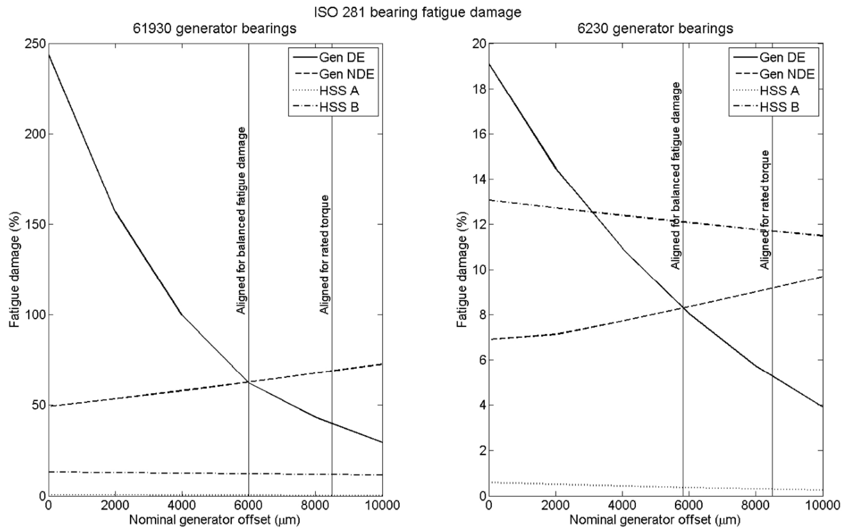

Figure 12 The sensitivity of the ISO 281 bearing fatigue damage to the generator offset
247x137mm (300 x 300 DPI)