Tribological advancements for reliable wind turbine performance

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Wind turbines have had various limitations to their mechanical system reliability owing to tribological problems over the past few decades. While several studies show that turbines are becoming more reliable, it is still not at an overall acceptable level to the operators based on their current business models. Data show that the electrical components are the most problematic; however, the parts are small, thus easy and inexpensive to replace in the nacelle, on top of the tower. It is the tribological issues that receive the most attention as they have higher costs associated with repair or replacement. These include the blade pitch systems, nacelle yaw systems, main shaft bearings, gearboxes and generator bearings, which are the focus of this review paper. The major tribological issues in wind turbines and the technological developments to understand and solve them are discussed within. The study starts with an overview of fretting corrosion, rolling contact fatigue, and frictional torque of the blade pitch and nacelle yaw bearings, and references to some of the recent design approaches applied to solve them. Also included is a brief overview into lubricant contamination issues in the gearbox and electric current discharge or arcing damage of the generator bearings. The primary focus of this review is the detailed examination of main shaft spherical roller bearing micropitting and gearbox bearing scuffing, micropitting and the newer phenomenon of white-etch area flaking. The main shaft and gearbox are integrally related and are the most commonly referred to items involving expensive repair costs and downtime. As such, the latest research and developments related to the cause of the wear and damage modes and the technologies used or proposed to solve them are presented.

Keywords: wind turbines; tribology; engineered surfaces; rolling element bearings

1. Introduction to the tribology of a wind turbine

Although current turbine designs have addressed and solved a number of the problems plaguing their predecessors, tribological issues still exist. This is evident from the data of Bell (2006), shown in figure 1, which indicate that the number of failures per turbine per year in Denmark and Germany dropped from 1 and 2.5, *Author for correspondence (michael.kotzalas@timken.com).

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Figure 1. Data from Bell (2006) indicate that the number of failures per turbine per year in Denmark (grey circles) and Germany (black triangles) dropped from 1 and 2.5, respectively, in 1994 to 0.5 and 0.8 in 2004.

respectively, in 1994 to 0.5 and 0.8 in 2004. This represents a significant improvement in wind turbine reliability over the course of a decade; however, this number still indicates one failure per turbine every 15 months to 2 years. Faulstich et al. (2009) have concluded that the least reliable elements of wind turbines are the electrical system and electronic controls, each with more than 0.5 failures per year in typical wind turbine designs. Their annual system failure rate data are provided in figure 2 as an average over the turbine designs reported for each subsystem.

It is evident from figure 2 that the systems containing tribological components typically associated with wind turbine reliability—the pitch, yaw, generator and gearbox systems—experience less than 0.2 failures per year. Although electrical components tend to be small, relatively inexpensive and easily replaceable, the gearbox, pitch, yaw and generator systems are very large, have great mass and require large cranes and expensive rigging equipment when undergoing repair or replacement. Including the lost production owing to downtime, the crane rental costs and the system replacement costs, it becomes clear that a tribological failure in one of these systems becomes very expensive. These maintenance, reliability and operating (MRO) expenses significantly affect the overall generating cost per kW-hour and, therefore, the competitiveness of wind in the overall energy marketplace.

The most common wind turbine design is referred to as modular since the mechanical systems are split into segments or modules. This design is shown in figure 3 (Oyague 2009). The blades connect through the pitch bearings to the rotor hub, and the rotor hub connects to the main shaft, which is supported by large rolling element bearings or main shaft bearings. The main shaft connects to the gearbox, which takes the low-speed, high-torque of the wind and increases speed up to the typical 1500–1800 r.p.m. required for the generator to connect to the local electric grid. All of this sits in an enclosure, or nacelle, which is mounted to the tower through the yaw bearing.
Newer turbine architectures have been developed in recent years that eliminate the gearbox and connect the main shaft directly to a permanent magnet generator. This design style is often termed direct drive, and is schematically illustrated in figure 4 (Oyague 2009). Direct drive turbines use solid-state electronics to control the electrical connection requirements to the grid instead of using the output speed of the gearbox. Both direct drive and modular wind turbine designs still face tribological issues with critical components in order to decrease MRO costs due to reliability issues.

In the following sections, we will attempt to summarize the functions and the tribological challenges facing the primary systems of a wind turbine generator. Additionally, we will also discuss some of the more recent solutions that have been enacted or proposed to address the tribological issues that currently limit the reliability of wind turbine generators.

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2. Blade pitch and yaw systems

Mechanical and tribological issues limit the reliability of blade pitch and nacelle yaw systems in both modular and direct drive wind turbines. These systems consist of large slewing bearings with integral gear teeth and a geared motor that accurately turns the system. The pitch system adjusts the blade angle relative to the wind flow in order to control the torque and speed of the main shaft for optimal generator performance. The function of the yaw system is to keep the nacelle faced into the wind. Marson (2010) quotes one pitch and yaw system manufacturer’s viewpoint that ‘technology is not as important in these systems as reliability’. However, it may not be possible to significantly improve the reliability of pitch and yaw systems using current technological practices.

Pitch and yaw bearings are typically an eight-point contact ball bearing, a four-point contact ball bearing or a cylindrical crossed roller bearing. Four-point contact ball bearings can accept combinations of radial, thrust and moment loads, which is made possible by the geometry of the raceways. Each raceway ball groove has two radii that are larger than the ball radius, with the centres of these two radii offset from the centre of the ball radius. This creates a ‘Gothic arch’ configuration in each of the raceway grooves, making it possible for the two grooves to contact the ball at four points.

An eight-point contact ball bearing is an annular bearing with two rows of balls using a four-point contact internal geometry in both rows. Crossed roller bearings are designed with V-groove raceways, providing two roller paths in each ring. The rollers have a length slightly less than their diameter and are positioned so that adjacent rollers contact different sets of raceways, with the axes at right angles to each other. In this geometry, the rollers transmit load along perpendicular sets of 45° contacts; therefore, the action of this bearing under various types of loading is similar to that of the four-point contact ball bearing. All assemblies are pre-loaded axially to ensure a stiff system without play and incorporate seals to prevent ingress of water and contamination.

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Figure 5. Example of false brinelling wear in a rolling contact bearing.

Yaw and pitch bearings are case-hardened and the heat-treatment process is intended to produce a raceway hardness of 58 on the Rockwell C scale (HRC). Through the ‘case’, the hardness decreases gradually to HRC 50, and then decreases rapidly to the core hardness of the ring material. The application of a concentrated (Hertz contact) ball or roller load results in significant subsurface shear stresses, which reach down into the core material. Because of the size of the bearings, it is becoming more common to use medium carbon steels induction hardened for increased rolling contact fatigue of the surfaces. In these steels, hardness values might fall in the HRC 56 range, which would statistically correlate to about 70 per cent of the predicted L₁₀ life of HRC 58 bearings according to Harris et al. (2009).

Ball and roller bearings used for wind turbine yaw and pitch bearings are thrust-type bearings. The principal load is an eccentrically applied thrust, which results in an axial load and an overturning moment load. If these loads are not distributed satisfactorily over the case and core of the bearings, raceway deformations can occur.

Lai et al. (2010) studied the effect of case depth and core hardness of the steel on static load ratings of large, induction-hardened bearings. They found that the tensile residual stresses are greatest at the case–core interface, and that the stresses are dependent on the case depth, rolling element diameter, yield strength of the core material and the applied contact load. Under large static loads, these bearings may experience core crushing where cracks initiate at the location of maximum tensile residual stress and propagate to the surface. The challenging implications of this are that maximum case depths do support loads similar to those of small-bore, high-volume bearings and shallower case depths cannot support as much load without significant permanent deformation.

The tribological issues that limit the reliability of pitch and yaw systems are false brinelling, fretting corrosion and friction. Errichello (2004) has described the difference between the false brinelling and fretting corrosion and how they occur in wind turbines during idle times. An example of this type of bearing wear is shown in figure 5.

Fretting is known to occur in pitch and yaw systems when the bearings and gears are not rotating and are subjected to structure-borne vibrations caused by wind loads and/or small motions from the control system, termed
dither. Under these conditions, lubricant is squeezed from between the contacts and the relative motion of the surfaces is too small for the lubricant to be replenished. Natural oxide films that normally protect steel surfaces are removed, permitting metal-to-metal contact and causing adhesion of surface asperities.

Fretting begins with an incubation period during which the wear mechanism is mild adhesion and the wear debris is magnetite ($\text{Fe}_3\text{O}_4$). Damage during this incubation period is referred to as false brinelling. If wear debris accumulates in amounts sufficient to inhibit lubricant from reaching the contact, then the wear mechanism becomes severe adhesion that breaks through the natural oxide layer and forms strong welds with the steel. In this situation, the wear rate increases dramatically and damage escalates to fretting corrosion. Relative motion breaks welded asperities and generates haematite ($\alpha\text{-Fe}_2\text{O}_3$)—a fine powder that is reddish-brown in colour.

Harris et al. (2009) created a full bearing design guide to mitigate the risk of failure of pitch and yaw bearings due to mechanical reasons by considering the fatigue life and deformation calculations for the induction-hardened steels in oscillating applications with flexible rings. They also gave specific design parameters to prevent fretting corrosion and false brinelling. The critical dither angle of rotation for the bearing is defined in equation (2.1) as

$$\theta_{\text{dither}} = \frac{720^\circ b}{\pi Z (1 \mp (D \cos \alpha / d_m))}, \quad (2.1)$$

where the negative sign refers to the inner raceway, $b$ is the Hertzian half-width of contact, $Z$ is the number of rolling elements per row, $D$ is the rolling element diameter, $\alpha$ is the contact angle and $d_m$ is the bearing pitch diameter. When $\theta \leq \theta_{\text{dither}}$, fretting corrosion is likely to occur. Harris et al. (2009) recommend to avoid operating the pitch and yaw bearings under these very small oscillations. They also advise rotating the bearings as often as possible to redistribute grease to the rolling element contacts. Greases with adequate base oil viscosities and good anti-wear additives should be used as the lubricant. Additionally, it could become necessary to coat the raceways with hard coatings and/or solid lubricants (Leonard et al. 2010).

The frictional torque of the pitch and yaw bearings is highly dependent on the lubrication as it is required for consistent, low-friction operation. Too much lubrication may cause churning and, therefore, increased bearing torque, and too little increases the metal-to-metal contact of the surfaces and the bearing torque.

To investigate these issues, Gonzalez et al. (2009) studied the four-point contact ball bearing load distributions and how this affected overall bearing friction torque, with an overall goal to develop an understanding of the relationship between the frictional torque and lubrication. Similar studies have been conducted by Leblanc & Nelias (2007) on how contact loading in all 5 d.f. affected individual ball loading and therefore the frictional contacts. To address these types of failure modes, Garcia (2008) invented a method to dynamically re-lubricate pitch bearings based upon the operating conditions and motions that the system has experienced.
3. Gearbox lubrication systems

One issue that has been studied by several researchers deals with contamination in the gearbox lubricant. As the modular design is the only architecture using a gearbox, the problem is specific to this design style. The major contaminant in older turbines was hardened wear particles from the bearings and gears; however, more recent issues have revolved around water ingress, or high water concentration in the gear oil.

Hard particle debris in a tribological system can cause damage to the contacting components, making it difficult to achieve the design life expectation. This was shown by Sheng (2010), Holzhauer (1991) and Ai & Nixon (2000). Both Holzhauer (1991) and Ai & Nixon (2000) found that surface dents from hardened debris particles can lead to micropitting, which is a significant wear mode found in turbine gearbox bearings and gears today.

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Not covered in this article is the requirement for lubricant filtration. The filtration is important in the initial fill, for which Errichello & Muller (2002) suggested using a $3\mu m$ filter with efficiency of at least $\beta_3 > 200$. Further, they suggested that new oil added to a gearbox should meet International Organization for Standardization (ISO) standard ISO 4406 (1999) specification of 16/14/11, and during service should be maintained at 18/16/13.

Water is the other large culprit in wear damage of turbine gearbox components. Water causes the lubricant effective viscosity to change and therefore the film formation capability in the gear and bearing contacts. The damage can be significant. Cantley (1977) provided the following equation for bearing fatigue life as a function of lubricant water ingress:

$$LF = \left(\frac{100}{X}\right)^{0.6},$$

where $LF$ is the life ratio due to water contamination and $X$ is the amount of water in the lubricant measured in parts per million (ppm). Water can also lead to decomposition of ester-based fluids and additives, as well as other particulate-based anti-wear additives, which can lead to massive additive dumping. The additive dropout can foul sensors and clog filters, which can lead to more serious problems with the turbine system.

To mitigate the risk of water in the lubricant, Needleman et al. (2009) explored the use of a low relative humidity air blanket in the lubricant sump to remove water from the system. They also suggested the use of regenerative drier filters on the gearbox breather to prevent humid air from bringing water into the gearbox as it cools. Use of this type of drier system can bring a gearbox operating with 400–500 ppm water down to, and maintain it at, 175 ppm water.

4. Generator systems

In the generator, high-frequency currents are induced on the shaft. If not diverted or otherwise insulated, the stray currents that develop in the generator pass through the rolling elements causing extreme local heat and small burnt pits in the bearing surfaces such as the case shown in figure 6.
Figure 6. Electrical current damage on the outer raceway of a bearing: (a) photograph of the body raceway surface; and (b) cross section of a pit, mounted, polished, natal-etched and magnified 600× showing the removed material, reharden and retemper layers. Scale bar, 20 μm.

One solution has been to use ceramic balls for rolling elements in the bearing, otherwise known as hybrid bearings. Even with the use of hybrid bearings, some turbines have still experienced this type of damage. Srinidhi et al. (2009) studied the retainer and lubricant’s role in the development of arcing damage. They found a brass retainer, polymer retainer and no retainer in the bearing all experienced similar arcing damage; however, there was a significant difference between oil and grease. Oil was able to prevent damage up to higher voltages applied through the bearing. Therefore, oil lubrication would seem to be more effective in this position.

5. Other systems

Though the above systems may present challenges to the tribologist, the main shaft bearings and gearbox still represent the key issues for reliability. Owing to the larger body of research on these topics, they will be covered in detail in the following sections.

6. Main shaft bearings

(a) Micropitting wear

Spherical roller bearings are most commonly used to support the main shafts in current wind energy turbine modular designs because of their ability to accommodate misalignment between the shaft and the bearing housing. However, many of these bearings are experiencing damage from wear that reduces their serviceable life to much less than that for which they were designed. This life-limiting wear leads to expensive downtime and excessively high maintenance and warranty costs.

The wear that limits the life of main shaft spherical roller bearings is not classical rolling contact fatigue, but predominately micropitting wear. Micropitting is caused by interaction of the raceway and roller residual finishing marks, or asperities, leading to high stresses in the part.

The normal stress alone is not typically sufficient to cause a crack to initiate at or very near to the surface early in the life cycle of a bearing. However, the addition of frictional shear stress increases the bulk contact stress values and
Figure 7. Micropitting of 230/600 series spherical roller main shaft bearings: (a) onset of micropitting wear at the centre of the inner raceway where its velocity is slower than the roller; and (b) advanced micropitting wear that caused geometric stress concentration spalling initiated at the edge of the wear track.

brings the maximum values closer to the surface, as shown by Harris & Yu (1999), allowing these localized stresses under the asperity contacts to become significant. This type of interaction typically occurs when the lubricant film is insufficiently thick to separate the contacts and when there is relative sliding between the two contacting surfaces based on Averbach & Bamberger (1991), Ueda et al. (2005), Webster & Norbart (1995) and Chiu (1997), and wear resulting from this type of interaction is termed low-cycle micropitting.

The evolution of low-cycle micropitting to raceway spalling of a 230/600 main shaft spherical roller bearing is shown in figure 7. Figure 7a shows the onset of micropitting, where two distinct wear tracks have emerged in the centre of the raceway. As the micropitting continues, more and more material is worn away, leading to a loss of the design contact geometry in the centre and increasingly higher stress concentrations at the edges of the wear track.

Fatigue spalls initiate at these areas of high geometric stress concentrations and propagate to the centre of the raceways such as that shown in figure 7b. The raceway spalling of these bearings is not because of classic surface-initiated or inclusion-related fatigue on which the predicted bearing life is calculated, but is due instead to the loss of the designed contact geometry owing to micropitting wear and a concomitant increase in geometric stress concentrations at the edges of the roller/raceway contact.

Low-cycle micropitting is caused by high amounts of sliding between rollers and ring raceways generating considerable shear stresses in the contact zone. From the viewpoint of an element of area on a ring raceway, the cyclic shear stresses imparted by each passing roller ultimately generate microcracks, which
propagate in the direction of the sliding shear stress, on the slower moving component according to Ueda et al. (2005) and Webster & Norbart (1995). As these microcracks propagate, pieces of the raceway begin to break away from the surface leaving pits that are micrometres in size.

A cross-sectional scanning electron microscopy (SEM) image of a raceway showing microcracks and the onset of micropitting is shown in figure 8. The bearing in figure 8 was a tapered roller bearing tested in the authors’ laboratory with a high degree of roller end-rib sliding. The sliding resulted in larger traction forces that reduced the roller’s rotational speed, making it the slower component at the roller–raceway contacts. That, along with a small lubricant film thickness relative to the surface texture, caused the microcracks.

Considering a typical main shaft spherical roller bearing such as that shown schematically in figure 9, analyses of the typical radial and thrust forces in the application with the radial clearances inherent in the bearing design indicate that the entire load is supported by the downwind row of the bearing and the upwind row is essentially unloaded, as shown by Ionescu & Pontius (2009). This results in higher loads on the downwind row, as well as a full 360° loaded arc of rollers. The full-loaded arc of rollers increases the number of stress cycles occurring on
a point on the inner raceway for every shaft revolution. The main shaft typically rotates at slow speeds in the 25–35 r.p.m. range, which, even with higher viscosity lubricants, does not generate significant lubricant films. This results in higher loads, more stress cycles and thinner lubricant film thickness on the downwind row increasing the risk of micropitting, especially if sliding is present.

Rollers will slide on the downwind row raceway in main shaft spherical roller bearings owing to an effect called Heathcoat slip. Heathcoat slip is a geometrical constraint suffered by all spherical roller bearings. As illustrated in figure 10, if surface velocities between the inner ring and the rollers match at locations 1 and 3, then the surface velocities must differ at location 2, which means that there is sliding between the roller and the centre of the raceway. Specifically, the inner raceway will have a slower velocity, which was shown by Webster & Norbart (1995) as the component that micropits. This makes the risk of micropitting extremely high for the downwind row.

All spherical roller bearings experience Heathcoat slip, but all spherical roller bearings do not exhibit micropitting. For micropitting to occur on spherical roller bearings, the lubricant film thickness must be insufficient to separate the residual finishing texture, or asperities, on the roller and raceway. That is, the lambda value (ratio of the lubricant film thickness to the composite surface roughness) must be less than 1. Since the thickness of the lubricant film is a function of the entrainment velocity, low lambda conditions commonly occur at slow bearing rotation speeds. Therefore, the micropitting experienced by main shaft spherical roller bearings, typically rotating around 25–35 r.p.m., is due to highly loaded roller/raceway sliding in low lambda conditions.

However, without highly trained tribologists familiar with this type of wear engaged in many turbine design and maintenance facilities, the response by some in the industry is to retrofit their problematic main shaft spherical roller bearings with larger, higher fatigue load-rated bearings. For example, a common main shaft spherical roller bearing for 1.5–2 MW wind turbines is the 230/600 series. These bearings are being replaced in the field with the 240/600 series bearings, which have longer rollers that spread out the load on the raceway and therefore have increased fatigue lives. These bearings are compared schematically in figure 11.
But, as discussed above, the life-limiting problem that main shaft bearings suffer is not related to classic surface or inclusion fatigue, but micropitting. Will the 240/600 series bearings be less prone to the micropitting problems that face the 230/600 bearings? Probably not, since greater roller/raceway sliding will occur in the 240/600 bearings than in the 230/600 series. Although the 240/600 bearing will have a greater dynamic load rating, analysis shows that the upwind row will still be unloaded and the downwind row will again support the entire load. Further, because the rollers are longer, the 240/600 rollers will experience even greater Heathcoat slip than the 230/600 rollers. Therefore, the cyclic shear stresses which lead to raceway micropitting will be even greater.

Fortunately, there are design solutions to the micropitting wear problem of main shaft roller bearings. Elimination of roller/raceway sliding by using preloaded tapered roller bearings would greatly reduce the possibility of micropitting. But for those applications that absolutely require a spherical roller bearing to allow shaft misalignment, engineering the surfaces of the rolling elements to provide reduced asperity contact and a continued barrier to the wear responsible for micropitting should be employed.

(b) Surface engineering

Surface engineering is the practice of altering the chemical and/or topographical properties of the surface of a component or device. An engineered surface that has been shown by Doll & Osborn (2001) and Doll et al. (2004) to work extremely well at reducing and eliminating wear in rolling element bearings is an amorphous carbon coating applied to superfinished rolling elements.

A superfinishing process described by Hashimoto et al. (2009) can produce surfaces such as that shown in figure 12 in which most of the traditional machining features have been removed. These types of surfaces are especially beneficial to mechanical components operating in boundary layer lubrication since the opportunities for asperity interactions in the contact areas are greatly reduced. First referred to as metal-doped diamond-like carbon by Dimigen & Hubsch (1985), these coatings were later determined to be nanocomposites consisting of metal carbide precipitates in amorphous hydrocarbon matrices according
to Sjostrom et al. (1993). Since the tungsten carbide/amorphous hydrocarbon coatings can usually be deposited at temperatures below the tempering points of engineering steels, these coatings are currently used in many mechanical applications as per Holmberg & Matthews (2009).

Generally, WC/aC:H coatings are two to three times harder than steel, use adhesion-enhancing Cr interlayers, are less than 3 μm thick and have low friction coefficients when sliding against steel. The durability of these coatings greatly depends upon their processing conditions and their microstructures. For example, whereas the WC/aC:H coating microstructure shown in figure 13a exhibits very high durability and no measurable wear in highly stressed rolling contact, the WC/aC:H coating microstructure shown in figure 13b does not.

Although both coatings have identical chemistries, details of their processing conditions produced the different microstructures. The coating shown in figure 13b does not delaminate from the steel rollers, but wears through microscopic fracture of the columnar microstructure in a process similar to that modelled by Kang et al. (2008).
A durable WC/aC:H coating and a superfinished surface on the main shaft bearing rollers can eliminate or significantly delay the onset of micropitting by dramatically reducing the cyclic shear stresses associated with moderate $Pv$ roller/raceway sliding. One specific MC/aC:H coating used in this test was designed to be harder and physically polish or wear the mating surface. This allows the coating to continuously polish and repair the mating surface during operation, which keeps the lambda ratio low throughout the life cycle of the bearing.

Validation testing of this concept was performed at the authors’ laboratory. In these tests, rollers were first isotropically finished, then the MC/aC:H coating was applied. Raceways were left in their normally finished state. After 44 million shaft revolutions at a radial load of 150 per cent of the fatigue load rating based on 90 million revolutions, the surface texture of the raceways was dramatically changed with coated rollers. The wear on the raceways operating against traditionally finished steel rollers was minimal. In both tests, a non-additized mineral oil with viscosity aimed at obtaining a lambda ratio of 0.2 was used. Surface textures obtained using white light interferometry are shown in figure 14.

The engineered surfaces’ ability to mitigate micropitting risks on a spherical roller bearing in a wind turbine main shaft position was demonstrated in a subscale test on spherical roller bearings aimed at inducing surface degradation. Again, in this test only the rollers were treated with the engineered surface combination as the polishing effect will modify the texture on the mating surface.

For this test, 22 216 series spherical roller bearing raceways were deliberately damaged with steel debris prior to testing to stimulate early surface damage which generated the fatigue spalls shown in the images in figure 15. The debris damage is very evident on the spherical roller bearing raceway that ran against untreated steel rollers in figure 15a, as is the wear track associated with the Heathcoat slip in the centre of the contact. Conversely, figure 15b shows a debris-damaged raceway that ran against the engineered roller surfaces, and demonstrates that the rollers did repair the debris damage and eliminated the wear track from the Heathcoat slip.

7. Gearboxes

Most wind turbines are designed to run at slow rotor speeds; hence, gearboxes are employed to transfer the torque from a slowly rotating input shaft (25–35 r.p.m.) to a high-speed output (1500–1800 r.p.m.) suitable for the AC generator.

A schematic of a wind turbine gearbox from Musial et al. (2007) is shown in figure 16. The low-speed stage of the gearbox is a planetary configuration with either spur or helical gears. The sun gear drives a parallel intermediate shaft that in turn drives a high-speed stage. The intermediate- and high-speed stages use helical gears. Wind turbines rated below 500 kW tend to use parallel gears, whereas epicyclic gears are favoured in larger wind turbines.

Wind turbine gearboxes operate in conditions that are rather unusual for industrial gearboxes. In order to handle the large amounts of torque, wind turbine gearbox components need to be massive. The heavy components can generate flexing of the gearbox casings and shafts, causing misalignment of the gear mesh and bearings.

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Figure 14. Surface textures from an interferometric white light microscope. (a) Bearing raceway assembled with traditional steel rollers measured pre-test; (b) bearing raceway assembled with traditional steel rollers measured post-test; (c) bearing raceway assembled with smooth, isotropically finished, MC/aC:H-coated rollers measured pre-test; and (d) bearing raceway assembled with smooth, isotropically finished, MC/aC:H-coated rollers measured post-test.
Figure 15. Test results after subscale testing of spherical roller bearings with debris-induced surface damage: (a) inner ring after running with traditional steel rollers, centre of the wear track shows significant damage owing to Heathcoat slip; and (b) inner raceway after running with smooth, isotropically finished, MC/aC:H-coated rollers.

Figure 16. Schematic of wind turbine gearbox with critical bearing positions shown. Adapted from Musial et al. (2007).

As the size of a gearbox increases, so does the opportunity for lubricant debris contamination. Debris can enter gearboxes during manufacturing, be internally generated, ingested through breathers and seals, and inadvertently added during maintenance. Depending upon their location within the gearbox, some of the bearings and gears need to carry large loads at low speeds while others carry lower loads at much higher speeds. Since only one lubricant is used throughout the gearbox, the lubricant film thicknesses will vary greatly from one location to another.

During periods when the wind is not driving the rotor, small-amplitude vibrations can lead to fretting wear of the gearbox components. Wind turbine gearboxes experience rapid accelerations and decelerations when connecting and disconnecting the generators to the power grid, respectively. These events can produce a torsional wind-up of the components and a release of potential energy that generates occasions of high-amplitude torque reversals.

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For example, Robinski & Smurthwaite (2010) have measured that a premature engagement of the generator to the grid forces the intermediate shaft of a 1.5 MW-rated gearbox to accelerate from 375 to 422 r.p.m. in a 2 s period, and a torque change from negative 800 kNm to positive 430 kNm occurs in less than 100 ms. During these events, high-contact stresses are experienced by the non-driven flanks of the gears, which can result in catastrophic tooth fracture.

Torque reversals also generate an almost instantaneous change in the shaft-loading direction. For example, in the generator/grid engagement event described above, the intermediate shaft of the 1.5 MW-rated gearbox was measured to have a displacement greater than 600 μm, far greater than the radial clearance specified for the bearings (120 μm).

Most rolling element bearings have loaded and unloaded zones in operation. In the loaded zones, rollers are well aligned and have high traction forces. On the other hand, the traction forces are much smaller in the unloaded zone and consequently the rollers are not as well aligned with the raceways and have higher slide/roll ratios. An instantaneous change in shaft loading from a torque reversal relocates the load zone of the bearing, and high contact stresses are applied to misaligned and sliding rollers. Because of the variability of wind conditions, torque reversals from generator engagement can happen thousands of times per year.

McVittie (2006) has conducted an analysis of wind turbine gearbox reliability and has reported that gear tooth breakage and fatigue pitting rarely occur, and that, although micropitting still occurs on the gear flanks, better lubrication practices and better surface finishes have significantly reduced the amount of micropitting wear. Fretting wear from non-rotating gearboxes and damage from debris still accounts for some gear failures. In many cases, steel from bearing wear has been identified as the source of debris that damages the gears.

Musial et al. (2007) have made the following observations and conclusions regarding the reliability of wind turbine gearboxes: (i) most of the problems with wind turbine gearboxes are generic in nature and not specific to a single manufacturer or turbine model, (ii) the preponderance of gearbox failures suggests that poor adherence to accepted design practices is not the primary source of failures, (iii) most of the gearbox failures do not originate as gear failures or tooth design deficiencies, (iv) the majority of wind turbine gearbox failures appear to initiate in the bearings, and (v) problems that manifested themselves in early 500 kW to 1000 kW generators exist in the larger >1 MW gearboxes being built with similar architecture.

Further, Musial et al. have identified that there are three critical bearing positions in wind turbine gearboxes that exhibit a high degree of field failures. These are the planet bearings, the intermediate shaft-locating bearings and the high-speed shaft-locating bearings, which are identified in figure 16. Failures are not usually observed on the planet carrier bearings, the hollow shaft bearings and non-locating bearings.

Roller bearings for wind turbine gearbox applications are selected according to their ability to endure and function properly for the entire design life. This duration has universally been established as 20 years.

Bearings have generally been selected according to the dynamic and static load ratings. The dynamic load rating $C_1$ is a measure of the bearing’s ability to withstand rolling contact fatigue, which by typical industry practice is according to the ISO load and life rating standard ISO 281 (2007). The static load rating

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\( C_0 \) is a measure of the bearing’s ability to withstand the maximum applied load without function-reducing permanent deformations or bearing ring destruction according to ISO 76 (2006). The dynamic load rating is used in the ISO 281 (2007) standard life rating equation

\[
L_{nm} = a_1 a_{ISO} \left( \frac{C_1}{P} \right)^{10.3},
\]

(7.1)

where \( a_1 \) is the life modification factor for reliability, \( a_{ISO} \) is the integrated life modification factor accounting for material, lubrication and hard particle contamination, and \( L_{nm} \) is the modified rating life in millions of revolutions. Nominally, roller bearing endurance is calculated as \( L_{10} \), the rolling contact fatigue life in millions of revolutions that 90 per cent of the bearings will survive. Bearing life is based upon the statistical probability of inclusion or surface-originated fatigue; it does not account for life-limiting wear. The prevalent wear modes experienced by the three critical bearing locations of a wind turbine gearbox are micropitting, smearing and flaking from the so-called white-etch areas.

A number of wind turbine gearbox designs initially used spherical roller bearings in the planet positions. A great many of these bearings experienced micropitting that ended their serviceable life well below their predicted fatigue life. Because of the preponderance of micropitting and smearing issues, fewer wind turbine gearboxes are being designed with spherical roller bearings, using cylindrical and tapered roller bearings instead.

The dynamic load ratings \( (C_1) \) of gearbox bearings are very high; however, these bearings spend very little time operating at loads \( (P) \) for which they were designed. When \( C_1/P \) is large, the area of contact between the rolling elements and the raceways is small. Under these conditions, the traction forces between the rollers and raceways which are responsible for rolling motion become small and the slide/roll ratios increase.

Some wind turbine gearboxes employ full complement cylindrical roller bearings in planet positions (location 1 in figure 16) in a desire for greater load capacity. Forces at the roller/roller contact of full complement cylindrical roller bearings oppose the traction forces at the roller/raceway interfaces and encourage higher slide/roll ratios, especially at large \( C_1/P \) values. As discussed above, the torque reversals and shaft movements arising from the accelerations and decelerations associated with the generator/grid engagement can rapidly relocate the load zone of a bearing applying large contact stresses to misaligned and sliding rollers. The magnitude of the \( Pr \) values arising from these transient torque events will depend upon the bearing location in the gearbox and should correlate with whether the bearing experiences no wear, micropitting, smearing or flaking. Since roller/raceway sliding is responsible for micropitting and smearing in wind turbine gearbox bearings, the surface engineering and preloaded tapered roller bearing approaches can mitigate or eliminate these life-limiting wear modes.

The root causes of raceway flaking from white-etch areas are less understood. A cross-sectional SEM image of a wind turbine bearing raceway possessing a white-etch area from C. Bugiel (2010, personal communication) is shown in figure 17a. Cracks propagating from these white-etch areas can cause flaking-type wear of bearing raceways such as that shown in figure 17b. Because hydrogen embrittlement is known to produce similar flaking-type wear of bearings, several
Figure 17. White-etch areas in (a) are ferrite-formed in cracks that nucleate from non-metallic inclusions. When the cracks propagate to the surface, brittle flaking-type wear ensues, as shown in (b).

studies by Iso et al. (2005), Kohara et al. (2006) and Kino & Otani (2002) have suggested that the hydrogen diffusing from the lubricant into the steel might also be the root cause of flaking of wind turbine gearbox bearings. A catalytic interaction with Fe is the proposed mechanism that dissociates hydrogen atoms from water and/or organic molecules; however, this type of surface chemistry is only known to occur on transition metal surfaces in ultrahigh vacuum.

Several studies by Hyde (1996), Ochi et al. (1999) and Hiraoka et al. (2006) have observed that high cycle loading applied to highly stressed regions around non-metallic inclusions like Al$_2$O$_3$ in bearing steels are known to produce ‘butterfly’ cracks, and white-etch areas were found to occur on one side of the crack.

White-etch areas have been studied by several groups. Ochi et al. (1999) first identified the white microstructures in through-hardened AISI 52100 and in case-carburized AISI 4118 as being ferrite and noted that in many cases they were accompanied by black-striped microstructures of a pseudo-pearlite phase associated with the precipitation of cementite. Absent in the starting material, these phase transitions were created in a pin/disc tribometer with contact stresses of about 5.8 GPa after more than 100 million revolutions.

In their study of AISI 52100 bearing steel that had endured 3.9 GPa cyclic stresses, Hiraoka et al. (2006) proposed that microcracks form first, which then generate white-etch areas. More recently, Grabulov et al. (2007) performed rolling contact fatigue testing at 2.6 GPa for 130 million revolutions on AISI 52100 articles, then examined the microstructure in the wear tracks using transmission electron microscopy on specimens removed by focused ion beam milling. They summarized that high cyclic stresses can drive a separation of an Al$_2$O$_3$ inclusion with its surrounding steel matrix causing a nucleation of voids, which leads to the formation of cavities.

Under continued strain, these cavities grow and coalesce into a central crack that propagates away from the inclusion along localized shear planes at 45° with respect to the raceway. When the cracks reach the raceway surface, pieces of the steel are removed in flakes. Furthermore, they speculated that the ferrite or white-etch area is the result of a low-temperature recrystallization process, whereby new grains form from the highly deformed steel matrix adjacent to the
crack faces. The low-temperature recrystallization process they proposed is that crystal point defects generated in the martensite during plastic deformation from rolling contact are stabilized in the presence of carbon in a solid solution forming nanocrystalline ferrite. Further study into the feasibility of this low-temperature recrystallization hypothesis is warranted.

The white-etch areas in the studies discussed above were all generated with very high contact stresses. Although wind turbine gearbox bearings spend most of their duty cycle operating at loads well below their dynamic load ratings, they can experience brief intervals of very high contact stresses. For example, a 600 \( \mu \text{m} \) displacement of an intermediate shaft from a 1.5 MW gearbox could produce a transient raceway stress exceeding 3.1 GPa assuming that the stress is absorbed elastically. Since this stress level exceeds the yield strength of AISI 52100, a significant portion of this transient load will be accommodated plastically. It, therefore, seems likely that generator/grid engagements and disengagements could produce the high magnitude shear stresses needed to generate white-etching areas in the vicinity of non-metallic inclusions, which in turn lead to flaking and a reduction in the service life of the bearing.

All the experiments described above that produced white-etch areas in AISI 52100 specimens incorporated some degree of sliding in the tribological contact. It is reasonable to conclude that shear stresses from roller/raceway sliding plays a role in the creation of white-etch areas.

It stands to reason that bearing steels with fewer non-metallic inclusions would have a reduced probability of experiencing crack initiation from periodic instances of overloading than bearing steels with more inclusions.

Second, inhibiting crack propagation by using steels with finer grained microstructures would also be beneficial towards reducing the occurrence of flaking wear. Case-hardened steel has a lower overall carbon content (0.1–0.3 wt% in the case) than through-hardened steels (0.9–1.1 wt%). Although generally more expensive, bearings fabricated from case-hardened steels have much finer microstructures, much smaller carbides and are much more resistant to fracture than bearings made from through-hardened grades. Bearings fabricated from induction-hardened steels with carbon content between 0.4 and 0.8 wt% have been reported by Luyckx et al. (2009) to show little tendency to form micro-cracks, white-etching areas and brittle flaking wear. However, reduced carbon content in bearing steels usually indicates softer raceways and lower predicted \( L_{10} \) lives.

Third, engineering the surfaces of rollers with isotropic finishes and WC/aC:H coatings will reduce the shear stresses associated with roller sliding and diminish the role that this contact may play in the creation of white-etching areas and brittle flaking.

8. Conclusions

In this article, we have tried to provide an overview of the mechanical functions of the systems that comprise a modern wind turbine generator, and to describe some of the tribological challenges that limit the reliability of those systems. We have also tried to summarize some of the solutions that have been applied to address these challenges and improve the reliability of the component systems.

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