Heliyon 10 (2024) e25860

Contents lists available at ScienceDirect

Heliyon

journal homepage: www.cell.com/heliyon

Research article

A surface study of failed planetary wind turbine gearbox Bearings to investigate the causes of the Bearing premature failure issue

Jasim H. AL-Bedhany [a], Tahseen Ali Mankhi [b,*], Stanisław Legutko [b]

[a] University of Misan, Engineering College, Mechanical Engineering Department, Iraq
[b] Poznan University of Technology, Faculty of Mechanical Engineering, Poland

ARTICLE INFO

Keywords:
Surface investigation
Planetary bearings
Wind turbine gearbox
Premature failure
Flaking
Indentation
Bearing overloading

ABSTRACT

The advanced software used in designing Wind Turbine Gearboxes (WTGs) does not solve the premature failure of the gearbox bearings, which still fail within 5%–20% of their design life by flaking. This issue increases the maintenance, downtime, and early replacement costs that limit the investment in the wind energy field. The majority of the previous research have focused on bearing subsurface investigation and the microstructural changes associated with the failure patterns. Conversely, surface investigation can elucidate significant information about the loading levels and contributors to the premature bearing failure. In this study, two bearings from a planetary stage of a failed multi-megawatt wind turbine gearbox underwent surface investigations and analyses. The analyses include indentations, hardness, roughness, and severe damage regions. The study shows that the contact loading stress exceeds the recommended and more than the compressive yield stresses of the bearing materials. The loading distribution on the bearing inner race during the gearbox operation is quite different from the theoretical loading. The transient loadings throughout the service reduce the Wind Turbine Gearbox Bearings (WTGBs) service life. Furthermore, the significant effects of skewing and slipping have been confirmed. Accordingly, the, lubricant filtration system and the design of the planetary stage are recommended to be improved to extend the fatigue life of the wind turbine gearbox bearings.

1. Introduction

Wind Turbine Gearbox Bearings (WTGBs) in different gearbox stages usually suffer from premature failure by Flaking entirely below their design lifespan of 20–25 years. Flaking is a failure pattern of removing the material from the contact surfaces. The bearing premature failure causes unplanned maintenance and early replacement of the gearbox components that considerably increase the cost of wind energy and limit investments in this field. Premature bearing failure is often associated with subsurface microstructural changes (beneath the contact surfaces) [1–4]; however, the loading levels and the mechanism of the bearing damage initiation and propagation are not fully understood [5,6]. In recent years, a considerable number of publications focused on the subsurface investigation of the damage features such as cracks, damage of inclusions, White Etching Bands (WEBs), butterfly wings, and White Etching Cracks (WECs) [7–10]. On the other side, several studies tried to estimate the bearing damage throughout the service and predict the bearings' fatigue lives [6,11,12]. The mechanism of rolling contact fatigue failure is a debatable issue; however, two main hypotheses were usually reported; the first postulated a surface initiation of cracking damage which propagates towards the subsurface and results

* Corresponding author.
E-mail address: tahseen227@gmail.com (T.A. Mankhi).

https://doi.org/10.1016/j.heliyon.2024.e25860
Received 5 April 2023; Received in revised form 17 January 2024; Accepted 4 February 2024
Available online 10 February 2024

in flacking [13–15]. The second one hypothesized a subsurface crack initiation from inclusions, microstructural alterations, and/or voids, then propagating toward the contact surface and causing severe damage [1,5,16]. Some researchers support the opinion that surface and subsurface crack initiation compete to trigger the damage [6,8], and the dominant of one depends on the component design and the operating conditions under cyclic loadings [17,18]. Dhanola and Garg [19] presented a review of the tribological challenges in Wind Turbine Bearings (WTBs), the development of bearing design, the improvement of the surface engineering of the bearing components, and the use of advanced lubrication systems. Furthermore, the monitoring and detecting of the premature bearing failure have been incorporated in this study. Hydrogen embrittlement as an influential chemical factor in damage initiation is also reported as one of the leading causes of fatigue failure triggers [20,21]. The causes of failure and analyses were investigated by focusing on the subsurface microstructural evolution and analyses of cracks [6,16], the plastic damage evolution [22], characterizing the microstructural alterations [23,24] and evaluating the bearing material cleanliness [25]. Krishnan [26] performed a comparative tribological study of the WTGBs to show the effect of lubrication system on the damage. This effect directly reflects on the bearing contact surfaces as surface damage patterns. The causes and mechanisms of damage initiation and propagation in WTGBs is a debatable issue till now days [5], and there is no confirmed theory can sufficiently describe the early damage initiation and propagation. Furthermore, the formation mechanism of the microstructural alterations is not completely understood [27]. The role of these alterations as a cause or a consequence of bearing damage remains a debatable issue [28]. Doll [29] Discussed the coating of bearing contact surfaces as a process to overcome the tribological issues in WTGB. He addressed the effect of black oxide coating on the mechanism of surface pitting which depends on the surface roughness.

Selection of wind turbine bearings is usually performed according to an operating contact stress levels specified in ISO 281 [30] and BS EN 61400–1 [31] which should be in the range of 1.5–1.7 GPa. During the turbine operating, loading stress levels have a considerable variation due to wind gust (instantaneous wind speed variation). Furthermore, the following events throughout the operating such as emergency stop, start-up, braking, the connection/disconnection of the generator, shut down and the on/off of the electrical grid have a considerable effect on the WTGBs lifespan. According to the standards [32,33], these events are annually occurring around 3000 times [5] with about five transient overloading cycles for each event. Loading levels throughout the events are unknown due to depending these levels on the operating conditions at which the event is taking place. The loading levels and slipping ratio (the relative velocity of the bearing contact surfaces), affect the contact surface roughness and hardness if the overloading level exceeds the compressive yield stress of the bearing material. Furthermore, there are a considerable number of factors affect the WTGBs premature failure, which can be classified as follows.

1. Mechanical factors including (residual stress, service loading, stress concentration, heat treatment and the bearing material mechanical properties).
2. Tribological factors (lubricant, operating temperature, contamination of the lubricant and surface roughness).
3. Chemical factors (carbon emigration and hydrogen diffusion in addition to operating temperatures), and
4. Microstructural factors (bearing material grain size, material cleanliness and the inclusions types, sizes, locations and orientations).

Studying the individual and interactive effects of these factors is very complicated issue; thus, further studies are still required to have a better understanding for the premature failure phenomenon. Oila [34] assessed several factors affecting the rolling and sliding fatigue and he specified the contact stress to be the most effective factor followed by slipping. In the planetary stage of the wind turbine gearboxes, the non-rotating planetary pin carries the planet gear, which usually set on two non-rotating inner race bearings. The inner race in this design has a concentrating contact loading in a unique location and this probably raises the possibility of damage initiation at this location. To increase the bearing fatigue life; Haward [35], postulated applying a “MultiLife” mechanism from Ricardo UK Ltd., by changing the location of the maximum contact stress on the planetary inner races using periodic rotations of the planetary bearing inner race. Other researchers specified other types of bearings (Taper Roller Bearings (TRBs)) to be used instead of the cylindrical roller bearings which currently using [36]. When there is a contamination with hard particles (debris) within the lubricant, either from the gearbox components and/or from the bearings, the overloading levels affect the nature of the contact surfaces, especially that causes surface indentations. Furthermore, indentation size and depth greatly depend on the contact loading level and the debris size. WTGB failure by flaking manifests on the inner race contact surface; thus, mapping the surface damage and the distribution of indents can provide a better view of the loading level and loading distribution upon the contact surface, which can probably be the leading cause of premature bearing failure. The effect of surface roughness on fatigue life had been studied by Sheng Le [37] and confirmed by Evan [5]. The previous research focused on the subsurface investigation of damage patterns and their evolution; however, the causes of the subsurface damage are also reflected on the bearing contact surfaces. The previous literatures are very rare in investigating the bearings’ surfaces damage patterns. In this research, a study of the surface evolution of two real planetary bearings inner races were conducted, including surface hardness, indentations and roughness to predict the loading levels and distributions within the bearings loading zones, and estimate the contact loading levels for expecting some causes of the premature bearing failure phenomenon. Several recommendations to elongate the service life of the wind turbine planetary bearings are also concluded in this study.

2. Methodology

Two inner races from multi-megawatt WTGBs are taken from the planetary stage. The inner races were provided by a Wind Turbine (WT) company. The first was from the Up Wind (UW) bearing located in the turbine rotor side, a Single Row Cylindrical Roller (SRCR) bearing. The second race is from the Down Wind (DW) bearing i.e., the bearing, located away from the turbine rotor, which was a Double Row Cylindrical Roller (DRCR) bearing. The investigated inner races are presented with details in Fig. 1.

The surfaces of the races are categorized into three regions according to the damage levels, as follows.

1. Region A consists of the severely damaged areas.
2. Region B has indented areas with a patch of the rolling elements' strike and
3. Region C is the undamaged areas with no roller strike patch or indentations.

The first step is registering the dimensions of the three mentioned regions. Region "A" in both inner races has a maximum length in the circumferential direction of ~23 mm. The axial lengths of these regions were slightly longer than the length of the rolling elements; this is probably due to axial loading, which causes roller skewing and misalignment. In the UW bearing, region A has approximately the same circumferential length along all the axial direction. However, region A has a tapered shape in the DW bearing starting with approximately the same circumferential length as in the UW one of ~23 mm (at the bearing rim side). This severe damage in the circumferential direction was ~3 mm at the axial end of the first roller strike, i.e., away from the bearing rim side. There is no severe damage on the second row of the DRCR bearing (far from the bearing rim). By comparing the two areas of the severely damaged regions, the UW race has a larger area than the DW one. The mentioned analysis supports the previous postulation of carrying the UW bearing ~40% more than the DW bearing [12]. Indentations produced from lubricant contamination with hard particles (debris) can be observed on both sides of the severely damaged regions, i.e., in bearing loading-in and loading-out regions. Suppose the debris source is the damaged bearing only. In this case, indentations should be observed only in front of the roller, i.e., in the bearing loading out region which is located in the direction of rotation. However, observing the indentations on both sides of the severely damaged region confirms the possibility of other sources of debris, such as gears, shafts, and/or bearing houses. It also confirms the required improvement of the lubricant filtration system to release the debris and the need to improve the bearing and housing design.

The second step is analyzing and counting the indents after classifying them according to their size and mapping their distribution. The number of indents was counted by investigating the indentation, after dividing the bearing contact surfaces to a net of (50 mm × 20 mm) in the circumferential and axial directions respectively and counting the indents' number in each net cell for each size. The indents were classified into three groups according to their central axis lengths, as follows.

- The large indentations (L), having a maximum length of 3 mm or more,
- The medium indentation (M), having a maximum length between 1 mm and 3 mm, and
- The small indents (S) are less than 1 mm in length.

The dimensions of the rolling elements are essential for calculating of Hertzian contact stresses. These dimensions were estimated depending on the patch length of the roller strike in the bearing axial direction (due to not providing the bearing rollers by the company). The bearing characteristics are illustrated in Table 1. The estimated values were also performed depending on the closest bearing type manufactured by SKF which has the same inner race dimensions, which is SKF-NCF 3048-CV [38]. The rollers' number and dimensions were checked by drawing the bearings on full-scale using Solid Works software.

In Hertzian contact theory, the bearing material's mechanical properties considerably affecting the results of the contact stresses. These properties relate to the material's microstructure and its heat treatment. The latter is a confidential procedure for bearing manufacturing companies.

Table 2 contains the chemical composition of 100Cr6 steel, usually used as a bearing material for commercial large-scale bearings [39–41]. In comparison, the material's mechanical properties are the modulus of elasticity E = 210 000 MPa, the compressive yield stress σ_y = 2550 MPa, and Poisson's Ratio ν = 0.3 [39,42].

The third step of the analysis is calculating the contact stresses using Hertz contact theory to compare the results with the bearing selection stresses in the standards as will be detailed in the next section.

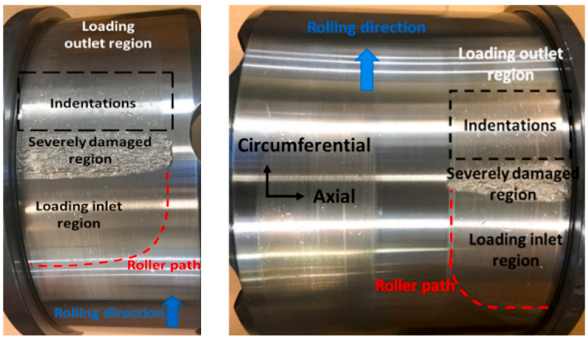

Fig. 1. The investigated inner races of 2 MW wind turbine gearbox bearings, upwind race (on left hand side) and down-wind race (on the right).

Table 1
Measured and estimated specifications of the inner races and rolling elements (in mm).

Bearing type	UW SRCR	DW DRCR
Inner race outer Radius	**145**	**145**
Inner race thickness	**25**	**25**
Axial length	**96**	**210**
No. of rolling elements	**16**	**16 for each row**
Rolling element Dia.	**50**	**50**
Rolling element length	**70**	**70**

Table 2
The bearing material chemical composition (AISI 52100/100Cr6) [39,40].

Matter	Cr	Mn	Si	P	S	Cr	Mg	Fe
% wt	0.98–~1.10	0.25–~0.45	0.15–~0.35	0.025 max	0.025 max	1.30–~1.60	~0.10	Remaining

3. Contact stresses on the planetary races

Predicting the contact stress levels on the planetary bearings' races throughout the service will be very useful to compare with the design values illustrated in the standards. The procedure is based on calculating the torque on the generator shaft by using the power law to calculate the contact force on the planetary bearing, as follows:

$$Power\ of\ generator = Torque \text{ on the generator shaft} * generator\ speed \tag{1}$$

The generator's power is in *Watts*, the torque on the High-Speed Shaft (HSS) is in *Nm*, and the generator rotational speed in *rad/s*. This power can be transferred to the input shaft (the turbine rotor) and then to the planetary stage. Planets number (n), gear ratio (GR), radii of the ring and sun gears (R_r *and* S_r respectively), the number of rollers' rows in each planetary pin (K), and the rollers' number in each row (Z_{rol}) can be used to have the following formula:

$$P_N = \frac{56 * GR * P_{gen}}{K * n * \eta (R_r + S_r)\omega_{gen} * Z_{rol}} \tag{2}$$

Where, the overall drivetrain efficiency η, P_N is the maximum contact force on one roller. More details about the deriving of the above equations can be seen in AL-Bedhany [6]. Fig. 2 demonstrates the contact pressure distribution on the inner race and a roller. Assuming there are three planets ($n=3$), the total rows' number of rolling elements in every planetary stage is $K=3$ (two for the DRCRB and one of the SRCRB), and the UW bearing is overloaded by ~40% more than the DW one [12]. The three forces on the planets (F_P) produce the torque transmitted into the sun gear and then to the intermediate and high-speed stages of the gearbox. The calculated contact stress was found to be ~1.8 GPa. This contact stress is less than the compressive yield stress of the bearing material; however, it is more than the bearing selection stress recommended in the standards [30,31].

The subsurface pressure and stress distributions under the contact region of the roller and the inner race are present in Fig. 3. These stresses were calculated using Hertzian contact theory. Then, the technique of super-position was used to apply the stresses induced due to friction or traction force (the traction in opposite direction of the friction) [43]. The effect of traction force added as a uniformly

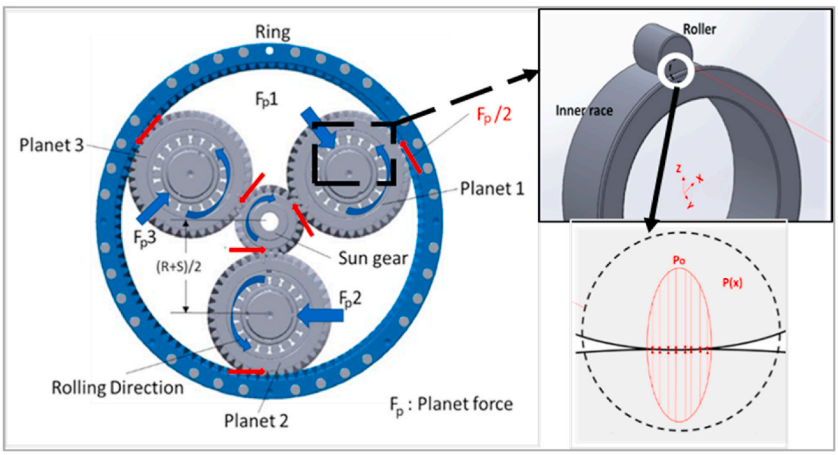

Fig. 2. Reactions on the planetary stage components (left) and Hertzian contact of the inner race and the rolling element (right).

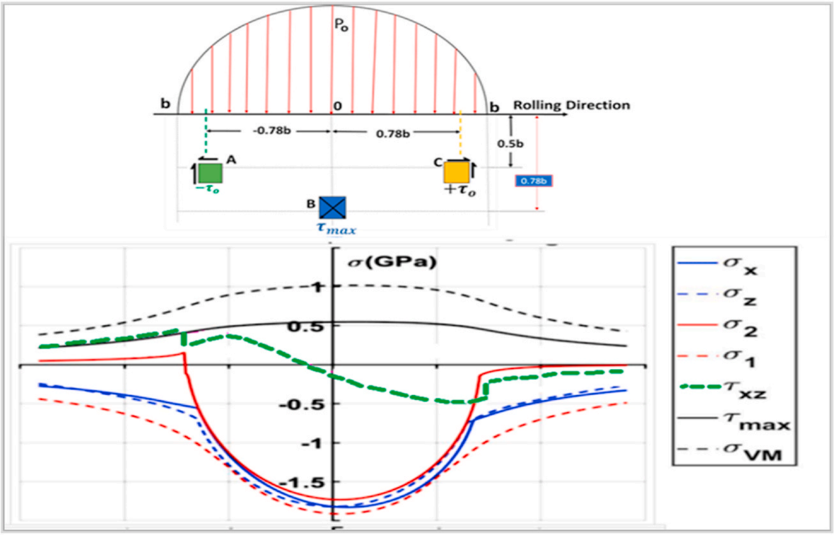

Fig. 3. Distribution of contact pressure and maximum shear locations (Top) and subsurface stresses due to contact force and friction (friction coefficient μ = 0.05) (in the lower).

distributed load on the orthogonal shear stress as can be seen only within the contact region. Applying a traction force in the contact region has two effects: increasing the contact stresses and moving the induced subsurface stresses towards the contact surface. It will be helpful to specify that the sudden variation from positive stress to negative one in σ_2 (in red in Fig. 3) at the entry of the contact region makes this location as the prefferred for trigging surface cracking. However, Von-Mises stresses and maximum unidirectional shear are in the subsurface probably responsible for the initiation of subsurface damage [6,7].

The surface hardness and roughness were also measured within and outside the bearing loading regions as an additional analysis step. The measurements of hardness can show whether the bearing loading level is beyond the elastic limit by investigating the hardening phenomenon within the loading regions. On the other hand, comparing the bearing surface roughness within and outside the loading regions will show the roughness variation throughout the bearing service. The inner races' surface hardness was measured within and out of the maximum contact loading regions to identify the hardness variation due to overloading and evaluate the contact loading levels throughout the turbine operation. Measuring the hardness of the contact surfaces should be conducted away from the indents with at least three times the indent's size, as recommended by Solano-Alvarez et al. [44]. This is to avoid the hardening phenomenon caused by plastic deformation in the affected region. To measure the hardness, the *DuraScan®Struers* digital hardness device was used. The accuracy of the results can be increased by performing seven measurements out of the contact area (within the space between the two rollers' strikes of the DW bearing), i.e., out of the roller strike for the UW bearing then, calculating the mean value. The mean Vickers Hardness (HV) out of the loading contact region was 738 (~61 HRC). This hardness value represents the inner race surface hardness before the service (new bearing); however, the hardness within the contact region represents the hardness after service.

Seven to ten hardness measurements within the contact region were taken in straight lines, starting from the bearing rim in the axial

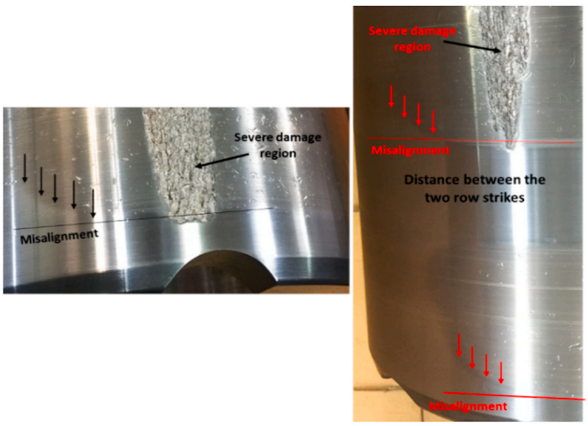

Fig. 4. The surface damage patterns in the investigated races and the confirmation of misalignment.

direction. The measurements were according to the available locations away from the indents. Four locations in the loading-in region of the UW bearing, in addition to five locations in the loading-out region, were chosen to conduct the measurements. This procedure also applies to the DW bearing.

The availability of real wind turbine gearbox bearings is the main limitation of this study since wind turbine companies do not provide the failed parts easily. Furthermore, the companies do not agree to share the details of the results due to their confidential policies.

4. Results

4.1. Surface examination

The overloading of subsurface stresses due to contact forces and pressure (presented in Fig. 3) reflected as surface damage. The patches of the rollers' strikes (seen in Fig. 4 as darker areas) confirm the misalignment at the roller contact edge by having non-straight patches. Accordingly, two issues are confirmed: the misalignment of the rollers' contacts throughout the bearing service and the stress concentration due to skewing at the rollers' edges when the roller enters the loading region. Skewing is the deviation of the rotation centerline of the rolling element from the normal position parallel to bearing centerline to a non-parallel state. The skewing causes overloading at the entry of the rollers to the loading-in region. The effect of skewing appears clearly on the roller patch strikes at the loading-in regions in both the investigated races, as can be seen in Fig. 5. Despite the modern techniques used in the bearing' design, the misalignment and skewing issues are still confirming.

Measuring the surface hardness at the start of the roller strikes showed a hardening phenomenon, i.e., hardness increases within the loading regions and confirms a sudden overloading at the entry of the rollers to the loading-in region. More studies are still required to specify the line in which the hardness to predict the skewing angle and try to avoid this issue by improving the design of the rolling elements and/or the bearing design. This issue can be avoided by using Taper Roller Bearings (TRB) at the planetary stage, and this supports the selection of TRB for the planetary stage of the wind turbine gearboxes [45].

5. Surface indentations

During the bearing manufacturing process, the inner race contact surfaces and the rolling elements should be carefully treated to reduce the friction force throughout the operation. For that, coating the bearing components with black oxide was one of the choices of some manufacturing companies. This oxide layer also prevents the bearing material from contacting with the lubricant and their possible chemical reactions [46]. Debris in the lubricant cannot be avoided despite using advanced filtration systems. Contamination of the lubricant with hard particles (debris) produces indents on the contact surfaces due to high contact loading. These indentations distorted the contact surfaces due to pressing the debris on them to produce the indentations. These indents and contact surface distortions introduce high-loading locations that may be trigger premature bearing failure. Comparing the coated and uncoated bearing surfaces of the inner races shown in Fig. 6 supports this conclusion. It also refutes the possibility of introducing these indentations from the subsurface regions since the indentations on the coated race have a black oxide on the lower indents' surfaces. No specific trend of indents' orientation was observed, i.e., the direction of indentation maximum axis length relative to the rolling direction was irregular. Despite the high number of bearing rotating cycles and the extreme contact stresses around the indentations, no subsurface crack initiation was observed within and around the indented regions (for more information see AL-Bedhany [6]). The border of the indentations, which underwent high contact stresses due to the material rising (produced from the surface deformation),

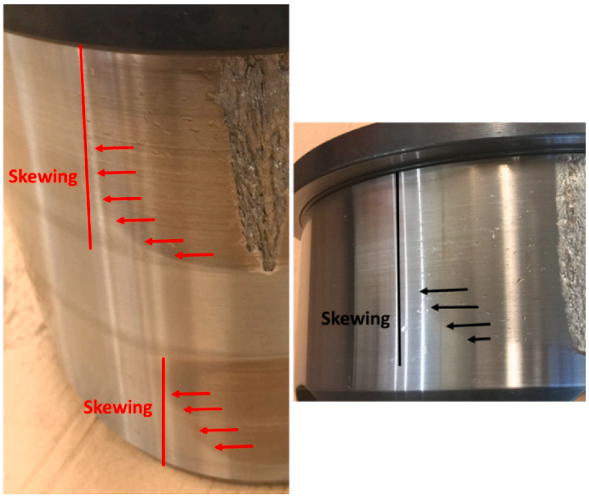

Fig. 5. The skewing effect on the rolling element strikes. For the DW bearing (left) and for the UW bearing (on the right-hand side).

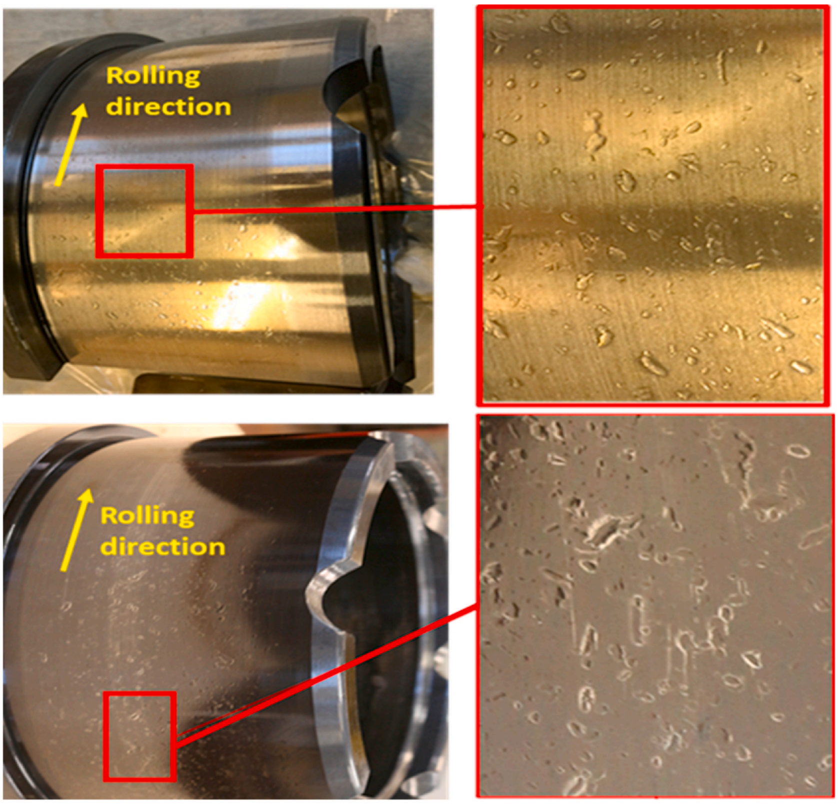

Fig. 6. Indentations on the inner races, uncoated bearing (upper) and coated with black oxide (lower).

appeared lighter in the coated inner race.

The permanent deformation of the indented regions and the rising up of the indentation circumferences make these rising-up areas carry most of the contact and friction loads after removing the debris throughout the rotational movement. It makes the indentation borders subject to very high shear and contact stresses. These extreme stresses on the indents can be confirmed by their white borders on the coated race (due to removing the black oxide from these borders). Fig. 7 demonstrates the procedure of forming the observed features of indents; however, applying Hertz contact theory on a small piece of debris (3 mm in diameter) on the inner race to introduce a permanent indent results in a compressive contact stress quite higher than the material compressive yield stress. Indentation size and depth depend on the contact pressure more than the debris size; therefore, the load distribution can be represented by the distribution of the indents.

For regions labeled by Bs, it has been noticed that indentation frequency (the number of indentations) is lower in the loading-in side than on the loading-out. It may be due to the considerable role of the debris formation from the other gearbox components and the extreme contact stress levels in the loading-out region. The indentations' maximum lengths have no specific orientation trend in both directions (circumferential and axial); however, the indentation frequency in the circumferential direction has a noticeable increasing

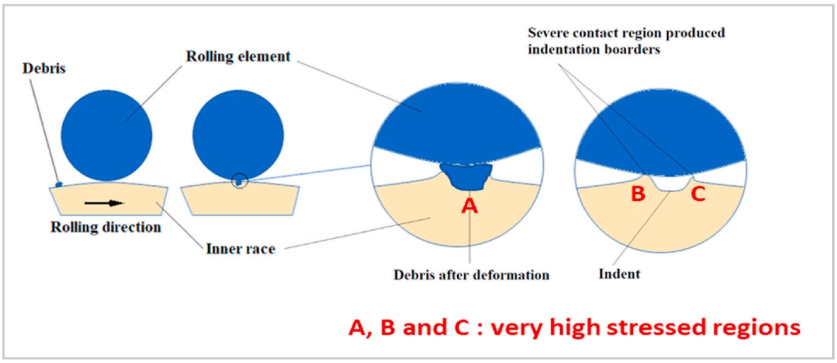

Fig. 7. The procedure of indentation initiation and the severe loading regions.

trend towards region A (the severely damaged region), as can be seen in Fig. 8.

The indentation frequency in the loading-out region the possibility of introducing the indentations from the severely damaged bearing locations. In contrast, the indentations were also observed on the DW bearing, not only in front of the severely damaged region (towards the loading-out region), but also in the second row of the rolling element strike, where no severe damage was observed (see Fig. 9). It supports the possibility of introducing debris from other gearbox components.

In the DW bearing, presenting the indentations of the investigated two roller' strikes confirms the loading distribution on the inner race predicted from the UW bearing, as seen in Fig. 10. However, it is expected that there is less contact loading on the second roller strike (where there is no severe damage). It is probably due to misalignment and bending on the planetary pin and the low loading levels on the DW bearing, as confirmed by Lacava et al. [12]. Furthermore, observing the indentations on the second roller strike, where there is no severe damage, supports the debris formation from other gear components and the severely damaged area of the inner race is not a unique source of debris. According to Hertzian contact theory, the severely damaged regions are only expected to be on the inner race because the contact stress on the outer race is less than on the inner one.

Fig. 11 shows the loading pressure distribution depending on the indentations distribution, which significantly differs from the ideal bearing loading zones presented by Harris and Motzala [47]. When the contact loading level is less than the bearing material compressive yield stress, no noticeable damage (a patch of a roller strike or indentations) can be observed; thus, the region with no patch and indentation can be considered an unloading region. The presented loading distribution is approximately similar to that of Keller-et al. [48] using advanced techniques. This finding lead to predicts that the calculated Hertzian contact stresses throughout the design stage are pretty below the real stresses introduced throughout the service. For that, the theory of non-Hertzian contact is probably more efficient for describing the problem of bearing rolling contact in WTGs, especially with the considerable friction introduced from the debris' entry between the contact surfaces and the fluctuation of load throughout the turbine operating events which causes further slipping [49].

6. Surface hardness

Fig. 12 presents the locations where the hardness was measured on each bearing's race. This figure also demonstrates the fretting damage on the inner surface of the inner race (the contact surface of the inner race with the planetary pin). Despite using an interference fit for these components, the contact load was not uniform, and the fretting distribution (shown in the dotted oval) may require more investigation. Because of the inner race curvature in the race circumferential direction, the investigated races should be rotated after completing each line of the hardness measurements.

The hardness measurements of the two inner races are presented in Fig. 13. The hardness variation in the circumferential direction is beneficial because it reveals the loading distribution within the loading-in and loading-out regions and the nature of the rollers' entry. Before the use, the mean Vickers Hardness (HV) was 738 (~61 HRC), and there are apparent hardness differences between the hardness within the loading and unloading regions. This confirms the hardening phenomenon within the loading regions (loading-in and loading-out regions). For that, the loading level on the planetary bearing inner race was above the stress recommended in BSI 61400–1 [31] and ISO 281 [30] and also more than the compressive yield stress of the bearing material.

In both the investigated bearings, the trend of increasing the hardness is toward the severely damaged regions at the loading-in locations; however, in the loading-out regions, the hardness gradually decreases with being away from the severely damaged regions. It means the maximum contact stress is concentrated at a unique location (severely damaged location) due to a non-rotated planetary pin. It causes a surface hardening, which probably continues till reaching the bearing material's saturated hardening level (material strength), then causes a surface cracking and develops into severe damage. Despite this logical and noticeable

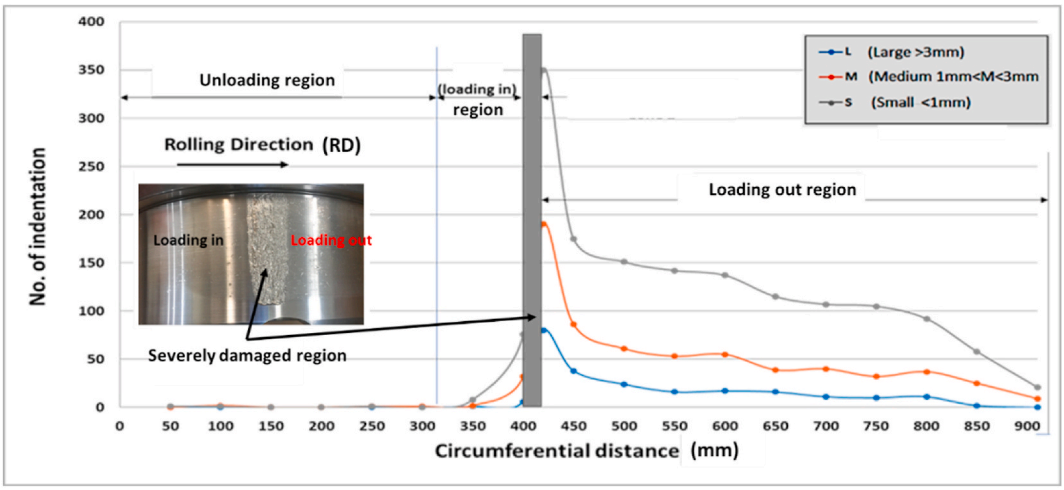

Fig. 8. The distribution of indents on the Up wind bearing inner race.

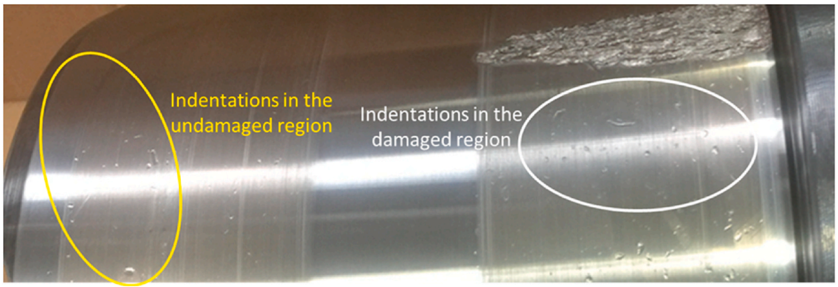

Fig. 9. Surface indents in and out the severely damaged rollers' strikes.

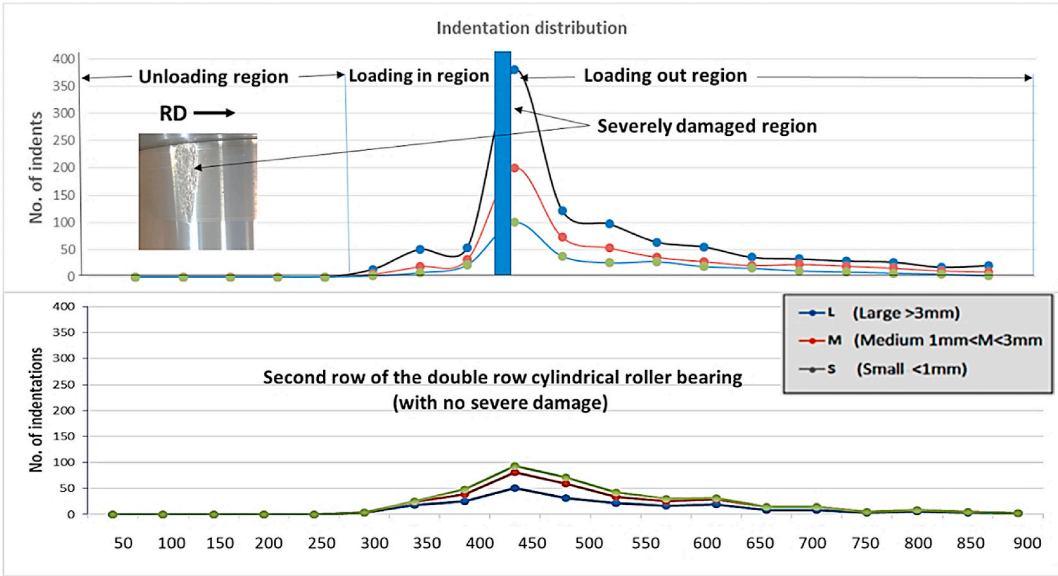

Fig. 10. Indentation distribution on the downwind (DW) bearing, on the first strike with severe damage (top) and on the second strike with no severe damage (down).

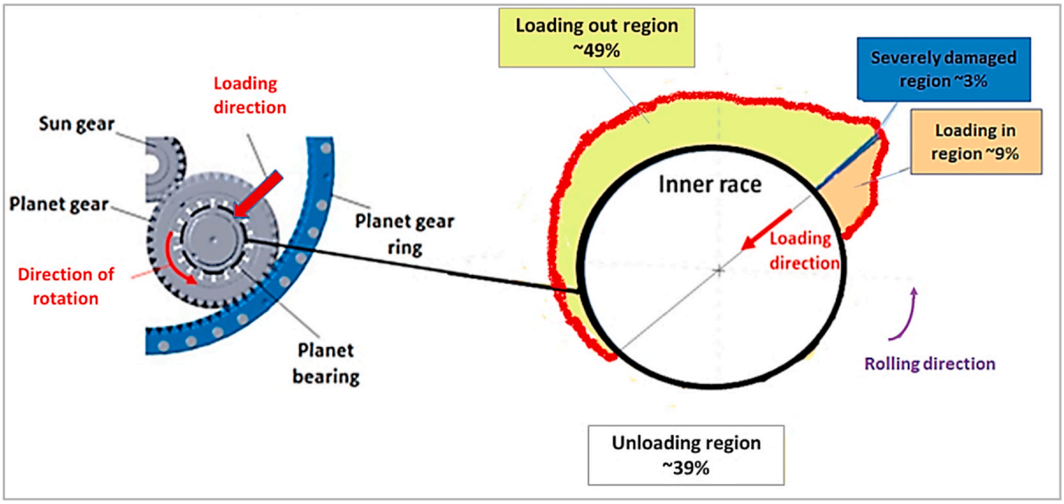

Fig. 11. Pressure distribution predicted from indents distribution on the investigated inner race.

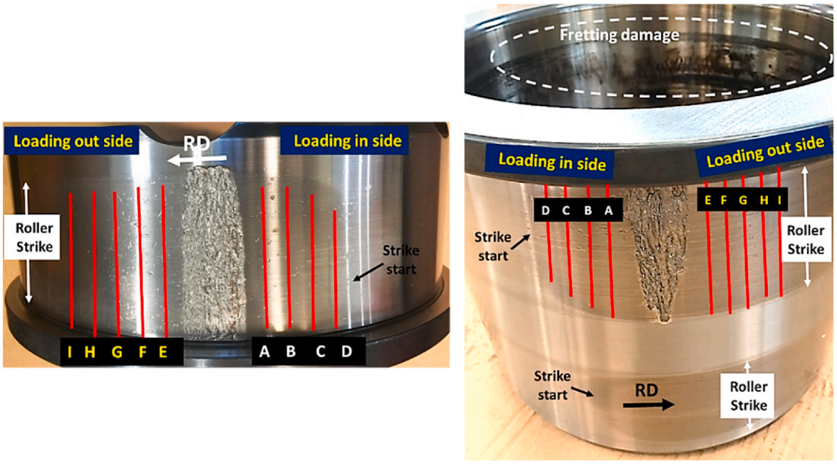

Fig. 12. Locations of hardness measurments on the investigated bearings.

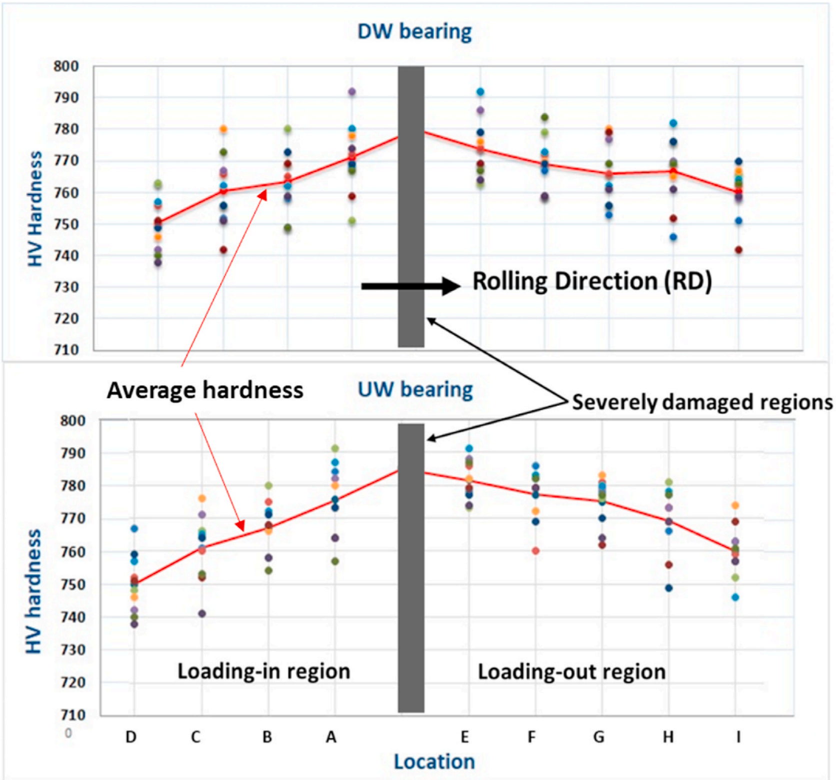

Fig. 13. Surface hardness measurements of the investigated inner races.

prediction, the possibility of subsurface cracks causing a premature failure cannot be terminated.

The trends of indentations and hardness confirm a sudden loading increase within the loading-in regions; however, in the loading-out regions, there is a gradual decrease in the contact stress levels (when going away from the severely damaged region A). The sudden loading increases in the loading-in region probably causing a transient and impact loading between the rollers and the bearing inner race and raising the possibility of damaging this location. Haward [35] tested a novel mechanism with a periodic rotation of the inner race of a planetary WTGB to increase the bearing fatigue life by a considerable percentage (five-fold enhancement). This study also confirms this finding and points out that the planetary stage's non-rotated pin and the contact loading's unique location represent the preferable location of premature bearing damage initiation.

7. Surface roughness

The aim of studying the surface roughness is to confirm the effect of rolling and sliding by comparing the rollers' strikes with the non-contact surfaces. The contact asperities deformed on the micro-level and were affected by the loading level and slipping of the bearing components. It causes a noticeable change in the surface roughness. Therefore, the surface roughness inside and outside the loading regions was measured in different directions (axial and circumferential). Then, the mean roughness was calculated. This had been conducted for the investigated bearings by the first author [6]. Despite the importance of circumferential roughness due to its effect on the rolling direction and the slipping throughout the movements, axial roughness also plays predictable roles due to the extreme contact stress, misalignment, skewing, and bearing axial loading. The arithmetical mean roughness (Ra), which represents the average height deviation of the surface asperities from a mean line, was measured for the investigated surfaces. The mean roughness was calculated for at least five measurements (4 mm length). These five measurements were conducted outside and inside the loading regions of the bearing to increase the measurement accuracy.

MITUTOYO®SJ.400 contact profilometer and non-contact type *Contour GT® Veeco* were used. The former can make a curvature radius compensation, i.e., measuring the roughness on curved surfaces, while the latter has higher accuracy.

For comparison, the average roughness in the region between the rollers' strikes was considered as the inner race roughness before use. In contrast the roughness within the rollers' strikes was considered after-service roughness. The roughness in both circumferential and axial directions was reduced after service from ~0.22 μm to ~0.16 μm and from ~0.17 μm to ~0.1 μm, respectively [6]. The asperities in the circumferential direction underwent more deformation due to the effect of the rolling direction.

8. Discussions

Designing the wind turbine gearboxes includes a single or multiple planetary stage(s). The non-rotating pins in this stage usually have two roller bearings where their inner races are set on the pin. The planetary bearings suffer from premature fatigue failure below their design life of 20–25 years. Studying this issue has two impacts: The first is scientific due to the need to recognize the leading causes and mechanism of failure initiation and propagation. The second impact is economic due to increasing maintenance and energy cost, that reducing investment in this field. Most previous research focused on the subsurface investigation of damage patterns and their evolution; however, the causes of subsurface damage also reflect on the bearing contact surfaces. Different surface damage patterns have been observed on the investigated non-rotating inner races, including roller strike patches, severe damage regions, and indentations. Hardness due to overloading had been investigated by measuring it along the axial and circumferential directions, and this can confirm the occurrence of hardening phenomena, which only occur when the contact overloading stress is beyond the bearing material yield stress. The hardness variation within the overloading regions can point out the loading distribution in these regions.

Measuring surface roughness also provides valuable information about the slipping, which is postulated as one of the leading causes of the bearing's failure. The shape of the roller's strike patch confirms the skewing and misalignment throughout the bearing rotation; these regions underwent a careful investigation. The indentation distribution depends on the contact pressure more than the debris' size. Furthermore, the concentrating of the severely damaged region on a unique location and the increasing indentation frequency towards this location, in addition to the trend of hardness variation to this location, confirm the non-rotating inner race failure due to concentrating the load in this unique location and pointing the weak design of this component (as non-rotating). Several conclusions and recommendations are postulated to elongate the fatigue life of the planetary WTGBs, which will be presented in the next section.

9. Conclusions and key findings

The non-destructive surface investigations of two failed races of the planetary WTGBs have been conducted, and the following conclusions and key findings can be pointed out.

- This study is the first one that depends only on investigating the bearing surface, including surface roughness, hardness, indentation distribution, and other damage patterns, such as the roller's strike patch; these damage patterns reflected from the bearing overloading and the turbine operating conditions. The surface study can provide fruitful information about the causes of premature failure of WTGBs; for example, indentation distribution and surface hardness can be used to estimate the loading distribution and level on the inner race of the bearings by using low-cost investigation.
- The hardening of the contact surfaces within the bearing loading regions confirms that the contact pressure on them exceeds the compression yield stress of the bearing material. Therefore, the selection stress of the planetary WTGBs in the design standards requires re-evaluation depending on the actual service loading.
- The bearing loading distribution differs from the ideal bearing contact condition on which the designer selects the WTGBs—the Taper Roller Bearings (TRB) are probably more suitable for the WTG planetary stage due to less skewing in it.
- The increasing indentation frequency towards the severely damaged regions confirms the high possible role of the indentations in bearing damage by flaking. It occurred due to the extreme contact loading levels of introducing hard particles (debris) between the contact surfaces. Furthermore, improving of lubricant filtration system in WTGs is beneficial to elongate the fatigue bearings' life.
- The skewing and misalignment have been confirmed in both of the investigated bearings. The stresses due to these extra loadings should be considered throughout the bearing selection and gearbox design stages.
- The non-rotating pin in the gearbox planetary stage concentrates the operating and transient loadings at a unique location. It is one of the leading causes of the premature bearing failure in the planetary stage. Improving the planetary stage design, for example, by

making the planetary pins as rotating components and introducing additional pins without planet gears, i.e., to support the two enclosed parts of the planetary gears, can be applied to reduce the misalignment. This will be very useful to elongate the bearings service lives by reducing the bending in the planetary pins.
- Surface roughness affects the contact stresses at the micro-level, and improving the circumferential surface roughness to Ra ~0.1 μm probably increases the bearing fatigue life by reducing the deformation of the contact surface asperities. The contact profilometer gives approximately the same results as the non-contact one. The first type is recommended due to its low cost and its facilities of curve compensation for measuring the roughness on curved surfaces.

10. Recommendations

The following recommendations for future works and investigation can be drawn to overcome the premature failure issue in WTGBs.

1) Investigating the depth of the indentations using different debris sizes and compression loading to confirm the actual overloading levels within the service of the WTGBs.
2) Improving the surface roughness of the bearing components and studying its effect on the bearing service life.
3) Improving the design of the gearbox planetary stage and selecting a more efficient bearing type to elongate the bearing service life.

Data availability statement

Data are available with the corresponding author. It can be provided after a request.

Additional information

No additional information is available for this paper.

CRediT authorship contribution statement

Jasim H. AL-Bedhany: Writing – original draft, Software, Resources, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Tahseen Ali Mankhi:** Writing – review & editing, Validation, Investigation, Formal analysis. **Stanisław Legutko:** Writing – review & editing, Validation, Supervision, Methodology.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgment

The authors would like to acknowledge the provider of the investigated bearings. The authors gratefully thank the Iraqi Ministry of Higher Education and Scientific Research (MOHESR), the University of Misan – Iraq, and Poznan University of Technology - Poland for their support throughout the completion of this study.

References

[1] M.H. Evans, A.D. Richardson, L. Wang, R.J.K. Wood, W.B. Anderson, Confirming subsurface initiation at non-metallic inclusions as one mechanism for white etching crack (WEC) formation, Tribol. Int. 75 (2014) 87–97, https://doi.org/10.1016/j.triboint.2014.03.012.
[2] Y. Murakami, Effects of small defects and nonmetallic inclusions on the fatigue strength of metals, JMSE Int. J. 32 (2) (1989) 167–180.
[3] K. Stadler, A. Stubenrauch, Premature Bearing Failures in Wind Gearboxes and White Etching Cracks, October, 2014, pp. 34–40.
[4] H. Peng, H. Zhang, Y. Fan, L. Shangguan, Y. Yang, A Review of Research on Wind Turbine Bearings ' Failure Analysis and Fault Diagnosis, 2023.
[5] M.H. Evans, An updated review: white etching cracks (WECs) and axial cracks in wind turbine gearbox bearings, Mater. Sci. Technol. 32 (11) (2016) 1133–1169, https://doi.org/10.1080/02670836.2015.1133022.
[6] J.H.I. AL-Bedhany, Effect of Compression, Impact and Slipping on Rolling Contact Fatigue and Subsurface Microstructural Damage, The University of Sheffield, 2020 [Online]. Available: http://etheses.whiterose.ac.uk/27474/.
[7] J. Al-bedhany, H. Long, Microscopic Investigation of Subsurface Initiated Damage of Wind Turbine Gearbox Bearings Microscopic Investigation of Subsurface Initiated Damage of Wind Turbine Gearbox Bearings, IOP, 2018.
[8] L.M. Keer, M.D. Bryant, G.K. Haritos, Subsurface and surface cracking due to hertzian contact, J. Tribol. 104 (3) (1982) 347–351, https://doi.org/10.1115/1.3253217.
[9] A. Grabulov, R. Petrov, H.W. Zandbergen, EBSD investigation of the crack initiation and TEM/FIB analyses of the microstructural changes around the cracks formed under Rolling Contact Fatigue (RCF), Int. J. Fatig. 32 (3) (2010) 576–583, https://doi.org/10.1016/j.ijfatigue.2009.07.002.
[10] T.A. Mankhi, J.H. AL-Bedhany, S. Legutko, Investigation of subsurface microcracks causing premature failure in wind turbine gearbox bearings, Results Eng 16 (August) (2022), https://doi.org/10.1016/j.rineng.2022.100667.
[11] G. Fajdiga, M. Sraml, Fatigue crack initiation and propagation under cyclic contact loading, Eng. Fract. Mech. 76 (9) (2009) 1320–1335, https://doi.org/10.1016/j.engfracmech.2009.02.005.
[12] W. LaCava, Y. Xing, C. Marks, Y. Guo, T. Moan, Three-dimensional bearing load share behaviour in the planetary stage of a wind turbine gearbox, IET Renew. Power Gener. 7 (4) (2013) 359–369, https://doi.org/10.1049/iet-rpg.2012.0274.

[13] P. Rycerz, A. Olver, A. Kadiric, Propagation of surface initiated rolling contact fatigue cracks in bearing steel, Int. J. Fatig. 97 (2017) 29–38, https://doi.org/10.1016/j.ijfatigue.2016.12.004.
[14] Z. Ren, B. Li, Q. Zhou, Rolling contact fatigue crack propagation on contact surface and subsurface in mixed mode I + II + III fracture, Wear 506–507 (2022) 204459, https://doi.org/10.1016/j.wear.2022.204459.
[15] T. Hagemann, H. Ding, E. Radtke, H. Schwarze, Operating Behavior of Sliding Planet Gear Bearings for Wind Turbine Gearbox Applications — Part II : Impact of Structure Deformation, 2021.
[16] J.H. Kang, R.H. Vegter, P.E.J. Rivera-Díaz-del-Castillo, Rolling contact fatigue in martensitic 100Cr6: subsurface hardening and crack formation, Mater. Sci. Eng., A 607 (2014) 328–333, https://doi.org/10.1016/j.msea.2014.03.143.
[17] A. Greco, S. Sheng, J. Keller, A. Erdemir, Material wear and fatigue in wind turbine Systems, Wear 302 (1–2) (2013) 1583–1591, https://doi.org/10.1016/j.wear.2013.01.060.
[18] A.V. Olver, The mechanism of rolling contact fatigue: an update, Proc. Inst. Mech. Eng. Part J J. Eng. Tribol. 219 (5) (2005) 313–330, https://doi.org/10.1243/135065005X9808.
[19] A. Dhanola, H.C. Garg, Tribological challenges and advancements in wind turbine bearings: a review, Eng. Fail. Anal. 118 (May) (2020) 104885, https://doi.org/10.1016/j.engfailanal.2020.104885.
[20] M. Alexander Stopher, P.E.J. Rivera-Diaz-del-Castillo, Hydrogen embrittlement in bearing steels, Mater. Sci. Technol. 32 (11) (2016) 1184–1193, https://doi.org/10.1080/02670836.2016.1156810.
[21] R.H. Vegter, J.T. Slycke, The role of hydrogen on rolling contact fatigue response of rolling element bearings, J. ASTM Int. (JAI) 7 (2) (2010), https://doi.org/10.1520/JAI102543.
[22] A. Bhattacharyya, G. Subhash, N. Arakere, Evolution of subsurface plastic zone due to rolling contact fatigue of M-50 NiL case hardened bearing steel, Int. J. Fatig. 59 (2014) 102–113, https://doi.org/10.1016/j.ijfatigue.2013.09.010.
[23] S.X. Li, Y.S. Su, X.D. Shu, J.J. Chen, Microstructural evolution in bearing steel under rolling contact fatigue, Wear 380 (381) (2017) 146–153, https://doi.org/10.1016/j.wear.2017.03.018.
[24] A. Warhadpande, F. Sadeghi, R.D. Evans, Microstructural alterations in bearing steels under rolling contact fatigue part 1-Historical overview, Tribol. Trans. 56 (3) (2013) 349–358, https://doi.org/10.1080/10402004.2012.754073.
[25] Y. Murakami, Metal Fatigue: Effects of Small Defects and Nonmetallic Inclusions, First., vol. 47, Elsevier, Oxford, UK, 2002, https://doi.org/10.1515/ijmr-1962-530203, 12.
[26] S. Krishnan, A comparative review of the lubrication of bearings in wind turbines and Spacecraft to address wind turbine bearing failures, Tribol. Ind. 45 (2) (2023) 257–271, https://doi.org/10.24874/ti.1452.02.23.05.
[27] M.H. Evans, An updated review: white etching cracks (WECs) and axial cracks in wind turbine gearbox bearings, Mater. Sci. Technol. 32 (11) (2016) 1133–1169, https://doi.org/10.1080/02670836.2015.1133022.
[28] K. Stadler, R.H. Vegter, D. Vaes, White Etching Cracks - a consequence, not a root cause of bearing failure, SKF Evol 1 (2018) 21–29 [Online]. Available: http://evolution.skf.com/us/white-etching-cracks-a-consequence-not-a-root-cause-of-bearing-failure/.
[29] G.L. Doll, Surface engineering in wind turbine tribology, Surf. Coating. Technol. 442 (May) (2022) 128545, https://doi.org/10.1016/j.surfcoat.2022.128545.
[30] I.O. for Standardization, BS ISO 281:2007 Rolling Bearings - Dynamics Load Ratings and Rating Life, vol. 3, May, 2007, p. 58.
[31] IEC, International Standard IEC 61400-1, Wind Turbines Part 1: Design Requirements, 2005.
[32] ANSI/AGMA/AWEA 6006-A03, Standard for Design and Specification of Gearboxes for Wind Turbines, vol. 102, 2003.
[33] Germanischer Lloyd, Guideline for the certification of wind turbines safety system, Protect. Monitor. Dev. (2010) 1–384.
[34] A. Oila, S.J. Bull, Assessment of the factors influencing micropitting in rolling/sliding contacts, Wear 258 (10) (2005) 1510–1524, https://doi.org/10.1016/j.wear.2004.10.012.
[35] Thomas Haward, Development of a Novel Bearing Concept Forimproved Wind Turbine Gearbox Reliability, The university of Sheffield, 2016.
[36] T.A. Mankhi, S. Legutko, J.H. Al-Bedhany, A.A. Muhsen, Selecting the most efficient bearing of wind turbine gearbox using (analytical hierarchy process) method 'aHP, IOP Conf. Ser. Mater. Sci. Eng. 518 (3) (2019), https://doi.org/10.1088/1757-899X/518/3/032050.
[37] S. Li, A computational study on the influence of surface roughness lay directionality on micropitting of lubricated point contacts, J. Tribol. 137 (2) (2015) 1–10, https://doi.org/10.1115/1.4029165.
[38] S. Product, "http://www.skf.com/uk/products/bearings-units-housings/roller-bearings/cylindrical-roller-bearings/single-row-full-complement-cylindrical-roller-bearings/single-row-full-complement/index.html?highlightid=NCF%203048%20CV.
[39] M.-H. Evans, White structure flaking (WSF) in wind turbine gearbox bearings: effects of 'butterflies' and white etching cracks (WECs), Mater. Sci. Technol. 28 (1) (2012).
[40] H. Singh, R.V. Pulikollu, W. Hawkins, G. Smith, Investigation of microstructural alterations in low- and high-speed intermediate-stage wind turbine gearbox bearings, Tribol. Lett. 65 (3) (2017) 1–13, https://doi.org/10.1007/s11249-017-0861-5.
[41] H.K.D.H. Bhadeshia, Steels for bearings, Prog. Mater. Sci. 57 (2) (2012) 268–435, https://doi.org/10.1016/j.pmatsci.2011.06.002.
[42] S. Fujita, S. Matsuoka, Y. Murakami, G. Marquis, Effect of hydrogen on Mode II fatigue crack behavior of tempered bearing steel and microstructural changes, Int. J. Fatig. 32 (6) (2010) 943–951, https://doi.org/10.1016/j.ijfatigue.2009.06.005.
[43] G.W. Stachwowiak, A.W. Batchelor, Engineering Tribology, third ed., Elsevier, Oxford, UK, 2006.
[44] W. Solano-Alvarez, E.J. Pickering, H.K.D.H. Bhadeshia, Degradation of nanostructured bainitic steel under rolling contact fatigue, Mater. Sci. Eng., A 617 (2014) 156–164, https://doi.org/10.1016/j.msea.2014.08.071.
[45] J. Keller, Y. Guo, L. Sethuraman, J. Keller, Y. Guo, L. Sethuraman, "Gearbox Reliability Collaborative Investigation of Gearbox Motion and High-Speed-Shaft Loads," Denver, 2016, https://doi.org/10.13140/RG.2.1.2520.4888.
[46] H. Singh, G. Ramirez, O. Eryilmaz, A. Greco, G. Doll, A. Erdemir, Tribol. Inte. Fatigue Resis Carbon Coat. Roll. Slid. Conta. 98 (2016) 172–178.
[47] T.A. Harris, M.J. Motzala, Rolling Bearing Analysis-Essential Concepts of Bearing Technology, 2007, https://doi.org/10.1201/b15723-9. Boca Raton, USA.
[48] J.A. Keller, B. Gould, J. Keller, B. Gould, A. Greco, Investigation of Bearing Axial Cracking : Benchtop and Full-Scale Test Results Investigation of Bearing Axial Cracking : Benchtop and Full- Scale Test Results, August, 2017.
[49] J. K, P.G.J. Helsen, Y. Guo, Experimental Investigation of Bearing Slip in a Wind Turbine Gearbox during a Transient Grid Loss Event, vol. 19, Wind Energy, 2016, pp. 2255–2269, https://doi.org/10.1002/we.