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Review

Oscillating rolling element bearings: A review of tribotesting and analysis approaches

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ABSTRACT

Rolling element bearings, when subjected to small oscillating movements or vibrations, run the risk of being damaged by mechanisms such as Standstill Marks and False Brinelling. Damages resulting from these phenomena can decrease bearing fatigue life and increase wear-induced friction torque. These failures do not correlate well with standard life estimation approaches. Experimental studies play a crucial role in gaining knowledge in this area. The review integrates knowledge from experiments ranging from single contacts to laboratory and full-scale bearings in wind power and aerospace applications. The generalization is achieved using a non-dimensional amplitude parameter that relates rolling element travel during an oscillation to the Hertzian contact size. The review encompasses testing methods, procedures, reporting practices, result scaling, and application-specific considerations.

1. Introduction

Rolling element bearings that are subjected to an oscillating motion (Fig. 1) are defined as oscillating (rolling) bearings. Oscillating bearings have numerous industrial applications in robotics, avionics, wind energy, construction machinery, equipment used in space exploration, amongst others. Failure modes caused by this operation do not correlate well with various approaches used to calculate standard service life. This leads to major financial losses and hinders the advancement or the implementation of new technologies. In a worst-case scenario, lives may be endangered. In oscillating bearings, damages leading to failure modes are characteristically described as Standstill Marks and False Brinelling. Such damages can cause a tenfold decrease in rolling bearing fatigue life. They can cause wear-induced increases of the friction torque which may also lead to failure. Bearings subject to vibrations under standstill conditions can be considered oscillating bearings as well. In fact, the harmful effects of standstill vibrations first observed and described in the 1930s, brought this phenomenon to the attention of the engineering community. Wheel hub bearings of newly built cars, which were exposed to vibrations during transportation, failed shortly after these cars were put into use. Although oscillations were never intended to be part of the operating conditions of these early wheel hub bearings, the situation for modern-day wind turbines is the exact reverse: rolling bearings have to be designed to be able to operate under oscillating conditions.

Bearings are complicated systems to study. The fact that most bearings are grease-lubricated adds further complexity to the study of what is actually taking place at the contacts between the rolling elements and the raceways. Full-film lubrication is not the dominant lubricating mechanism in most oscillating bearings. Instead, the mixed and boundary lubrication regimes favor surface-induced wear mechanisms and limit the applicability of ISO281 calculations which focuses on rolling contact fatigue. As of today, there is no reliable analytical or numerical model available to predict the occurrence or the progression of wear in oscillating bearings. The main reason is that it is very difficult to model the time and temperature-dependent behavior of a multi-phase material like grease, especially in a transient motion case. Additionally, taking into account chemical interactions, dependent on the lubricant composition, remains out of reach. Therefore, experimental studies will be,

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Nomenclature

x	Ball-raceway travel (peak-to-peak) [mm]
α	Contact angle [degrees]
μ	Coefficient of Friction (COF) [–]
θ	Oscillation amplitude (peak-to-peak)[degrees]
$2a$	Major length of an elliptical contact [mm]
$2b$	Minor width of an elliptical contact [mm]
d_m	Pitch diameter [mm]
D	Rolling element diameter [mm]
F_t	Tangential force [N]
p	Pressure distribution [N/m^2]
q	Traction distribution [N/m^2]
s	Dahl's rest slope [N m/degree]
Ts	Dahl's running torque [N m]
$x/2b$	Amplitude ratio [–]

for the foreseeable future, the primary means of understanding these damage mechanisms. The number of publications of which the main subject is research into oscillating bearings is still relatively small but has been increasing over the decades since the field was first introduced in the 1930s (Fig. 2). It is clear that the field faces challenges within current and future engineering practice. It is therefore regrettable that much of the research has been carried out and described in such ways that complicate the generalization of results. A case in point is the often incomplete descriptions of contact conditions within the bearings in terms of geometry, contact pressures, oscillation amplitude relative to the contact size, surface-finishing and -preparation, and lubricant chemistry. This review discusses the importance of all these contact conditions and aims to arrive at a succinct description of the current state of research in oscillating bearings.

The review begins with a historical perspective, in Section 2. Definitions of terminology and damage mechanisms are presented in Section 3. The description of the standardized testing methods and the ball-raceway contact conditions that occur during these standard tests (Section 4) is followed by the insights gained from an overview of the most relevant fundamental research (Section 5). Section 6 describes the analytical and numerical modeling of oscillating rolling contacts. Specific applications, with emphasis on wind turbines and space exploration, are covered in Section 7. The last Section 8 gives a general summary of the knowledge thus far and presents recommendations and possible avenues for future research. This discussion section draws from a table that summarizes all of the bearing testing research, which is available within the Electronic Supporting Information (ESI).

2. From the 1930s to the 1960s

The first published work on oscillating bearings dates back to 1937. Almen [1] tested thrust bearings in oscillating conditions while varying amplitude, frequency, load, and lubrication (Fig. 3). This research was conducted at General Motors (USA) following the discovery of bearing damages which were apparently caused by the simple act of shipping new vehicles. Automotive bearings are typically designed to operate for years under conditions of variable speed and load, without having to be serviced or replaced. In the case of the GM cars, the damages, apparently caused by shipping, resembled the indentations typically encountered on metals following a Brinell hardness test. In addition, corrosion products were often retrieved from the bearings. These damages occurred under loads much lower than those required to induce plastic deformation on such a scale. Similar phenomena were subsequently found in variable-pitch airplane propeller components, in aircraft engine rocker shaft bearings, and in universal joints. In all these instances, the bearings had not been rotating but had been subjected to vibrations or small pivoting motions. Almen [1] reasoned that the damages had to be caused by wear, rather than by plastic deformation. He based his conclusion on the similarities with fretting. Fretting was first discovered and analyzed in 1911 [2]. It was further investigated in the 1920s and the 1930s when the first notions of its underlying mechanisms emerged [3,4].

Almen, in his seminal paper published in 1937 [1], coined the term False Brinelling. He developed many of the general notions, currently associated with this process, such as the importance of making distinctions between plastic indentations and wear-induced indentations, and the importance of guaranteeing complete ingress of the lubricant into the contact by reducing the viscosity of the lubricating medium. An increased viscosity of the lubricating oil when shipments were made during the winter, explained why the damages were exacerbated in the cold. Almen [1] therefore recommended fully flooded lubrication. To understand the association of corrosion products with False Brinelling, Almen [1] conducted further tests under vacuum conditions in the absence of lubrication. Damage and corrosion through oxidation were substantially attenuated, highlighting the role of oxygen in the wear process.

In the early 1930s, the Fafnir Bearing Company encountered problems with oscillating bearings when its products were introduced for use in the textile industry [5]. At the time, the company mistakenly concluded that the quality of the bearings was to blame for the premature failures when the bearings were subjected to vibrating or oscillating conditions of small amplitudes. In response to these perceived failings, a general cautious practice developed, which was characterized by the use of arbitrarily large safety factors. According to internal sources of the industry, the approach to computing equivalent rotary load-speed for a given rate of oscillation was "haphazard at best". Consensus on lubricant selection for these novel conditions was also speculative. The prevailing philosophy at the time could be formulated as follows: "good full rotary motion greases must also be good when subjected to oscillatory applications" [5]. However, a breakthrough was achieved in 1939 when it was shown that certain grease formulations could make cheap bearings with rough races outlast the highest quality precision bearings when the latter were lubricated with good greases as were specifically formulated for full rotary motion. This triggered the establishment of a long-term testing program within the Fafnir Bearing Company. Early tests indicated that lithium greases based on thin mineral oils were able to extend the life of the oscillating bearings by a factor of 30 when compared to the grease that was used at the time in bearings for aircraft control equipment. The timing of this finding, at the onset of World War II, was crucial in that the increased aircraft production could proceed using more reliable bearings which required less frequent inspections and replacements. This research later cascaded into the development of the Sikorsky R-4 helicopter, which pioneered the design of rotor-head systems and became the first mass-produced helicopter. Initially, although, this rotor-head design was displaying advanced corrosion-riddled damages in the blade bearings within less than 50 h of operation. USAF Air Technical Command reports [6–8] fully document this helicopter development program (cited in [9]).

Further research used glass bearings to visualize the grease lubrication in the contacts [5]. Greases that performed well under oscillating conditions would closely follow the small crevice volume in the vicinity of the contact whereas underperforming greases would quickly result in a starved contact. It was realized that lower consistency greases are favorable and that a critical situation arises when the contact patch between the balls and the race is not fully uncovered during the oscillation, leading to localized wear of the surfaces resulting from the starved conditions [5]. From further research at the Fafnir Bearing Company [10] it was proposed that the lubricant was pressed out of the contact zone by the high Hertzian pressure, forcing the surfaces into metal-to-metal contact with micro-welding occurring between the surface asperities. It was recommended to reduce the calculated lifetime

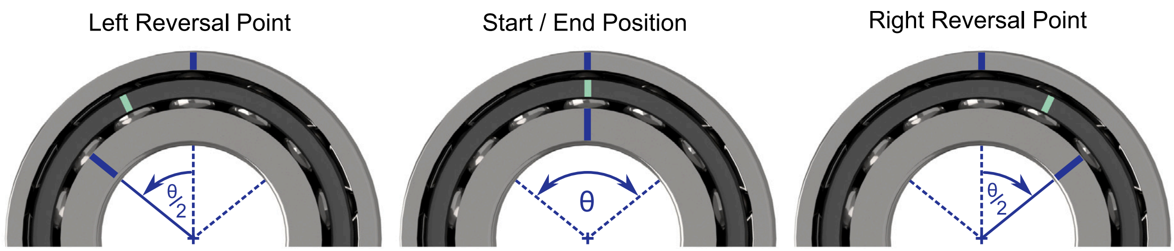

Fig. 1. Diagrammatic representation of an oscillating angular contact bearing: outer ring stationary, inner ring oscillating with oscillating angle θ. In the further course of the paper, the oscillation angle θ denotes the angle enclosed between the reversal points of the oscillation. Note that the relative angle between the bearing rings, which has to be considered for the calculation of the amplitude ratio, is the decisive one.

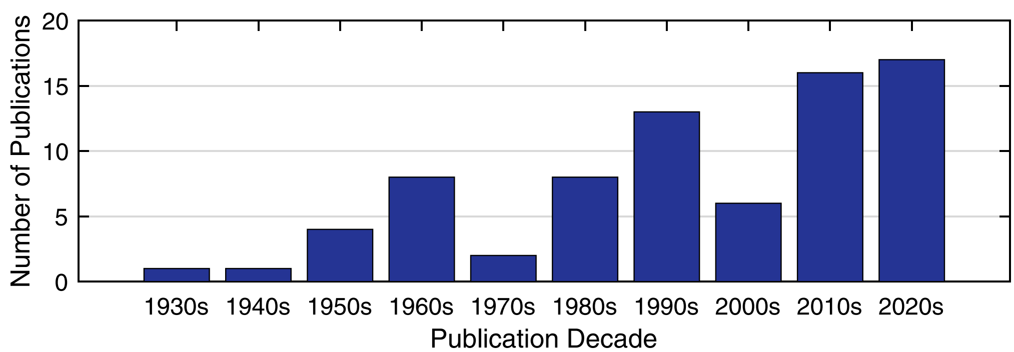

Fig. 2. Oscillating bearing testing publications through the decades.

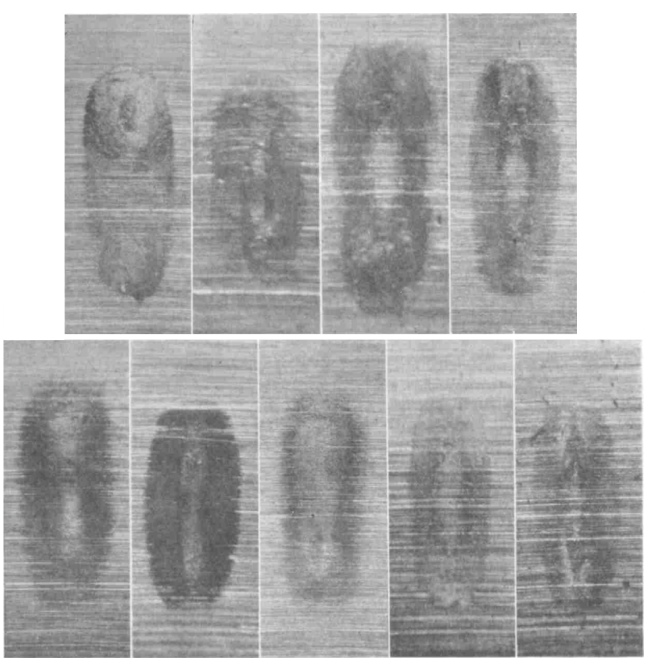

Fig. 3. Original images of raceway damages resulting from the bearing tests as conducted by Almen with his vibration wear-test machine: effects of varying lubrication [1].
Source: Image originally published in Mechanical Engineering magazine, Vol. 59, No. 6, June 1937. Copyright ASME. Reproduced with permission. All rights reserved.

of such bearings by a factor of 0.35 to 0.5 in order to guarantee the safety of the design. It was furthermore recommended that ball bearings should oscillate at least 180° to reliably prevent False Brinelling damage [10]. These conservative recommendations reflected not only the large margins of uncertainty, characteristic for the time but also the scope of possible improvements.

In the 1950s, several avenues for development were being pursued. Finishing the raceway surfaces by vapor blasting could improve bearing longevity when used with military specification greases [11]. However, MoS_2 or graphite when added to greases in excessive amounts could increase wear by up to 30 times [12]. The first two systematic studies [9,13] of grease parameters affecting oscillating bearings concluded that wear was significantly reduced by greases with lower consistencies, higher shear-susceptibilities, and higher bleed rates. The latter is an indication of how easily the oil escapes from the thickener matrix. Comparable results were obtained with softer greases. Several other types of greases, not exclusively Li-based greases, could prevent the generation of corrosion products. The role of additives was demoted because greases with MoS_2, ZnO, gallium, and tricresyl phosphate (TCP) did not improve performance. However, Herbek and Strohecker [9] warned against taking these shear-sensitivity and consistency parameters to extreme values, citing a discrepancy between a grease that showed great promise in laboratory tests but later showed a poor performance in the helicopter rotor application [7], inferring that the shear-susceptible grease broke down into a fluid and was ejected by centrifugal forces.

As aircraft technology continued to advance, operating conditions for bearings in air-frames became more demanding. This led to a research project on oscillating self-aligning roller bearings for high-temperature operation (detailed in [14], summarized in [15]). The objective was to provide air-frame designers and bearing-manufacturing companies with relevant information regarding materials, lubricants, bearing designs, and life expectancies at high temperatures. Periodical grease replenishment was necessary to achieve the desired length of bearing life, regardless of whether an overlap of adjacent ball contacts occurred during the oscillations or not. AISI 52100 bearing steel was found to be satisfactory at temperatures up to 230 °C, whereas tool steel and 440-C stainless steel could perform at up to 316 °C. Failure modes were described as predominantly driven by plastic deformation combined with wear and oxidation. Failure instances could be fitted to a Weibull distribution function. Further works also employed Weibull analysis for needle bearings where the adjacent contact areas do not overlap during oscillation. It was shown that no corrosion products were observed in fully flooded lubrication conditions and the

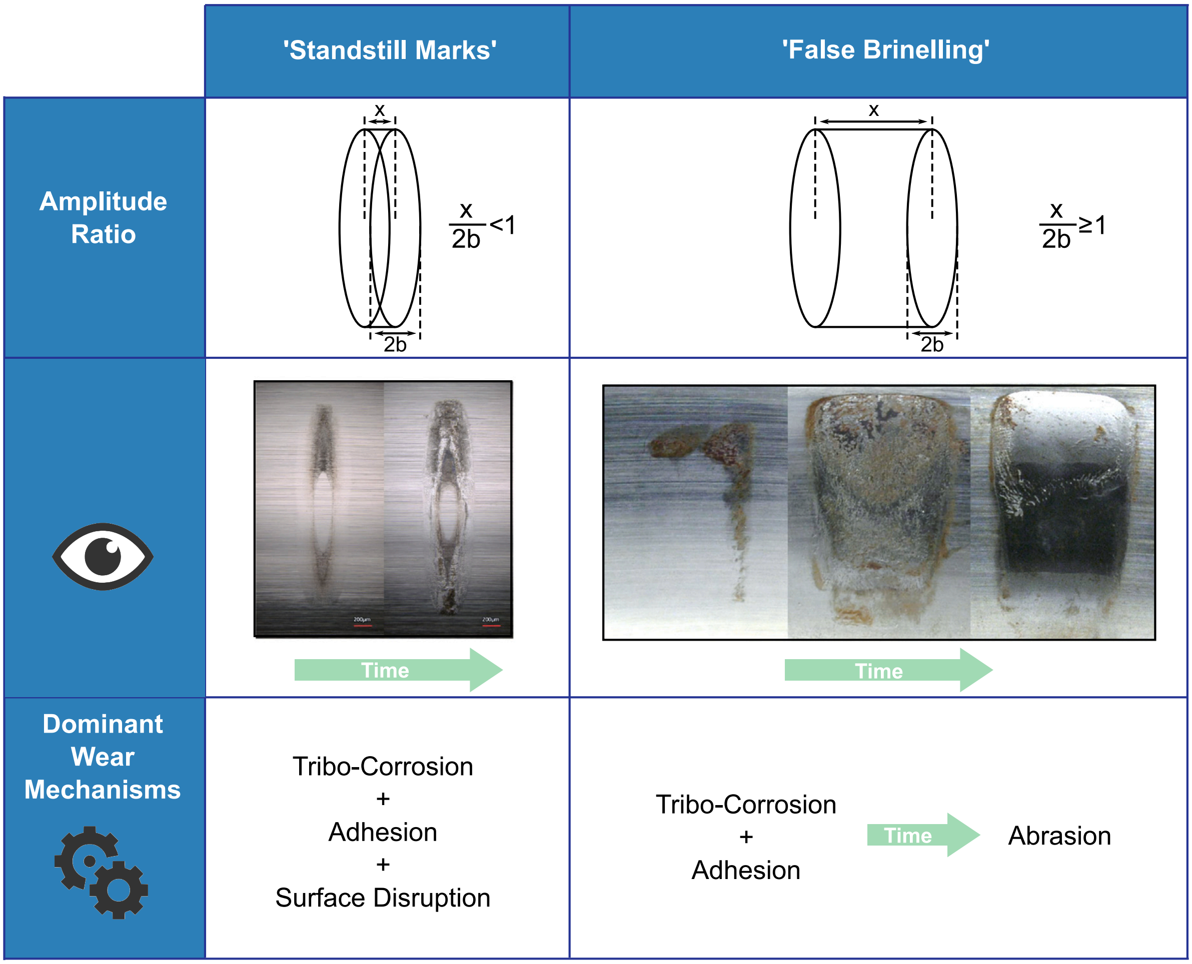

Fig. 4. Diagram to distinguish surface damage based on amplitude ratios. 1st row: suggested damage designation; 2nd row: schematic diagram of the amplitude ratio (schematic does not consider sticking zones); 3rd row: visual perception of increasing wear severity; 4th row: dominant wear mechanisms.

failure mechanism was fatigue, addressed in the statistical analysis [16–18]. This supported Almen's recommendation to provide fully flooded lubrication [1].

Simultaneously, Pittroff, at SKF, investigated the behavior of cylindrical roller bearings when subjected to load oscillations and presented an extensive body of experimental work together with theoretical considerations [19–21]. Pittroff varied the individual contact loads by an off-axis weight rotating on a shaft. The resulting raceway damages were shown to be originating from alternating elastic deformation, where the shearing strength of the asperities was exceeded and rapidly oxidized debris was produced. Despite the fact that the work and its conclusions were presented more than half a century ago and that the chemistry of lubricants has significantly evolved since then, the experimental results and their conclusions are still relevant and most of the original findings have been confirmed by recent studies.

3. Oscillating bearing terminology and damage mechanisms

Distinguishing between different operating conditions of the ball-raceway contacts is of paramount importance, particularly when the aim is to determine the exact surface degradation mechanisms that are taking place. The amplitude ratio or the oscillating angle of the bearing can be used for the classification of different operating mechanisms. This review uses the amplitude ratio $x/2b$. It is a dimensionless amplitude parameter that allows for uniform classification criteria even when differently sized bearings are compared. Attempting to classify and compare bearing operation and function by using only oscillation angles can often be misleading, certainly when bearing dimensions can vary between a few millimeters and several meters. The ball-raceway travel (the distance x that the rolling element covers on the raceways with each cycle) can vary from being well below the minor contact width $2b$ (Fig. 4, left column) to values well above it (Fig. 4, right column). The area of the raceway that comes into contact with the rolling element during the oscillation is called Contact Track in this review. In the second row of Fig. 4 an illustration shows how different amplitude ratios present themselves schematically (for x, see Electronic Supporting Information). The consideration of the damages in this paper is limited to the tribological contact between the rolling element and raceway of the oscillating rolling element bearing. Furthermore, only damage caused by relative motion or repeated stress cycling in the oscillating rolling contact is considered. Damage due to mechanical overload such as True Brinelling is not discussed.

In practice, the damage mechanisms are identified based on the appearance of the wear marks on the bearing raceways. In the literature, the designation of the wear damage is often ambiguous and imprecise as different terms such as False Brinelling, Fretting, or Fretting Corrosion are used interchangeably for the same observed wear features. Some of these terms describe the resulting wear based on visual perception (Fig. 4, row 3) while others refer to the underlying damage mechanisms (Fig. 4, row 4). In the authors' opinion, a clear assignment of the damage terminology is necessary because the wear processes are based on different fundamental mechanisms, for which different corrective measures may be necessary.

The term Fretting is used to describe a damage mechanism occurring predominantly in reciprocating sliding contacts. When occurring in conjunction with tribo-oxidation or tribo-corrosion the term Fretting Corrosion is used. The definition of the term Fretting was first formulated by Tomlinson, who investigated the influence of vibrations on a contact between spherical and flat steel surfaces in 1927 [3]. He also used the term Fretting in studies of conformal fits exposed to vibration. Further important investigations on Fretting with varying oscillation amplitudes were carried out with a ball-on-disc experimental set-up [22]. All these investigations have a common factor: they were carried out under reciprocating sliding conditions. From the point of view of contact mechanics and lubrication, reciprocating sliding conditions are different from the rotation of a ball around an axis perpendicular to the normal of the contact, as is the case in a rolling element bearing (even at small angles). Fretting and Fretting Corrosion share underlying damage mechanisms with an oscillating rolling contact but the latter usually presents other damage mechanisms as well. The relative contribution of each of these damage mechanisms to the overall wear process can vary depending on the operating conditions. For the purpose of clarity, the authors propose to use the term Fretting for oscillating contacts under pure sliding conditions, the term False Brinelling for $x/2b \geq 1$, and the term Standstill Marks for $x/2b < 1$ (the latter two for oscillating rolling contacts as they appear in rolling element bearings). These terms define different types of observed damage marks and point to several underlying damage mechanisms (Fig. 4).

The term False Brinelling (Fig. 4, right) was first used by Almen in 1937 and was originally associated with the formation of depressions in the bearing raceway which can easily be mistaken for True Brinelling marks (usually resulting from mechanical overload and thus from plastic deformation [1]). These trough-shaped damages occur at $x/2b \geq 1$. They can be observed after standard test procedures such as the SNR-FEB2 test, which are frequently used in the European wind industry for screening greases for rotor blade bearings [23]. At $x/2b \geq 1$, the angles are sufficiently large so that no surface element of the raceway is permanently in contact with the rolling element, allowing lubricant replenishment of the contact track. From the point of view of contact mechanics (Section 6) as well as lubrication (Section 5), the tribological conditions, therefore, differ from those operating under $x/2b < 1$.

False Brinelling occurs when a loaded oscillating rolling contact, subjected to slippage, is insufficiently lubricated. A large amount of wear particles is generated due to adhesion and tribo-corrosion, which can further restrict the lubricant replenishment. The wear particles subsequently act as an abrasive medium. Over time, the end result is the formation of typical deep depressions.

For $x/2b < 1$, the perceived raceway damage differs from False Brinelling. It has an undamaged central stick zone (Inner static zone in Fig. 5) and a damaged outer sliding zone (Influence zone, Fig. 5). These typical presentations of damages are called Standstill Marks [24] (Fig. 4, left). The damage starts to appear at both ends of the major elliptical axis, i.e. in the areas subjected to the greatest degree of microslip. Thereafter, the damage development spreads along the minor elliptical axis. The damaged and affected area then widens further due to the thickening and abrasive effect of the wear particles. In the case of severe damage, the wear volume is nearly linearly dependent on the number of cycles but is degressive in the case of moderate damage. The development of the depth of wear over the number of cycles is always degressive [25].

Although the elliptical Standstill Marks caused by small oscillation angles or mere elastic deformation might appear relatively harmless at first sight, there is significant local surface damage caused by several wear mechanisms (Fig. 5). The wear marks at the extremities of the ellipsis show tribochemical reactions and surface disruption. There are no visible changes in the central sticking zone of the contact. Micro cracks occur at the border between the sticking zone and microslip zone due to large tangential forces introduced locally onto the surface (Fig. 6) [26]. This can significantly reduce the service life of the bearing as was proven by tests on an FE8 testing machine with pre-damaged bearings [25].

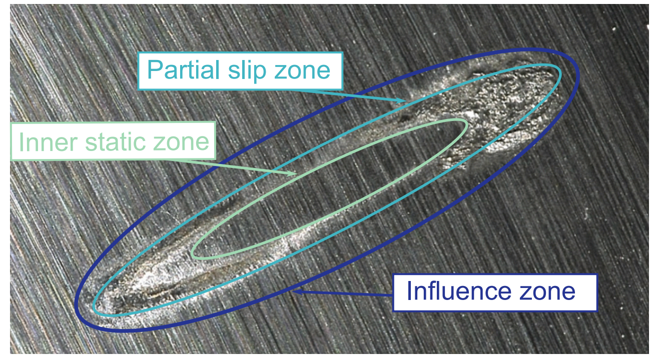

Fig. 5. Different zones of a typical standstill mark [26].

4. Standard testing methods

The ASTM-D4170 [27] (a.k.a. the “Fafnir” test) was the first standardized method used to test and characterize oscillating rolling element bearings. It was introduced in 1982 by the American Society of Testing and Materials, now ASTM International [28]. The only other standard is the NFT60-199 [29], which was introduced in 1994 by the French Standardization Association. It is widely adopted in Europe. These two standards aim to quantify grease performance in terms of wear attenuation. Both tests work with quite large oscillating angles (± 6° in the ASTM-D4170 i.e. $x/2b \approx 5.5$ and ±3° in the SNR-FEB2 (NFT60-199) test or $x/2b \approx 3.4$). As a result, wear symptoms in these tests cannot be directly compared to those wear marks in bearings that are subjected to smaller angles ($x/2b < 1$) or to micro-movements resulting from vibration or elastic deformation.

4.1. Fafnir fretting test (ASTM D4170)

The Fafnir test is conducted in accordance with [27]. Norm-specific test bearings (thrust ball bearings with ground races and 9 balls, similar to type 06 × 65) are subjected to oscillations of ± 6° at a frequency of 30 Hz for 22 h under 2 450 N (maximal contact pressure 1.87 GPa) alternatively 4 450 N (maximal contact pressure 2.28 GPa). After the test, the bearings are inspected visually and the loss of mass on the raceways is determined. According to [23], mass losses below 5 mg are considered acceptable but this criterion has been progressively tightened to 2 mg or even 1 mg for very good greases. The ASTM test is referred to as both a False Brinelling and a Fretting test. As it uses oscillating angles of ± 6°, it qualifies as a test for False Brinelling as per the definitions in this work. Overall, the test has a relatively strong scatter, which is why an alternative high-frequency, linear oscillation test device has been made available in 2016 (Schwingungs-Reib-Verschleiß-Tribometer or SRV, ASTM D7594 [30]). However, this SRV test is more representative of Fretting conditions (pure sliding, point contact, amplitude 0.3 mm). Particularly in the American market, all high-performance greases come provided with Fafnir test values. The test is not as widespread in Europe but has been included in the current NLGI specifications for high-performance greases [31].

4.2. SNR-FEB2 test (NFT60-199)

The SNR-FEB2 test is regulated by the French standard NFT60-199 [29]. The corresponding test set-up was developed and marketed by the rolling bearing company Société Nouvelle de Roulements (SNR, now NTN-SNR Group) [29]. Currently, there are no suppliers of this test stand but older devices or self-made ones are in use (Fig. 7). In the test, axial deep groove ball bearings (type 51206) are tested in two units (left and right) with an oscillation angle of ± 3°, at 25 Hz and a normal force of 8 000 N (maximal contact pressure approx. 2.3 GPa) for

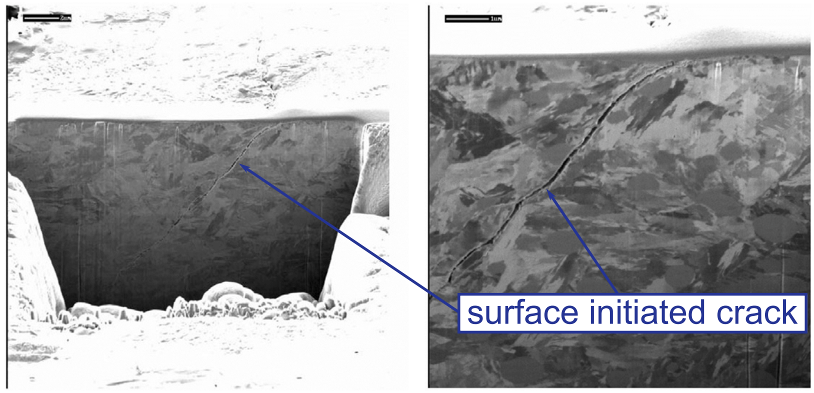

Fig. 6. Crosscut at the edge between sticking zone and microglide zone prepared by focused ion beam (FIB) shows a typical micro crack.
Source: KTM.

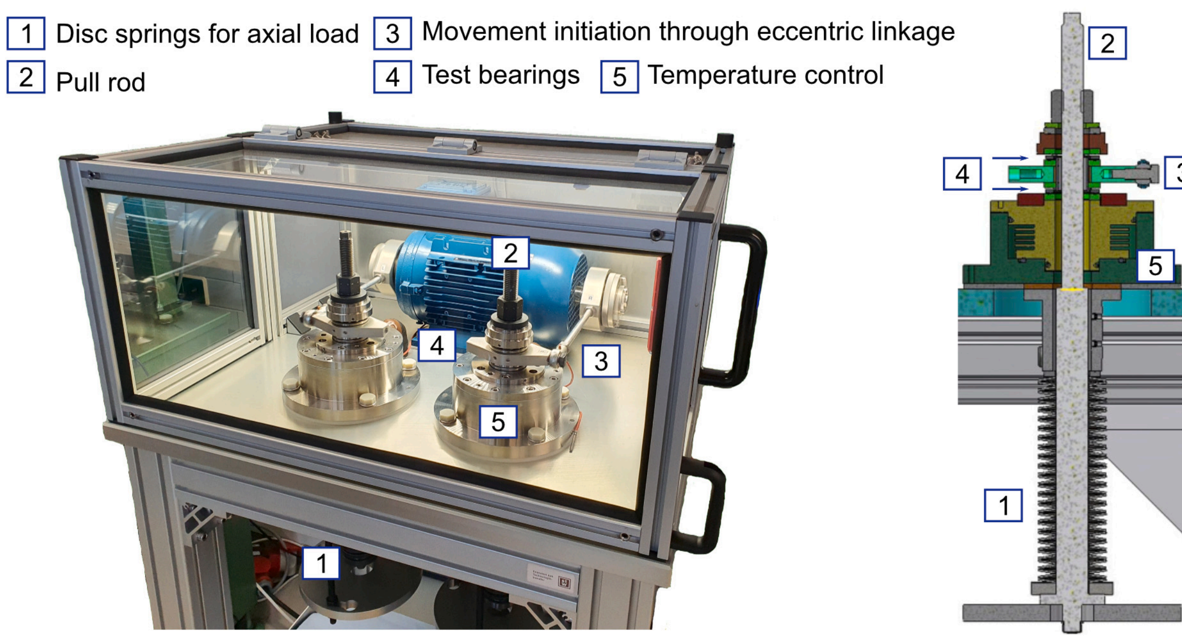

Fig. 7. Modern SNR-FEB2-test rig at the KTM (Photo and sectional drawing).
Source: KTM.

50 h. Test temperatures are usually either room temperature or −20 °C. After the run, a visual inspection and weighing of the rolling bearings are carried out. Since poor performing greases produce relatively deep hollows, the depth of the hollow is often also specified. In the SNR-FEB2 test, loss values < 2 mg are considered very good in terms of lubrication performance whereas values of up to 5 mg are still classified as good [23], (Fig. 8). Damage, once initiated, escalates very quickly which can then lead to a relatively large spread of the results. The test is widespread in Europe, particularly in the wind power industry, and is often required prior to commercial release. It emulates quite well the small pivoting movements typical of the pitch bearings in wind turbines. However, the investigation of Standstill Marks requires modifications of the experimental setup.

4.3. Non-standardized test methods

The so-called "IME Ripple Test" has not been registered as an official test protocol. It deserves mentioning in this context as it is a common and frequently required approval test for lubricating greases for wind turbines in Europe (similar to the SNR-FEB2 test [33]). The test utilizes 4-point ball bearings (QJ212-XL-TVP) tested for 1 million load cycles under an oscillating normal force of ±70 kN with a frequency of 10 Hz. Water with a 1% salt content flows at 6 ml/min through the bearings to simulate the corrosive conditions of offshore marine wind turbine parks. Micro-sliding movements occur in the contact points due to the alternating pressures and elastic deformations. The markings are thus comparable with those for small $x/2b$. "Good results" are those where the maximum scar depth of the wear is less than 10 μm, where the average depth is less than 3 μm, and where the degree of corrosion is better than 2 (1 = no corrosion; 5 = severe corrosion) (Fig. 9) [34]. Other non-standardized test methods for oscillating bearings are being developed and applied in Germany [35–44].

5. Fundamental research

5.1. Single-contact testing: Transient films in oscillating rolling contacts

As evidenced by the grease testing standards previously discussed, lubrication plays a fundamental role within the contacts found in oscillating bearings. Single-Contact rolling experiments have enabled the detailed study of lubricating films, subjected to both changing speeds

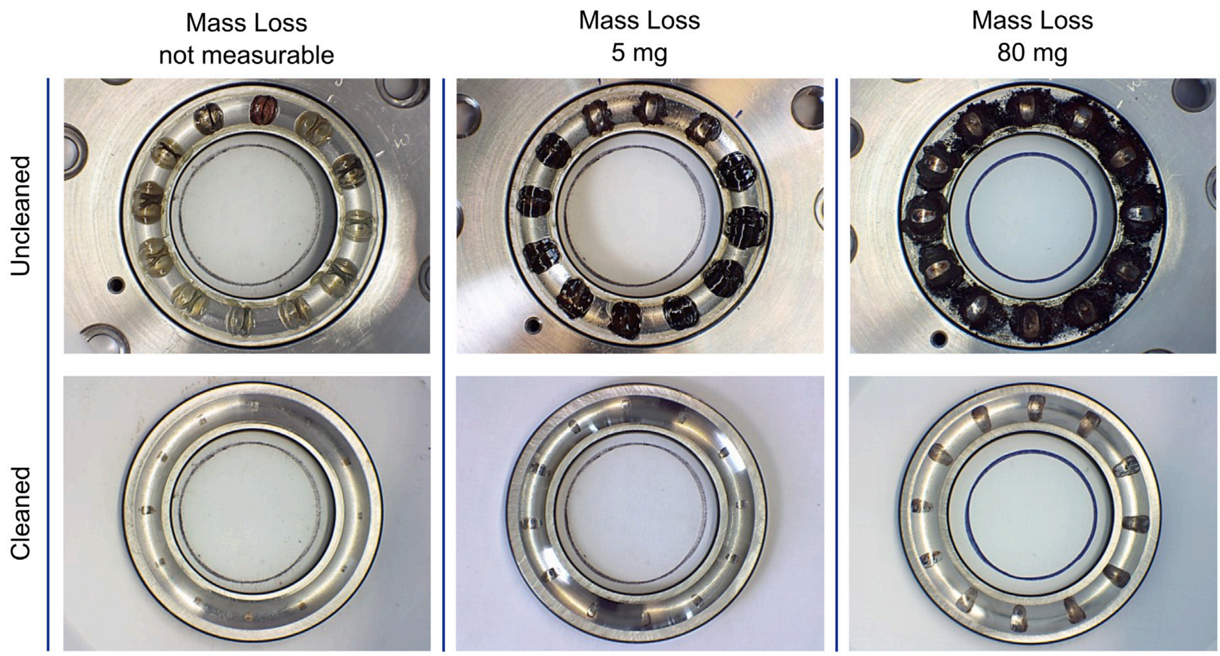

Fig. 8. Different mass losses after an SNR-FEB2-test [32].

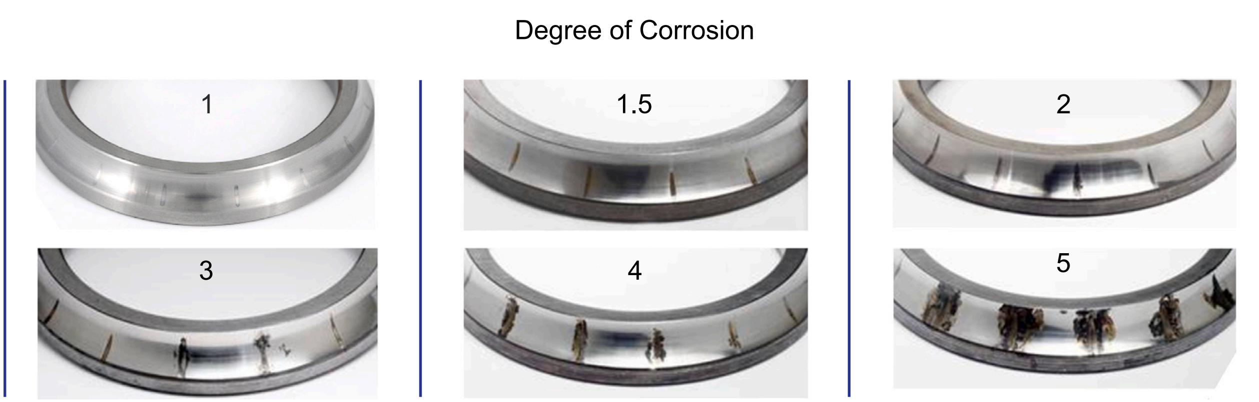

Fig. 9. Degree of Corrosion.
Source: IME.

and motion reversals. Such conditions are analogous to what oscillating bearing contacts must endure under real-life operating conditions. The reduced complexity of the singular contacting bodies facilitates the use of optical interferometry to quantify lubricant film thickness. Optical interferometry was used to study lubricant films as early as 1919 [45]. It was also used for the study of elastohydrodynamic lubrication (EHL) during the 1960s [46–50]. Nevertheless, it would not be until 1991 that initial attempts at the experimental study of transient EHL films were first undertaken [51]. The measurement accuracy and speed that are required to study these conditions would only be achieved in 1997 [52,53], enabling new insights into transient EHL lubrication.

5.1.1. Transient films in oil lubrication

Results from motion-halting experiments on the time-dependent collapse of an oil film show 2 stages [52], (Fig. 10). Initially, the film's height rapidly drops to about half of its original magnitude. Subsequently, a slow film thickness decay ensues. These two stages of oil film collapse were later observed and explained further [54]. The first stage of film collapse results from a superposition of lubricant entrainment and squeeze effect. This takes place during the short deceleration phase before motion is fully halted. During the second stage of film collapse, film thickness decreases due to a pure squeeze effect [55]. This thickness reduction is more pronounced in the contact periphery. The outflow of lubricant from the central contact zone is restricted due to the strong pressure-driven increase in viscosity. The film height at the start of the second stage was shown to be independent of the initial entrainment speed. Over a long period of 30 s, the lubricant is gradually squeezed out of the contact until the film thickness approaches zero. The shape of the lubricant entrapment is strongly dependent on the viscosity and the pressure-viscosity-coefficient of the lubricant. For oscillating bearings, the first stage free-fall reduction of the lubricant film thickness at the reversal point could result in contact between asperities, but a separation of the surfaces by entrapped lubricant is also possible despite zero entrainment speed. At the instance of absolute standstill, it is assumed that a small film thickness is not inherently problematic. For a complete oscillation, however, the lubricant film after the motion reversal must also be considered. This is precisely the focus of [56], where complete sinusoidal oscillations were studied, showing that immediately after the reversal of the movement, the previously described peripheral film constriction develops and, after entrainment reversal, the constriction moves through the entire contact area. This leads to a sharp drop in central film thickness as the

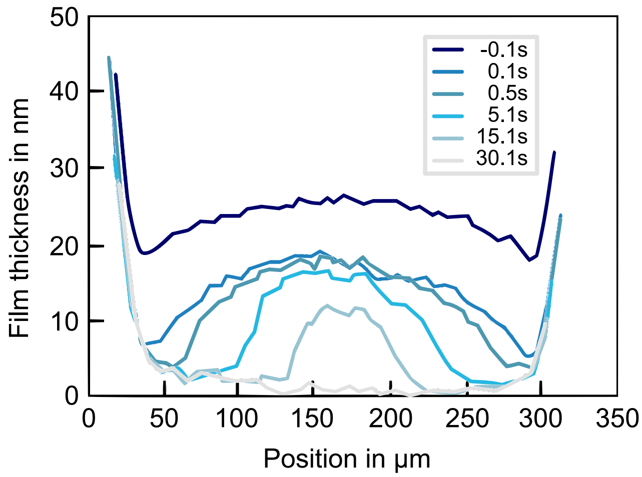

Fig. 10. EHD film shapes across the contact in a rapid halting test.
Source: Reproduced from [52] with permission from Elsevier.

constriction reaches the contact center. The minimum film thickness remains relatively constant until the constriction has traveled through the contact area. This phase can be critical, especially under high-slip motion, as metallic contact combined with relative motion can lead to heat generation, breakdown of the lubricant film, and wear. Similar behavior of the lubricant film was observed for reciprocating motion with linear acceleration/deceleration (but under pure sliding and mixed rolling–sliding conditions) [57] and also for reversal of entrainment with nominally constant speed for a rolling contact [53]. Theoretical and experimental investigations on the lubricant film behavior under different forms of motion, such as start-up of motion, unidirectional start-stop as well as the behavior under varying acceleration and deceleration and the influence of slide-to-roll ratio have also been conducted [58–60].

The investigations discussed so far have taken place at $x/2b > 1$. Experiments for $x/2b < 1$ have also been conducted [61] and indicated that a boundary lubrication layer cannot be replenished and sustained for long. Instead, it has been described as becoming progressively smaller. This is in line with numerous experiments on bearings subjected to $x/2b < 1$, considered Section 5.2.2.

5.1.2. Transient films in grease lubrication

In grease lubrication, both the thickener and the base oil play a crucial role in developing and sustaining a lubricating film [62–66]. This adds substantial complexity to the lubrication condition. In a fully flooded state, grease, by itself, can show a thicker film than its corresponding base oil. This is because of its higher effective viscosity, owing to its thickener content. In such fully flooded conditions, the film behavior during halting and reversal is similar to that of oils and the film thickness can be calculated based on this effective viscosity [67]. However, in practice, fully flooded conditions are not sustained indefinitely. A churning process pushes the grease away from the contact track, resulting in an effective viscosity that is different from the nominal value. It also results in a certain degree of contact starvation, with the contact track now populated by indeterminate amounts of released base oil, loose thickener clots and, eventually, a thin semi-solid thickener layer on the contacting surfaces [64,68]. Hence, results from experiments on oil lubrication are only of limited relevance for research on grease lubrication. For example, only a few oscillation cycles are needed for starvation and the associated pronounced decrease of the grease-lubricant film thickness to occur [69]. The starvation effect becomes stronger with increasing loads and entrainment speeds. Starvation is most pronounced in the middle of the track between two reversal points at maximum entrainment speed. This phenomenon resembles the behavior of rain and windscreen wipers in the middle of their sweep. At the reversal points, replenishment of grease can occur randomly, so that the maximum lubricant film thickness occurs at the reversal points. This starvation-driven film reduction makes grease lubrication particularly critical for use in oscillating rolling bearings. However, lubricating film recovery is possible through bleeding base oil from the grease reservoirs at the sides of the rolling contact. The usefulness of such a mechanism has already been highlighted for steady state conditions [63]. Replenishing from oil-rich reservoirs at the edges of the contact track has been directly observed in oscillating rolling contacts using a fluorescent marker in the base oil [70]. Variable frequency over-rolling experiments show this oil replenishment mechanism particularly well [68]. Oil appears to re-enter the track perpendicular to the motion of the ball, but starvation ensues if the over-rolling frequency is increased. The inability of this replenishment mechanism to work at higher frequencies explains high-frequency induced starvation (Section 5.2.1). The base oil simply does not have enough time to achieve sufficient replenishment of the contact track.

The investigations into oscillating rolling contacts under grease lubrication that have been discussed so far were under the condition of $x/2b \geq 1$. Regardless of whether grease or oil is the lubricating medium, there is only a small body of literature available that reports on investigations under the condition of $x/2b < 1$. It has only been shown that for $x/2b < 1$, the shape of the film profile remains mainly unchanged from its initial state during oscillation, but the maximum film thickness decreases gradually with rising cycle count, inevitably heading towards a lubrication failure [71]. Further research is necessary both for these smaller amplitudes, and in general, for the use of alternative techniques for film thickness quantification, such as capacitance or impedance methods [72], which can be more readily translated into practical applications.

5.2. Lab-scale bearings

The topic of Standstill Marks and False Brinelling has been intermittently investigated since the 1970s. It was not until 2006 that it once again became the focus of a more persistent research effort [32,37,41,73]. The main findings are presented here in terms of load, frequency, temperature, lubrication, and bearing design. The sections below will discuss False Brinelling ($x/2b \geq 1$), Standstill Marks ($x/2b < 1$), bearing design and coatings, and operating conditions.

5.2.1. False Brinelling: $x/2b \geq 1$

Bearing motion halting and reversal experiments [74] reveal that grease-lubricated bearings experience a quick collapse of the film during motion halt, but film thickness will be surprisingly large upon restart. The long time scales involved in this restart film build-up preclude the reliance on this effect in most oscillating bearing applications and dictate that mixed or boundary lubrication will prevail (Fig. 11). This transient nature of grease within oscillating bearing contacts is in agreement with general observations of single contact experiments (Section 5.1.2).

Grease adequacy has a stark effect on the resulting damage progression in conditions similar to the SNR-FEB2 test [75]. Poor quality of grease will result in a polishing effect after just one minute. The contact track is subsequently covered with red-brown oxides. After an hour these oxides are ejected and the contact track is free of these red-brown products. After 5 to 6 h, increasing surface defects emerge. These disappear again after a long running time (15 h). With a grease of a good quality, the prevalence of the emerging surface defects and the oxide products is noticeably attenuated, even though a similar initial polishing occurs as well. Surface roughness generally increases over time with exceptions for the much smoother deep depressions occurring with a poorly performing grease. This damage progression was imaged for increasing cycle counts in [32]. The wear coefficient in False Brinelling is particularly high in the early phase when oxide products

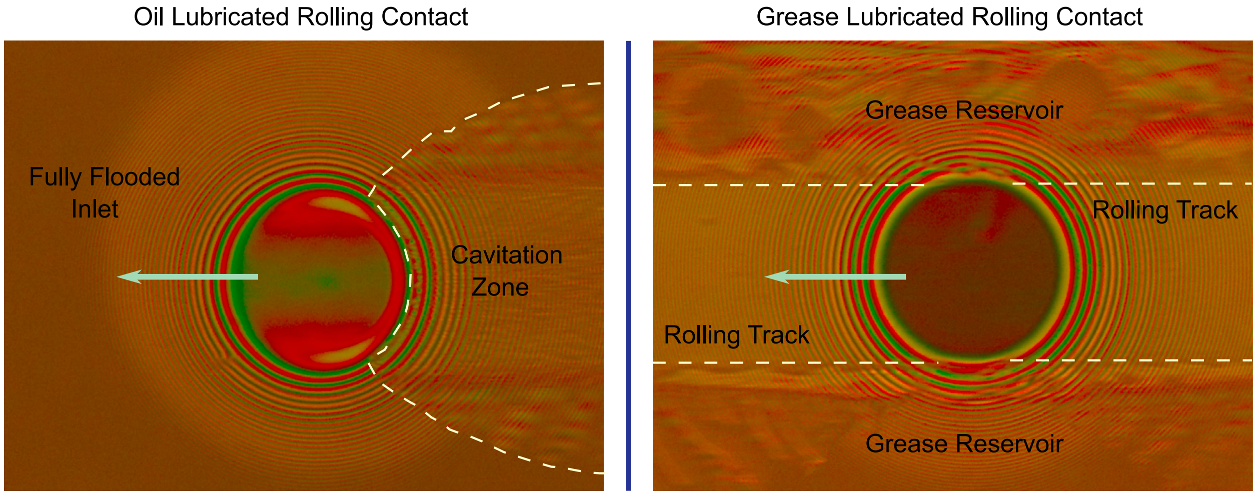

Fig. 11. Illustration of the basic difference between grease and oil lubrication of an oscillating rolling contact. Shown are the interference images of the rolling contact for an oil lubricated contact (left) and a grease lubricated contact containing the same oil as base oil (right) during an oscillation after 1000 oscillation cycles. The arrow indicates the direction of movement.

and roughened surfaces are formed due to adhesive damage [76]. The wear curve then takes a degressive course and the wear coefficient decreases, which is accompanied by a mirror-like appearance of the worn surface. The study of greases undergoing the Fafnir test shows that the ability of the grease to release oil is the critical factor that determines its wear attenuation potential [77]. Grease oil release (often referred to as bleeding) largely determines whether the ensuing wear process will be that of "good" or "poor" grease. Other grease rheological parameters, such as storage modulus, yield point, and adhesion force jump are insufficient indicators of grease performance [40]. Critically, the measurement of grease bleeding rates relies on heating a static grease sample [78]. This is different from the actual oil release mechanisms taking place in oscillating bearings, where temperature is not dictated by the self-induced heating of the bearing, but rather by its surrounding environment. This discrepancy between "measured" and "effective" bleed rates is systematically probed in [79]. Here, a calcium sulfonate grease is showcased. It has a measured bleed rate of nearly zero and it outperforms lithium and polyurea greases (having much larger bleeding rates and oil contents) in Fafnir tests. Although the sulfur in the calcium sulfonate grease might be acting like an Extreme Pressure (EP) additive, the oil release mechanisms responsible for replenishing the contact tracks are not well understood. These oil release mechanisms are the result of a complex interplay of thickener particle geometry and structuring, and their interactions with the base oil. These ultimately dictate the sensitivity of oil-rich grease reservoirs to mechanical disturbances, such as shearing, capillary forces, and other surface-wetting interactions that take place within the crevices at the immediate vicinity of the contact track.

For greases with varying rheology, this is convincingly illustrated by the onset of starvation effects, which has been shown to be dependent on the oscillating amplitude and frequency [37,38,80]. The degree of starvation in the contact hinges on a balance between lubricant displacement by the rolling element and contact replenishment by the lubricant [37]. A secondary re-lubrication mechanism is also emerging at larger angles, most likely related to grease being taken by the rolling element from the bearing cage. Although the benefit of high oil bleed greases is generally understood to involve base oil replenishment of the contact track, ease of oil bleed may also facilitate the formation of a compact thickener layer that protects the surface [81].

For these large oscillation angles, the use of different types of grease additives has been reported to result in performance improvements, despite early research suggesting negligible effects (Section 2). It must also be considered that the grease carrying the additives can have an impact on their effectiveness. Extreme pressure phosphate additives have been shown to reduce raceway wear more reliably in urea than in lithium soap grease [82] and graphene platelets have shown reductions in friction torque, both as an additive in barium complex soap thickened grease and as a dry coating [83]. Triisopropyl Borate did not show a significant effect when added to a lithium complex grease [79]. The study of fully formulated greases (of which only a partial content description is available due to commercial considerations) often obscures whether the observed performance is a product of grease type and rheology, a specific additive, or the whole additive package [84]. The use of dithiocarbamates, dithiophosphates, and metal sulfonates as grease additives were studied with ToF-SIMS in [85]. Unfortunately, the bearing contact conditions were insufficiently described in this study. A similar situation was present in [86], where substantial wear reductions were obtained using nanocomposite coatings consisting of nanocrystalline metal carbides embedded in amorphous hydrocarbon or carbon matrices. With lubricant ejection being a particular concern, coatings have an inherent appeal in oscillating bearing applications. Coated oscillating bearings are most commonly found in space applications, where the space environment, together with many other considerations, limits available lubrication approaches and offsets the additional cost of coatings. Hence, further discussion of coatings can be found in Section 7.

In some applications, oscillating bearing contacts operate at pressures that are high enough to require the consideration of plastic effects. Such conditions alter the resulting bearing degradation mechanisms [87]. With contact pressures in excess of 4 GPa, failure occurs once a subsurface volume of plastically deformed material undergoes fatigue and detaches from the ball [87]. The damage takes place at the highest loaded rolling element [88]. XPS and TEM analyses revealed that a modified layer is formed on the surface of the raceway with a structure and composition different from that of the steel substrate [89]. This layer is formed according to the sliding ratio, with higher sliding favoring the development of this modified layer. The additive package of the lithium complex grease employed is not fully disclosed, but elements found in the chemical analysis of this modified layer show that the grease chemistry is tightly related to it. The comparison of deep groove ball bearings and spherical roller bearings showed that, for spherical bearings, damage does not initiate at the rolling elements, it initiates with macroscopic detachments of material from the raceway [90]. Hybrid bearings with silicon nitride balls display a distinct and more gradual damage evolution when compared to steel bearings under these conditions [91].

5.2.2. Standstill marks: x/2b<1

Even though, at very small oscillation angles, a sticking zone appears in the center of the contact track which cannot simply be filled by free surface base oil re-flow, the lubricant also plays a decisive role in the initiation of Standstill Marks damage. In this case, it is important that adhesion in the slip zones at the edge of the contact zone is prevented by the lubricant. Micro-capillary effects due to the surface roughness seem to be sufficient to ensure at least a certain amount of lubricant penetration into the loaded contact area. This assumption is supported by the fact that an increased surface roughness is beneficial against damage initiation [11,19,35]. The role of the base oil is crucial in this area. This should already be sufficiently appreciated from the simple fact that an oil is significantly better in preventing wear than a grease [35,41,92]. When using a neat base oil, both viscosity and lubricant chemistry are important properties. At low temperatures (−20 °C) the severity of wear is increased compared with 20 °C. This is attributed to the higher viscosity and the reduced mobility of the base oil [41]. However, no correlation between wear volume and oil viscosity could be found when base oils in a viscosity range of 11.6-90 cSt were tested [25]. A comparison of different oils (poly-alpha-olefin (PAO), trimethylolpropane ester (TMP), polyglycol (PG), polyfluorinated polyether (PFPE), paraffinic mineral oil, and naphthenic mineral oil) at the same viscosity of 46 cSt at 40 °C showed that differences between surface damage patterns also arose due to the type of base oil [92]. Based on a visual evaluation of the damage marks, the paraffinic mineral oil showed the best results followed by the naphthenic mineral oil and PG. Even though the damage could not be prevented, significantly fewer corrosion products and surface disruptions were visible.

Approximately 90% of all rolling bearings are lubricated with grease [65]. Hitherto, no description has appeared in the literature of a grease formulation that can completely prevent damages [32]. A large number of model and industrial greases with varying NLGI grades (1–2), base oil viscosity (11.6-395 cSt at 40°), thickener (lithium & lithium complex, calcium complex, polyurea, silicone), and base oil type (PAO, ester mix, mineral oils), as well as additives (anti-wear and extreme pressure additives, solid lubricants), have been already tested [25,35,41,42].

In general, the sources agree that greases with a low NLGI grade, a low base oil viscosity, and high oil separation are to be preferred, although there are exceptions. It is stated [35] that a low dynamic viscosity of the grease, as well as a low loss modulus, prove to be effective against Standstill Marks. However, all these properties are usually positively correlated. This is consistent with the investigations at different test temperatures. In all sources, the damage increases with decreasing test temperature, which generally results in an increased viscosity of grease and base oil and in a decreased oil separation.

Looking at the influence of different thickener and base oil types, the picture is no longer so clear. Good results are obtained at room temperature, especially with mineral and polypropylene glycol base oils [41], which is in agreement with the studies on base oils [92].

A lithium complex thickener generally performs better than a polyurea thickener regardless of the ambient temperature [42]. Tests of different greases with the same base oil (PAO, 100cSt at 40 °C) but different types of thickener (calcium complex, sodiumtherephtalamat, lithium complex) and with NLGI grades varying from 1 to 2 at −10 °C revealed that all greases performed badly [35]. However, the calcium complex thickener showed superior performance at room temperature, but the performance advantage cannot be completely decoupled from the comparably low NLGI grade.

The use of additives can significantly reduce the severity of Standstill Marks, as shown in [42]. A combination of lithium thickener, 2% neutral calcium sulfonate (anti-corrosion) and 2% ZDDP-2-ethylhexyl (anti wear) resulted in a reduction in wear volume of 80% compared to the neat model grease at room temperature. The use of ZDDP additives in PU and Li greases was also reported to be advantageous in minimizing wear [93]. An admixture of 2% MoS_2 (black solid lubricant) could contribute to a significant reduction of wear volume at room temperature in Li greases [42]. 2% of Zn pyrophosphate (white solid lubricant) has been found to be less effective. [35,94] claimed that solid lubricants only have a small effect at the beginning of an experiment. A reduction in wear volume of over 60% at room temperature could be achieved through the addition of complete additive packages including MoDTC and MoDTP to a lithium grease with a PAO base oil (70 cSt) [25]. The combination of thickener and additives was also crucial as shown in [42]. The use of the additives listed above in a lithium thickener generally reduced wear, whereas in the case of a polyurea thickener, it lead to increased wear. At very low ambient temperatures (−20, −39 °C), the wear volume can be up to ten times higher than at room temperature and the effect of additives is marginal at best.

Influence of the bearing design and coatings

Multiple strategies to achieve raceway wear attenuation have been explored, including some pertaining to bearing design. Increasing the raceway surface roughness has been shown to decrease wear [94]. The working reasoning is that the prominent surface features serve as lubricant reservoirs, which, together with micro-capillary effects, improve the lubrication condition of the contacts. Furthermore, higher local contact pressures at the asperity peaks may also result in an overall decrease in partial sliding. Another way of achieving reductions in slip is to slightly decrease the conformity of the ball-raceway contacts, which reduces Heathcote slip and has been shown to reduce wear [95]. Changing the bearing type altogether can also have an effect in this regard. Angular contact bearings have been shown to be more susceptible to wear when compared to tapered roller bearings [41]. The use of different ball-raceway materials has not been thoroughly researched yet. However, changing the mechanical properties of the ball-raceway materials will affect the slip distribution at the contacts. For example, the use of ceramic balls will result in a contact pressure increase at the same normal force and geometry. The contact ellipse size is thus reduced and the partial sliding conditions change [96]. Despite this, hybrid bearings, i.e. with ceramic balls, do not show outstanding results, even though this material implies a clear reduction of the adhesive component of friction [24,94,97]. The use of balls made of zirconium oxide, which has a similar modulus of elasticity as steel, does not result in any advantages either, even though it is also less susceptible to adhesive interactions than steel while not significantly altering contact pressures at the same geometry and load [24,97]. Another way to reduce adhesive effects involves the use of coatings. Carbon nitride coatings could not withstand the applied loads in standard tests [98]. A DLC coating (non doped a-CH), however, showed excellent results and a large potential for future development [92]. The drawbacks are the high cost and the fact that the coatings have to be sufficiently resistant to wear and fatigue under conditions of normal operation. Black oxidation of the raceways leads to damages remaining similar or being worse, both at room temperature and at low temperatures. This approach is therefore not recommended [25,32].

Influence of Operating Conditions

Normal load levels at the contacts, dynamic forces, and oscillation frequency affect the resulting wear progression. In an effort to clarify the influence of such individual parameters, extensive research has been conducted [25,32,41,97]. Tests using the standard SNR-FEB2 revealed that normal force is only of secondary importance [94]. Damage can occur at relatively low contact pressures since the percentage of sliding at the contact point increases [96]. Even relatively small normal forces which arise from the dead weight of rolling elements or from the weight of the races during transport are sufficient to cause damage. However, as the normal force increases, the damage usually increases as well.

Dynamic, and even static, loading of the bearings are significantly less critical for Standstill Marks if small angular oscillations of the

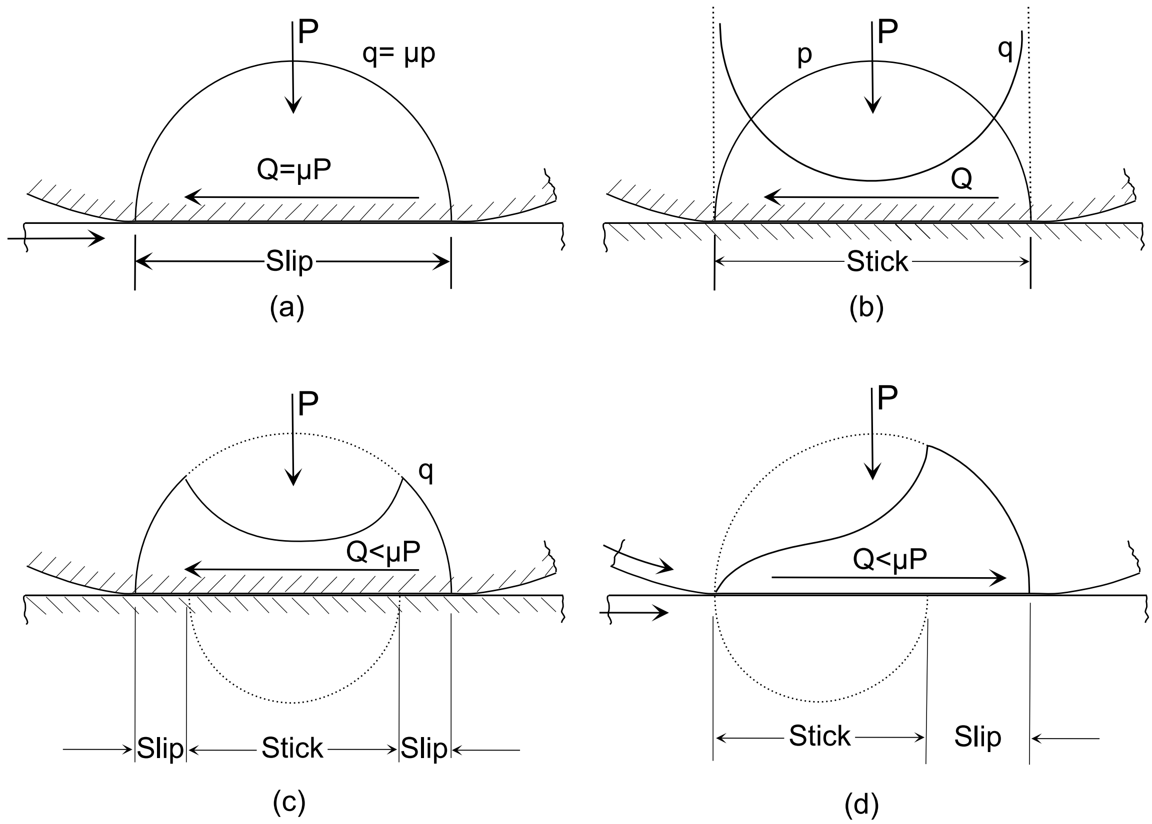

Fig. 12. Distribution of tangential traction in (a) sliding contact, (b) static contact with all slip prevented, (c) static contact with slip at the edges, and (d) rolling contact with slip at the rear.
Source: Reproduced from [100] with permission from Sage Publications Ltd.

raceways are restricted and are tending towards zero [41,99]. The influence of oscillation frequency is not so pronounced, based on the narrow range of frequencies studied so far (5 Hz–25 Hz in [24] and up to 45 Hz in [41]). The damage is marginally aggravated with increasing frequency to 25 Hz and this trend becomes more pronounced as the frequency continues to rise to 45 Hz. Since the contact point is never fully opened under the conditions considered, replenishment time is not as important as with larger amplitudes ($x/2b > 1$) [35,94]. It can therefore be concluded that this damage worsening should be mainly caused by inertial forces and an increased degree of sliding [24]. The impact of significantly higher frequencies (e.g. magnetic actuators) on Standstill Marks has not yet been reported.

In general, lowering the operating temperature below 0 °C will increase wear and tribo-corrosion [25] and will lead to a larger number and volume of wear particles [24,25]. This is related to a lowering of the lubricant's fluidity. There are, however, some greases that deviate from this behavior [24,41,97].

Caution is recommended when extrapolating these results to other tribological conditions or systems, in view of the complexity of the tribochemical processes involved.

6. Oscillating rolling contact models

6.1. Analytical models

A sensible starting point in understanding oscillating bearing contacts is to consider first a simplification: a static Hertzian contact loaded tangentially. It is assumed that full sticking occurs throughout the entire contact interface. In that case, infinite traction develops at the contact periphery upon the action of the tangential load (Fig. 12b). However, Cattaneo [101] and Mindlin [102] independently deduced that there is a peripheral contact region that experiences slip (Fig. 12c). This slip area occurs where the traction exceeds the product of normal pressure and friction coefficient. Thus, a central area of sticking develops and slip occurs in the surrounding annulus (Fig. 12c). A lower friction coefficient will result in a smaller central sticking area. This has been corroborated experimentally using base oils which are known for good lubricity, such as esters or polyglycols, both of which result in considerably smaller static friction zones [24]. As the tangential load is increased, the area of pure stick reduces in size until the annular slip area grows to occupy the entirety of the contact area. Only then does gross sliding ensue (Fig. 12a). The simplifications involved in achieving the neat analytical expressions for exclusive slip relief in [101–103] have been challenged [104]. A more complex process of relieving this infinite traction through the additional contribution of plastic effects was suggested. Yet, the main intuitive takeaway remains: the contacting interface, particularly at the contact periphery, is subjected to surface-altering interactions well before macroscopic movement occurs.

The next consideration is the rolling motion. Pure rolling implies that, at the contact point, both bodies have the same velocity. However, loaded elastic bodies do not meet at a single point, they develop a contact area with an associated deformation field. It is convenient to visualize the deformation field as static, and points within the material bodies as "flowing" through it as rolling occurs. Pure rolling of a Hertzian contact would imply that all points of the rolling bodies share the same velocity as soon as they enter the contact area. This is not the case. Even if the tangential or traction force is not present, a contact will experience slip upon the application of a normal load, under the condition that both bodies have different elastic properties and/or if they are geometrically different, the so-called Reynold's Slip [105]. Additionally, Heathcote [106] attributed the rolling resistance of a ball in a groove to the slip occurring as the result of the different circumferential velocities at different radii. This slip, now called Heathcote Slip, resulted in slip occurring everywhere in the contact area, the

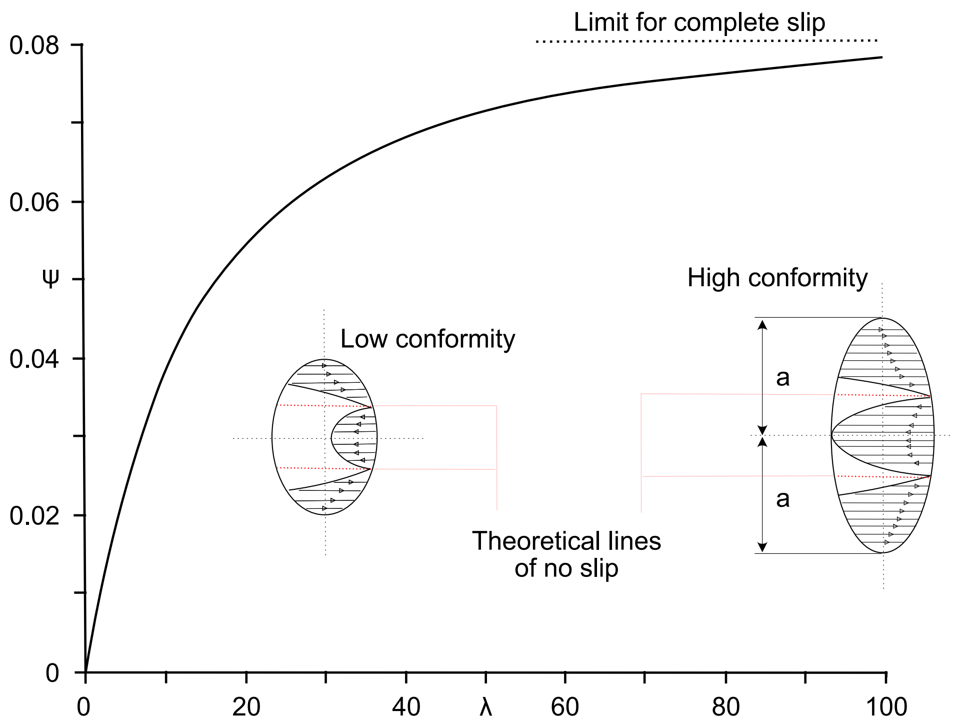

Fig. 13. Osculation's influence on stick/slip zones; broken line - limit for infinitesimally small slip; chain line - complete slip [115]. Where λ represents the osculation parameter and Ψ represents the rolling resistance. For specifics on the formulation of these parameters refer to [115].
Source: Reproduced from [115] with permission from Sage Publications Ltd.

only exception being two no-slip lines. However, it was argued that exclusively restricting the sticking to two lines dismisses the ability of materials to elastically accommodate such velocity differentials, at least to a certain degree. It was proposed instead that the contact area can be divided into stick and slip regions [107]. The same author [107] proceeded to use the Carter–Poritsky theory of a locomotive driving wheel [108,109] in thin strips along the rolling direction of an elliptical contact area and derived expressions for tangential tractions, micro-slip, and rolling resistance due to slip. Fig. 13, illustrates the influence of the osculation on the formation of the stick and slip zones, where a lower coefficient of friction and/or high ball-raceway conformity will lead to a larger slip area. Johnson [110] was the first to propose a theory on the effect of a spin component upon rolling elastic bodies, another source of slip that can be expected in rolling element bearings. A more refined analysis of this problem [111] yielded solutions for circular contacts and Poisson's ratio tending to zero. Kalker could remove this limitation of the Poisson's ratio [112,113] and proceeded to generate the first framework for arbitrarily accurate solutions of the elastic rolling problem. A summary of this and other related works can be found in [114].

Models outlined thus far describe steady-state rolling. However, stick and slip zones do not develop instantaneously [96]. Fig. 14 shows the changes within the stick/slip zone starting from idle. The traction distribution within the leading edge moves through the contact until it is engulfed by the slip at the trailing edge [116]. It is only at this point that a steady rolling model becomes a valid representation. The consideration of motion reversal is analogous, but transient effects are more pronounced [116]. Note that travel in the order of the contact width is required to reach steady state rolling [116]. Thus, in oscillating bearing contacts, it is unreasonable to expect steady-state rolling if $x/2b < 1$. For larger $x/2b$, steady-state rolling is an approximation incurred at the expense of neglecting the effects of motion reversal towards both ends of ball travel. Only 2-dimensional exact solutions to this problem have been obtained [96,116]. The spinning moment (turning moment about the ball axis perpendicular to the surface) and lateral creep must be taken into consideration in the 3-dimensional case. Only approximated solutions with the use of superposition are available [96]. The consideration of inertial effects is outside the scope of these models. Inertial effects will inevitably gain importance at higher frequencies and may result in more extensive slippage, beyond what is already caused by the previously established sources.

The application of analytical approaches to the study of oscillating bearing contacts has resulted in the development of several computer-implemented iterative analytical models [114]. Recent implementations of iterative analytical models include the study of bearing contacts subjected to alternating axial and/or radial loads with a stationary rolling bearing [117–119]. Another implementation focuses on oscillating rolling contacts [120] involving the use of the minimum rolling energy to calculate the contact kinematics. Comparisons of these models with bearing tests in the absence of lubrication show that friction energy density within the contact track, as estimated with the models, can be a good indicator of the resulting damage [120].

6.1.1. Dahl's empirical model

When a movement is initiated in an oscillating rolling contact, elastic deformation of the contact partners occurs first and micro-slip zones develop only in the peripheral contact areas, as explained earlier. A similar dynamic is present after the direction of movement of a rolling contact is reversed. This behavior, and the resulting rolling resistance, are thus initially governed by the stiffness of the contacting partners. As the motion's amplitude increases, the proportion of the surface that governs the response elastically decreases and the slip contribution increases. The rotational resistance continues to rise asymptotically until steady rolling is achieved. This behavior was first described mathematically by Dahl in 1968 [121]. Within this model, the hysteresis behavior can be defined on the basis of the steady rolling torque and the stiffness of the contact partners, the so-called "rest slope" (Fig. 15). The simplicity of this model very likely contributed to its quick adoption [122,123]. However, both the asymptotic value of steady rolling torque and the rest slope must be measured in order to implement the model. Dahl himself provided experimental details on how to measure these parameters in bearings [124]. However, it would

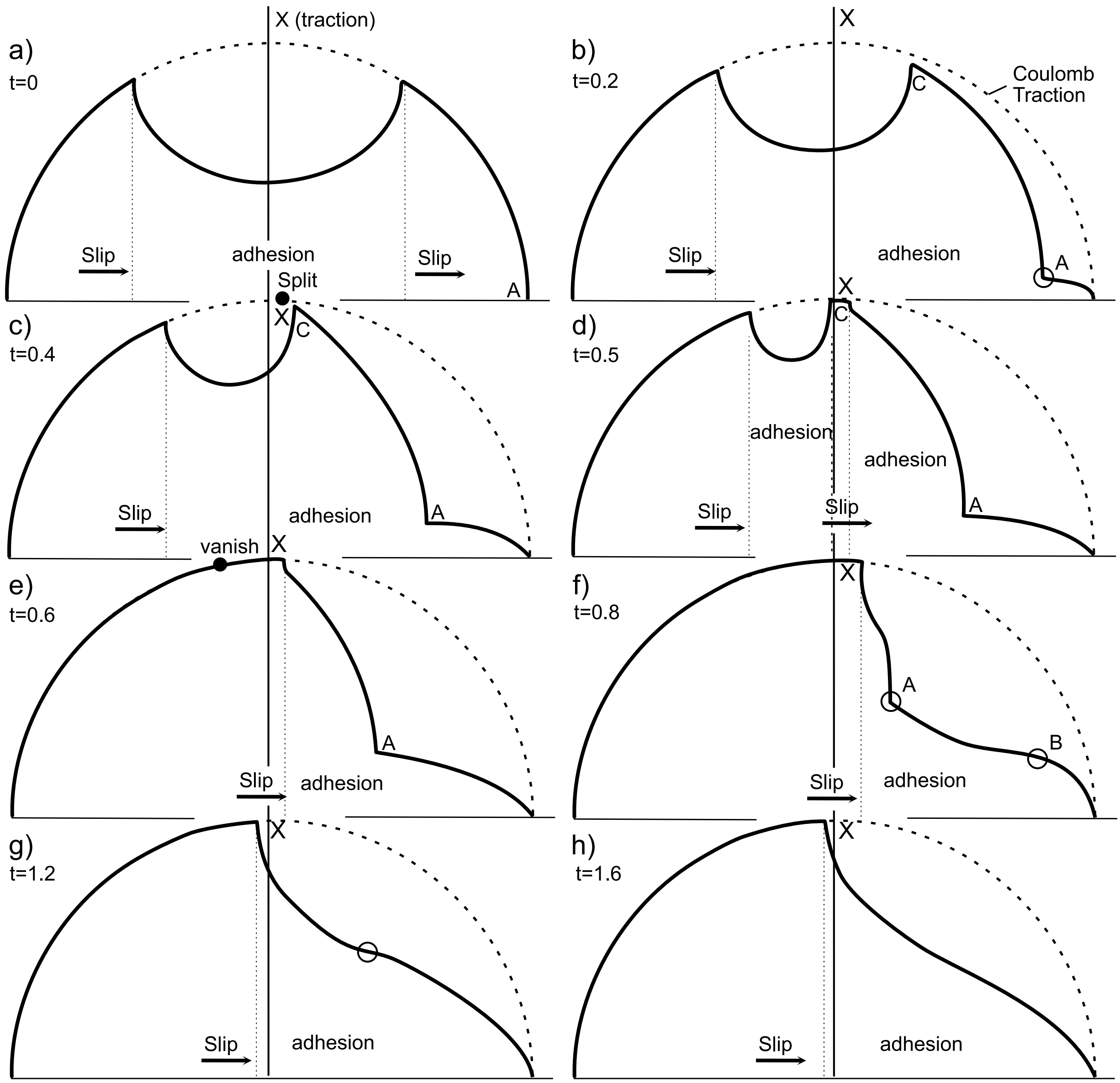

Fig. 14. Changes in stick/slip zones within the contact zones during start-up [116].
Source: Reproduced from [116] with permission from Taylor & Francis Ltd, http://www.tandfonline.com.

not be until 1987 that Todd et al. derived theoretical expressions for the steady rolling torque and the rest slope and, using Dahl's model, predicted and validated complete torque hysteresis for dry and oil-lubricated angular contact ball bearings [125,126]. Their approach also allowed taking into account the geometry of the bearings in order to model the influence of conformity and contact angle and thus Heathcote Slip and spin components. In 1992, Lovell et al. conducted extensive tests on the influence of various parameters (oscillation amplitude, angular velocity, normal load, the type and the viscosity of the lubricant, the material of the contact partners) on the rotational resistance hysteresis [127–129]. Experimental results were in very good agreement with the model developed by Dahl. It was shown that the hysteresis behaved in a linearly elastic way with minimal energy loss when the movement was reversed after very small angles in the pre-roll range. With decreasing angular velocity, increasing normal force, and decreasing the viscosity of the base oil, the rest slope increased. Steady-state rolling was achieved most quickly and with minimal energy loss in the unlubricated condition. In the case of oils with high viscosity, the condition of pure rolling was only reached much later. In experiments with silicon nitride balls, which have a significantly higher modulus of elasticity compared to steel, it was shown that the energy loss during oscillation was always greater in comparison to steel balls. This could be attributed to a steeper rest slope and higher resistance during pure rolling. Dahl's model has been widely adopted in the characterization of bearing torques for motion control purposes, where incorporating pre-rolling behavior into the control scheme improves response times and accuracy. Specific applications of this order can be found ranging from hard drives [130], to stage-controls for the purpose of nano-positioning [131].

6.2. Numerical models

There are a considerable number of applications of numerical models for rolling contacts with slip considerations within the contact mechanics field [114]. The use of such methods to specifically address oscillating rolling bearing contacts is a more recently documented occurrence. Schadow [97] presented a finite element model consisting of a rolling element and an inner and outer ring segment of an angular

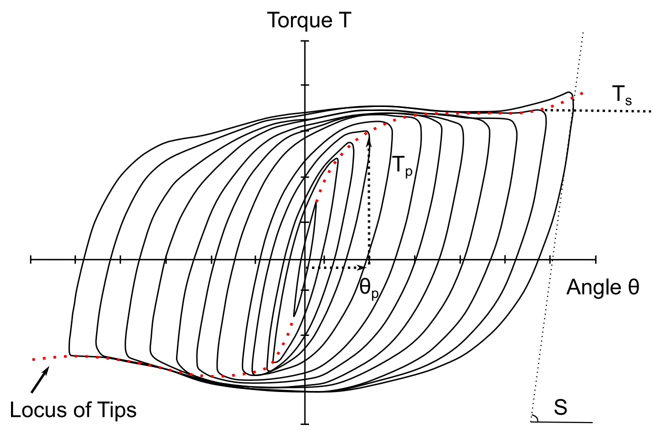

Fig. 15. Typical torque hysteresis for an oscillating angular contact ball bearing with the parameters for steady rolling torque T_s and rest slope *s*.
Source: Reproduced from [125] with permission from Elsevier.

contact ball bearing. The model allows the calculation of the pressure distribution and the slip components in the contact area between the rolling element and raceway due to the oscillation of the inner ring segment around the bearing's rotation axis. These pressure and slip magnitudes, calculated with a friction coefficient assumed to be constant, allow determination of the specific friction power, which according to [132], can be correlated with the resulting wear volume. A good correlation exists between this level of simulated specific frictional power and the local severity of wear damage in the bearing experiments [97]. However, no quantitative characterization of the wear was made. A comparable finite element model showed a good correlation between friction energy densities and damaged areas on the bearing raceway [36]. In addition, the influence of contact angle and osculation on the friction energy density was also investigated. It was shown that the maximum friction energy density during an oscillation cycle increases asymptotically with increasing oscillation amplitude until a steady state condition is reached [36,133]. This observation shows good agreement with the previously discussed considerations [121,124,125]. A FE-modeling approach was implemented to simulate micro-slip in the ball-raceway-contact of a thrust ball bearing due to changing normal loads [134]. A good correlation between the calculated frictional energy density and the damage areas occurring in bearing experiments was shown for dry contacts. An adaptive finite element model that used Archard's equation to predict the wear-induced evolution of the oscillating contacts in cylindrical roller bearings subjected to lateral and axial vibrations has also been reported [99,135]. Wear and friction coefficients were determined experimentally in reciprocating sliding tests and the model was verified using unlubricated bearings.

Massi et al. [87] presented a numerical model that was used to calculate the contact stresses and strain distributions in highly loaded oscillating bearings (considering both elastic and plastic properties of the bearing materials) and compared the results with experimental observations. The model shows that a thin layer of surface material remains in the elastic domain and that plastic alterations occur only in subsurface cap-like regions. This 30-micrometer thick layer of undamaged material is then replicated in an experimental setup and observed as structural differences in the ball material that emerged from an analysis by SEM. The tested bearing fails once this subsurface plastically deformed volume undergoes fatigue and detaches from the ball. The damages are of a similar shape to the residual plastic strain distribution computed in the model. Tonazzi et al. [136] presented a more in-depth exploration of the oscillating highly loaded bearing contact, employing a parametric study of friction coefficient, radial load, and surface conformity. Within the parametric analysis, the friction coefficient is increased in steps from 0.05 to 0.3. Revealing that, once the oscillation is introduced, the plastic behavior is completely different. The higher friction coefficient produces a migration of the plastic strain towards the surface. These differences in stress distribution are in good agreement with experimental results in bearings with and without lubrication [88]. The effect of radial load showed a predictable behavior where the plasticized zone increases in volume and that changing surface conformity resulted in changes to the dimensions and geometry of the plasticized zone while preserving its subsurface nature.

In general, numerical models have been used as *a posteriori* aids for understanding experimental and practical observations of oscillating bearing contacts. This limitation in usage stems primarily from the fact that these contact mechanics-based models do not consider the lubrication, both in terms of the lubrication condition and in terms of the chemical interactions of the lubricant components with the surfaces, which are a fundamental consideration in boundary lubrication.

7. Application-specific knowledge

7.1. Robotics

In principle, the operating conditions for rolling bearings in the field of robotics should favor bearing deterioration through an increased occurrence of either Standstill Marks or False Brinelling. However, there is, at the time of writing, only a small volume of published investigations on this topic. Operating conditions in robotics usually subject bearings to large oscillation angles [137]. Two angular contact ball bearings (type 6008, clamped with 2 000 N), subjected to 30° ($x/2b \approx 20$) oscillation angles at 1 and 5 Hz, showed signs of False Brinelling and tribo-corrosion only at the highest frequency. Insufficient lubrication could have been a probable cause. However, if the oscillating angle was further increased so that the area of the adjacent rolling element was swept over, the damage disappeared completely at both frequencies [137].

7.2. Wind power pitch bearings

Pitch bearings (or blade bearings) of wind turbines allow the rotor blades to turn around their primary axis and, by changing this pitch angle, the aerodynamic loads can be controlled. The turbine controller reduces the loads in case of excessive wind speeds. Pitch angles can also be changed individually to harmonize loads of the entire rotor and reduce fatigue of the structure. Pitch bearings have a preferred pitch angle and generate oscillations of varying amplitudes (see [36,138,139] for detailed explanations and descriptions of wind power pitch bearings). Pitch bearings of large off-shore turbines have a diameter of 5 m and are designed for extreme loads of 50 MNm and more [138]. Such bearings are made specifically for each turbine and they come at significant material costs. Sheer bearing scales, the lightweight designs of the surrounding structures, and the need to control rapidly changing wind loads by highly variable oscillating movements are some of the more challenging conditions of wind turbine operation. The unit cost of the energy generated by wind power, the paramount variable of determining how competitive this form of renewable energy is, makes efficient design indispensable.

7.2.1. Down-scaled bearings

A full-scale test of pitch bearings is both time and cost-intensive. Down-scaled tests constitute a more efficient solution to understanding damage mechanisms, influencing factors, and developing lubrication strategies. Schadow [41] and Grebe [32,35], refer to pitch bearings as a prime example of oscillating bearings but do not attempt to provide solutions to scaling down the operational parameters. Wandel [37] proposes scaling of the operational parameters via a dimensionless parameter (starvation number) and Bayer [38] confirms that such a transfer to large bearings and other bearing types is reasonable. None of the above-mentioned research groups directly tackle the problem of scaling down the operating parameters of wind turbine bearings.

An efficient scaling methodology is based on using *x/2b* ratio. Becker [39] scaled the oscillation angle by *x/2b* and maintained the entrainment speed of the rolling contact. This increased the frequency of the oscillation when compared with the results from a full-scale model. Controlled variables were the oscillation amplitude (constant) and the temperature of the bearing rings (between −20 and +70 °C). Six commercial greases (of unknown composition), each with and without the addition of a contaminating NaCl-solution differed in their suitability to prevent wear damage on the raceways [39]. A similar approach (scaling the oscillation angle by *x/2b*, maintaining entrainment speed) was used by Schwack et al. [36,140] to evaluate different greases at varying *x/2b* ratios. None of the tested greases was able to prevent the occurrence of severe wear at *x/2b* = 0.9 and *x/2b* = 13.3. However, a commercial lithium grease (NLGI 2, based on mineral oil) did prevent wear at *x/2b* = 30.7 [36].

An innovative approach to mitigate False Brinelling was demonstrated by Stammler et al. [138,141], when applying varying amplitudes to type 7220 angular contact ball bearings. The oscillation frequencies in the tested scaled-down bearings were kept similar to those encountered in the full-scale bearings. The test program's reference run (40 000 cycles, *x/2b* = 2.66) resulted in False Brinelling. When the sequence of narrow oscillations was interrupted by larger movements (every 30th cycle) no visible wear could be observed. Stammler coined the term Protection Runs for these larger movements.

Four-point contact ball bearings with a pitch diameter of 675 mm, which are closer to full-scale bearings were studied by Schwack et al. [142] and Schwack [36]. These bearings were manufactured using the same processes as those used for full-scale pitch bearings, including inductive raceway hardening. The lithium grease is the same commercial grease as in [138]. For *x/2b* = 2.67 and *x/2b* = 11.44, the raceways displayed False Brinelling at more than 80% of the contacts after 12 500 cycles. Based on the similarities between the damage marks of type 7208 angular contact ball bearings and 675 mm pitch bearing, it was concluded that scaling by *x/2b* is reasonable [142].

Although current scaling approaches appear to illustrate analogous wear processes, certain limitations persist and their impact remains to be addressed. The steel used to manufacture large pitch bearings is 42CrMo4, whereas smaller bearings are typically made of 100Cr6 steel. Discrepancies in surface finish may also be relevant since surface features are known to have an effect on boundary lubrication, lubricant retention, and wetting. Another limitation lies in the geometry of the bearings and their corresponding contact kinematics. For example, it would be expected that spin-slip contributions are over-represented in small bearing tests as opposed to actual slip conditions in large bearings. Furthermore, in wind power applications the pitch bearings are exposed to a varied sequence of load-amplitude–frequency combinations, which are also influenced by the specific pitch controller, over years of operation. Hence, scaled tests serve as a means to screen lubrication strategies, but not necessarily to provide accurate life estimations.

7.2.2. Full size bearings

Most full-scale experimental set-ups for testing wind turbine pitch bearings (Fig. 16) are operated by commercial companies which, as far as the authors know, do not usually publish detailed and scientifically relevant reports (as opposed to a plethora of publicity material [143–148]).

The Protection Run approach was demonstrated by Stammler [138] in full-scale pitch bearings (four-point contact, 5 m) on a hexapod testing set-up, applying a static axial load of 10 MN resulting in 2 GPa contact pressure. A steady oscillation (0.5 Hz, *x/2b* =3 and 40 000 cycles, and lithium grease as in [141]) resulted in False Brinelling. The same oscillation sequence when interrupted by Protection Runs (*x/2b* = 30) after each 30th cycle provides mixed results, with some contacts exhibiting False Brinelling and others not. These experiments show similarities between the damage marks of type 7220 angular contact ball bearings and 5 000 mm pitch bearings, indicating that scaling by *x/2b* is reasonable. The *x/2b* scaling was further confirmed by Behnke et al. [149], who studied full-scale four-point contact pitch bearings subjected to a static axial load and incorporating Protection Runs on a sandwich-type setup with variable rotating speeds, *x/2b*, and contact pressures [149]. However, the challenge remains, as it was shown that wear could occur for *x/2b* = 18.4 [149].

Field tests of pitch bearings in wind turbines are technically feasible but come at significant costs. To the knowledge of the authors, there are no data on the results of such tests publicly available.

7.3. Aerospace applications

Examples of oscillating bearings in aerospace applications are plentiful. They range from general pointing or scanning mechanisms for different types of instruments to the 34 bearings of a prototype high-pressure suite that will allow an astronaut to move without much impediment. Outer space presents a challenging domain for the implementation of oscillating bearings. The overarching design priority is the prevalence of low and consistent friction torques that allow for accurate, energy-efficient, and reliable bearing operation. These conditions have to be met while the bearings are operating under the extreme conditions of high vacuum, increased radiation, micro-gravity, bombardment by low-earth-orbit atomic species, and extremely wide operating temperature ranges. Increasingly ambitious mission objectives have to continue to rely on the predicted lifespan of oscillating bearings but are nevertheless making the conditions of operation ever more demanding. In spite of these ever-increasing challenging conditions, the extent of knowledge and data is only a fraction of what is available for more mundane terrestrial systems and devices.

The earliest published work on oscillating ball bearings in cosmic conditions dates from 1966 [150]. The aim of the study was to find lubricating media that showed potential for low-speed angular contact ball bearings. Although most of the tests were conducted under conditions of low-speed rotation in a vacuum, two lubricants were tested under oscillating conditions: a felt pad burnished MoS_2 film plus an adsorbed lubricant additive film and a silicate bonded MoS_2-graphite film. The latter was deemed more promising for both continuous rotations and oscillating motions (*x/2b ca.* 1). The observed torque increases were attributed to the wearing of the lubricant film and subsequent jamming of the bearings by the loose debris. Two years later, Demorest [151] produced a NASA technical report showing that bearings lubricated with similar dry films would meet the life requirements of the pitch and yaw gimbals of the Apollo Telescope Mount.

7.3.1. Transfer films

Another early example of oscillating bearing tests is found in [152], which evaluated a composite consisting of PTFE with MoS_2 and with glass fibers as a retainer material and found that the bearings were still in good condition after enduring 1 million cycles. If the oscillating angles are large enough, material transfer from the cage to the balls can have a positive effect. Transferring material from the cage to the ball-raceway contacts constitutes a mechanism for solid lubrication replenishment. However, the wear rates and their stability must be deliberately set to avoid "lumpiness" induced failures. This self-lubrication principle has allowed bearings to maintain arc-second-alignment accuracy of gimbal-pointing devices while subjected to continuous oscillatory motion for the equivalent of > 5 years in a high vacuum [153,154]. Matching retaining wear rates to the wear rate of the deposited film on the bearing is crucial to keep control of bearing clearances [154]. Excessive transfer of the PTFE composite retainer material functioning as a solid lubricant yielded unacceptable torque increases [155]. A review [156] of dry lubrication in rolling

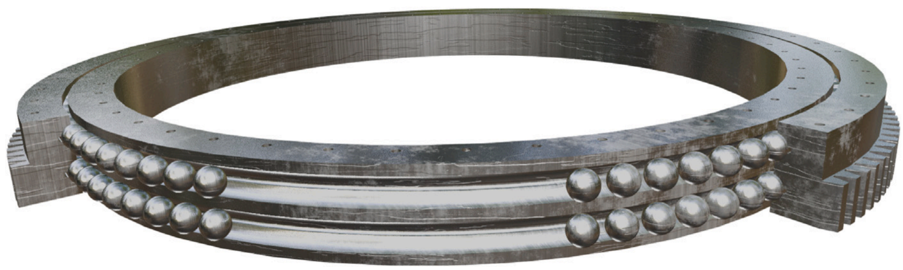

Fig. 16. Double row four-point-contact pitch bearing, common in wind power applications.

element bearings for space applications raised the issue that many early investigations did not account for a controlled initial transfer film formation. Consequently, the repeatability and reliability of these studies could have suffered. This concern was later addressed by studies where proper conditioning to achieve a minimized and delayed onset of torque bumps was implemented [157].

7.3.2. Coatings

The persistent focus on coated bearings in space applications is an inevitable consequence of the specific environmental conditions in this field. Solid lubricants have negligible vapor pressures, precluding vacuum-induced evaporation. They also have a wide operating temperature range and enable accelerated testing without the concern of hydrodynamic effects yielding unrepresentative results. Furthermore, they can be electrically conducting, which can reduce the risk of electrostatic discharge damage. However, good adhesion to the substrate is fundamental, moisture sensitivity can make them difficult to test and store prior to actual mission use, and the debris they generate, as they operate, results in torque noise and spikes which may be unacceptable.

When the oscillation amplitudes are so small that lubrication via film transfer from the ball retainers is not expected to occur, thinner sputtered MoS_2 coatings displayed less propensity to develop torque increases than transfer film lubrication [155]. Specifically, improved bearing performance in antenna pointing mechanisms undergoing small oscillations, was achieved when gold and silver ion-plated films, burnished MoS_2 films, or silver ion-plated plus MoS_2 films were deposited exclusively on the rolling elements [158]. Overall, the Ag+MoS_2 coating provided the lowest recorded starting torques and surface damages. It was also found that the vacuum conditions further improved the system's response as the effects of oxidation, typical for terrestrial conditions, were absent in a vacuum. The expected torque increases in thin film MoS_2 lubricated gimbal bearings could be managed by periodic shifts of the oscillating limits [159]. Furthermore, an analysis of torque signatures and surfaces indicated that there were two major contributions to the frictional torque evolution: a frictional loss and the work done to roll over the debris and the wear scars that had developed. The frictional loss is due to sliding in the rolling contact. The magnitude of this can be estimated using theoretical models. When end-of-travel torque bumps emerge, a thin layer of the MoS_2 coating remains [160]. This observation helped in devising a running-in procedure that would reduce the unnecessary coating thickness and thus extend the number of oscillations that the system can endure without emerging torque peaks.

Although it is an often neglected fact, pre-flight testing of other systems of the spacecraft may require flight bearings to endure operation while in an environment with an atmosphere. An understanding of the longevity of oscillating coated bearings in air, which is generally detrimental to MoS_2 coatings, becomes relevant in enabling pre-launch systems testing while the coating is compromised to a known degree [161]. Another important consideration is that variations in the specifics of the coating (among others, deposition process parameters, coating thickness, substrate preparation) will often result in major differences in coating performance. An up-to-date source on the coating varieties, formulations, and chemical and material properties guidance, specifically for implementations in spacecraft use, can be found in [162].

7.3.3. Bearing blocking phenomena

A specific kind of bearing torque increase was first labeled as "Blocking" in 1981 by Todd in [163]. Blocking has not been quantitatively modeled, although detailed qualitative descriptions of the phenomenon are available [164,165]. Blocking is a torque buildup mechanism that can occur when individual balls are orbiting at different speeds due to disparities in contact angle, ball size, and/or race geometry. These variations result in the balls squeezing or pinching the cage. While adjusting ball-pocket clearances can resolve the problem for unidirectional rotation, this is not necessarily the case for oscillating conditions. Blocking remedies include loosening conformity, reducing race waviness, reducing the contact angle, reducing the friction coefficient, and, ultimately, a cage redesign. An example of blocking and its subsequent remedy can be gained from the high gain antenna gimbal of the Hubble Space Telescope [164]. Life tests uncovered a torque anomaly in the gimbal after repeated full range slew oscillations of ± 97°, with the largest torques occurring at the end-points of travel. In this case, blocking was remedied by a slight decrease of the ball-raceway conformity. Such a solution brings contact pressures closer to the limits imposed by the liftoff loads. Blocking phenomena can also occur in oscillating duplex bearings even at extremely narrow angles of motion [166]. Other remedies for preventing blocking are lowering the preload and a meticulous attention to cleanliness. Blocking proved to be an issue in the prototype space suit AX-5 as well [167]. Bearings in the suit joints suffered from torque increases during underwater human testing. The investigation revealed that the increases in friction torque showed characteristics of blocking and that this underlying phenomenon was aggravated by lubricant washout. Many strategies to mitigate blocking are discussed and tested, such as raceway reliefs to give balls more freedom to adjust to speed variations, different cage designs and materials, and the use of undersized idler polymer balls.

7.3.4. Liquid and grease lubrication

By altering the detailed chemical composition of lubricating oils, the occurrence of torque increases at $x/2b > 1$ can be delayed by a substantially increased number of cycles [168]. Chloroarylalkylsiloxane (CAS) and a linear perfluoropolyether (PFPE) were outlasted 4 times by a polyalpha-olefin (PAO), fortified with the anti-wear additive tricresyl phosphate (TCP). A comparison of ion-sputtered MoS_2 coatings, greases, and oils when the bearings were subjected to composite motions representative of real-life gimbal operations showed that, regardless of coating, self-lubricating PTFE retainers were a necessity in order to achieve a long life of over 40 million gimbal cycles [169]. Meanwhile, bearings with polyimide retainers, silicon nitride balls, or steel balls sputtered with MoS_2 failed at early stages of the testing, regardless of the type of MoS_2 coating employed. Torque with liquid lubricants was found to be lower in magnitude and constant, followed by grease and then MoS_2 film-lubricated bearings, with the latter two being able to produce longer bearing life. The high torque of prematurely failing bearings was primarily related to MoS_2 or retainer film deposits. A way of addressing this problem is to use solid and liquid lubricant-impregnated retainers [170]. Experiments at amplitudes of ± 0.5, 5, and 20° revealed that no one lubricant technique was ideal for all angles tested. The largest of which is still not enough for the balls to perform the 90° rotation needed to achieve effective material transfer from the cage to the raceway. Interestingly, several balls were reported to have more than one pair of corresponding contact zone markings, which indicated potential pathways for cage material transfer to the raceway even at oscillation amplitudes where this is not initially expected. Bearing torques were reported to be many times higher than those expected for continuously rotating bearings with 5-fold increases at the point of reversal being common. Torque increases could not be explained solely as the result of changes in friction coefficient, preload, or the number of balls in contact. Instead, it is most likely the result of a change in the ball-raceway conformity due to a buildup of compacted debris. Comparing lubrication based on grease (multiply alkylated cyclopentane (MAC) sodium soap grease with undisclosed composition) to a solid media (MoS_2 PVD film with a self-lubricating composite PTFE/glass fiber MoS_2 loaded cage) revealed that both types of lubrication could keep torque below the predefined threshold at the completion of test runs [171]. However, other important differences between the two means of lubrication existed. Lower bearing noise was found for the grease-lubricated bearings while the coated bearings showed comparatively lower friction torques at low temperatures. Interestingly, microscopic observation of the grease-lubricated surfaces revealed a deposited tribofilm, located in the contact tracks in the form of a 1 μm thick layer. Liquid and solid lubrication approaches were reported in [172], where sputtered MoS_2 films (raceway and balls) were compared with perfluoropolyether (PFPE, Fomblin Z25). The bearings were subjected to accelerated scan motion profiles of 8.4° strokes, which were overlaid with ± 30° regeneration cycles, as the Protection Runs described before. Liquid lubrication provided less torque noise. Oil lubrication for oscillating bearings at elevated temperatures (75 to 122 °C) [173] showed that PFPE (Fomblin Z25) outperformed the MACs at the higher temperatures by over an order of magnitude in terms of cycles. However, at lower temperatures, this was completely reversed with the MACs now outperforming the PFPE by a similar margin. At lower temperatures the failure mechanism was lubricant decomposition in the contacts, whereas at higher temperatures, it appeared to be lubricant evaporation [173]. This trend at lower temperatures was also reported in [174,175].

7.3.5. Field and laboratory data correlations

In 1993, after three years in orbit, it was found by telemetry that the servomotor in one of the three Fine Guidance Sensors (FGS) aboard the Hubble Space Telescope displayed stall-magnitude torque anomalies. With the first scheduled servicing mission less than a year away, an investigation of these telemetry data was undertaken [176]. Accelerated bearing life tests replicated the anomaly and showed that a small angle (0.75°) pivoting motion referred to as Coarse Track, was responsible for the sudden bearing degradation. The investigation resulted in changes of the operation modes of the bearings referred to as Fine Lock and consisting of smaller angle oscillations in the order of magnitude of 0.003°. This Fine Locking would result in a motion that was contained in the elastic range of the bearing (before steady rolling, so-called Dahl's regime) and produced no noticeable change in torque signature of the test bearings even after hundreds of millions of Fine Lock cycles were performed. Although the results indicated that it would be possible to skip the second servicing mission as well, a new and distinct phenomenon in the torque signatures of the telescope was later observed. The authors referred to this new signature as a "reversal bump". At the moment of publication, this new torque signature was not yet fully understood nor had it been replicated on earth. The worsening of this anomaly would prompt the FGS 1 replacement during the second servicing mission. After 7 years in use, the unit (a duplex set of bearings lubricated with Bray 815Z oil and Tricresyl Phosphate (TCP) treated) underwent a post-flight examination [177]. The lubricant had degraded, and bearings appeared dry, with a tar-like degradation product that had adhered to the races and toroid ball separators (Fig. 17). No evidence was found of the TCP anti-wear additive. Ionic fluorine was detected on the ball surfaces, a further indication of lubricant breakdown. Significant contamination in the form of metallic particles, skin, and fibers was found in the bearings, which had not been detected at the pre-flight torque testing. Ultimately, the reversal bumps were never reproduced, however, it is believed that a complex interplay between oil degradation, particulate debris, and blocking was responsible for the anomaly. Several recommendations emerged as the result of this investigation: (i) race conformity changes to reduce ball spin and blocking risk; (ii) verification of adequate anti-wear additive coatings; (iii) stricter contamination controls and (iv) the addition of more lubricant to the bearings. The health and life expectancy of a gimbal bearing can be estimated using telemetry torque data and lubricant evaporation rate, as outlined in [178] for an Earth-observing satellite that had exceeded its designed life, which was experimentally estimated by [179] 2 decades prior.

8. Discussion

8.1. Researched space

A mapping of the research data available is presented in a Research Summary Table within the electronic supplementary information (ESI). This table compiles all experimental oscillating bearing publications in terms of bearing type, $x/2b$ ratio, additional track overlap descriptions, maximum Hertzian contact pressure, oscillating frequency, and lubrication. It also includes bearing designations and sizes, oscillation cycle ranges, test temperatures, wear analysis methods, and whether or not the bearing torque was monitored. The $x/2b$ ratio was, in the vast majority of cases, estimated by the authors because it was not explicitly reported in the corresponding work. To a lesser extent, the same is true for contact pressures as well. A reminder that the contact conditions are not usually clarified fully in the literature.

Angular contact ball bearings are the most common bearing type tested in oscillatory motions, representing over 40% of the entries. Twice as many as thrust ball bearings, for which there are actual testing standards. This is in part due to the fact that 4-point contact wind power pitch bearings can be economically emulated with angular contact ball bearings, and also due to the large majority of aerospace-oriented research, where angular contact bearings, usually thin section designs with much lower contact conformities and loads, are used extensively in pointing mechanisms and gimbals. Angular contact bearings offer several advantages in positioning accuracy, load-carrying capacity, pre-loading adjustability, and cost, making them the bearings of choice for oscillating applications. It is worth noting that slip conditions and contact pressure distributions are bearing type dependent and these will influence the resulting damage and its progression.

High frequency tests (50 Hz and above) have only been conducted at $x/2b < 1$ under conditions resembling vibrations. Although this is a reasonable reflection of general applications, it also represents a gap in knowledge relevant, for example, in high-speed motion control. Frequency-dependent starvation effects have been reported [37]. This occurs when there is insufficient time for the base oil to re-flow into the contact track before the next oscillation occurs. The extent to which this phenomenon is aggravated under conditions of higher frequencies is unknown.

Over 60% of the Research Summary Table entries use grease as a lubricant. The actual composition of the grease is frequently omitted, resulting in a knowledge gap in grease formulation practices. It remains a persistent challenge to find data for the viscosity and surface tension of the base oils used to manufacture greases. These parameters along with the contact angles of the oil with the lubricated surfaces govern the rate of contact replenishment [180] and oil flow in the narrow contact gap (Washburn's law [181]). The wettability of lubricated surfaces may also change due to thickener deposits in the rolling track [182], grease aging, surface topography, and chemistry evolution [183].

Fig. 17. Thin cross-section angular contact bearing with polytetrafluorethylene (PTFE) toroidal ball separators, commonly used in space applications as pre-loaded pairs.

Interestingly, using a low-viscosity liquid lubricant, as opposed to grease, has now frequently been shown to be superior in preventing wear and extending bearing life. Despite this evidence, the use of grease for lubrication purposes in oscillating bearings continues. This is probably a consequence of practical considerations such as reduced seal requirements, the possibility of lifetime bearing lubrication, and avoiding the need for a bulky oil lubrication supply system.

Coatings and transfer film lubrication, as alternatives to grease or oil lubrication, have been shown to be promising solutions for oscillating bearings insofar as the bearing is expected to operate under low loads (<500 MPa) or in environmental conditions of extremely low temperatures. The environmental and cost considerations within space applications, where coatings have been extensively implemented, are not necessarily relatable to other fields and this is reflected by the fact that nearly all the research pertaining to coatings is explicitly addressing space applications.

8.2. Testing procedures

Reports on investigations with oscillating bearings suffer at times from inadequately detailing all the experimental procedures utilized. These details include the mundane but essential reporting on (i) the measurement and evaluation of wear; (ii) the possible presence and, if relevant, quantification of parasitic loading; (iii) the procedures of lubricant application and cleaning. Reporting of these items together with detailed descriptions of the motion profiles, would guarantee the required scientific transparency necessary for meaningful reproducibility checks. It would also allow a more efficient implementation from laboratory-based findings to real-life applications.

8.2.1. Wear measurement and evaluation

There are different approaches for wear measurement and evaluation of Standstill Marks and False Brinelling. In False Brinelling tests (ASTM D4170 or SNR-FEB2, see Sections 4.1 and 4.2), the wear is usually abrasive and of such a magnitude that the determination of the mass loss by weighing alone suffices. Abrasive wear under steady state conditions results from the accumulation of hard particles in the contact track. By contrast, testing for and quantifying Standstill Marks is not as straightforward and can be difficult and costly [19]. The weight loss in the bearing could be less than 0.1 mg even with large marks. Acoustic measurements are useful but only in the case of advanced damage and are relatively costly. The size of the ripples can be documented using optical or stylus instruments. The presence of oxide particles inflicts significant uncertainties on the accurate quantification of wear volume and maximum wear depth [41]. Generally, these particles have a larger volume than the base material, they grow upwards and they adhere to surfaces. In addition, the measurement by white light interferometry is so complex and time-consuming that only those markings that, after a visual inspection, appear to be suffering the most damage, are further analyzed by this technique. It is evident that such an approach (i.e.based on the analysis of what are to be considered outliers) cannot be subjected to reliable statistical analysis.

Measuring the size of the markings and the modified area is not suited for tests of short duration. The dimensions of the markings which can be visually evaluated are mainly determined by the $x/2b$ ratio and the size of the Hertzian contact area. However, the size of markings or altered areas bears very little relation to the wear mechanisms and the type of damage. For example, mild smoothing is to be evaluated positively, whereas seizures, cracks, and disruptions are evaluated negatively as they are considered to be more critical when considering service life [73]. Reporting geometric parameters should not be the sole outcome of tests. Further information on e.g. color and topographical conditions provided by microscopy should also be taken into account [39,140,141]. Importantly, the most reliable damage evaluation method is based on a comprehensive comparison with results obtained for a reference system [73].

Whereas wear quantification takes place after an experimental run, continuous torque monitoring provides *in operando* information on the damage evolution. Depending on the application, such measurements may include larger amplitudes, than those used in the test conditions, to evaluate the torque needed for the rolling elements to exit the worn contact track.

8.2.2. Parasitic loading

In an ideal configuration, the axial loading of a pair of bearings makes the rolling elements support the entirety of the applied force. This would be the case if one of the bearings is fitted in a locating manner and the other one is not locating, meaning it is free to move axially under the applied load until it reaches a locating shoulder. In reality, there is friction on the outside rings of both bearings and part of the axial load is going to be supported not by the rolling elements, as intended, but by the bearing housings, resulting in Parasitic Loading. This can render the expected contact pressures inaccurate: the actual contact pressures will be lower than those anticipated. The extent of this reduction is unknown and difficult to estimate. The axial load can be verified by checking the expected contact size against the raceway wear patterns [138]. A precaution that should be encouraged, particularly with test setups that were not designed with this aspect of bearing fits in mind.

8.2.3. Cleaning procedures

The impact of cleaning procedures is not fully addressed in the literature. Nearly 70% of publications describing experiments on oscillating bearings do not provide any information on pre-test surface preparations. In the remaining 30%, there is often only a partial description of the cleaning process and solvents used. Since oscillating bearings operate largely under conditions of boundary lubrication, the degree of surface cleanliness will affect the surface interactions of the lubricant which, ultimately, will impact on bearing life. It has been shown that the solvent used during bearing preparation impacts oscillating bearing life under conditions encountered in space exploration [174,175]. Thus, the cleaning procedure should either be a standard one with a reference or, in the case of specific modifications, fully described.

8.2.4. Grease application

Similar considerations for cleaning apply to greasing the bearings. Very few publications state how much grease is added to the bearing and how it is distributed. In addition, there is often no information on whether a grease distribution run was carried out and what the specific conditions were of the grease distribution run. Since False Brinelling damage or Standstill Marks usually occur due to a local lack of lubrication, this information is essential for the scientific evaluation of the work.

8.2.5. Motion profiles

The influence that different types of motion profiles have on bearing damages has not been specifically reported on. That is to say, there is an unknown impact on the bearing life of sinusoidal, trapezoidal, triangular, s-curves, or other bearing motion profiles. Differences are expected to arise due to variations in acceleration which is much higher at pre-rolling breakaway for trapezoidal and triangular profiles than for sinusoidal profiles at identical frequencies. Entrainment speeds will also be different at the same frequency and amplitude. These differences should be negligible in most cases since they have not warranted targeted investigations yet. However, applications where particularly extreme motion profiles are the norm should certainly consider this as a potential source of divergence of operating parameters from those found in the literature.

8.3. Application specifics

Within the published primary literature on oscillating bearings, the interpretations, generalizations, and comparisons are severely hindered by several critical factors. There is the often re-occurring lack of a complete description of the tribological systems involved in the reported studies. Contact conditions ($x/2b$), bearing design specifics (osculation and other relevant parameters), and lubricant composition (base oil, thickener, and additive package) are rarely fully disclosed. Transferring experimental results to real-life applications is thus subject to a great degree of uncertainty. Furthermore, boundary application cases (highly loaded large pitch bearings of wind turbines, lightly loaded low conformity coated bearings of space technology applications), exhibit a distinct failure evolution and have unique terminal life criteria. Nevertheless, once a given novel tribological configuration or bearing application is characterized in terms of its similarities with existing bearings and descriptions in the literature, it will then be possible to at least provide initial estimates of life expectancies and to proceed with a screening of suitable candidate compounds or formulations for lubricating purposes. In general, testing will be necessary to obtain a robust life prediction and is likely to be heavily application-specific in nature.

Importantly, sharing knowledge from application-specific testing will benefit bearing reliability in a vast range of applications. The principle of Protection Runs, which was initially developed for wind power pitch bearings, can also prolong bearing service life in applications such as cranes and telescopes. In these applications Protection Runs are enacted daily with the largest possible amplitude, but, the latest research suggests that more Protection Runs with smaller amplitudes are better and less disruptive to the regular operation of the machine.

9. Conclusions and outlook

Oscillating bearings undergo distinct wear mechanisms that are particular to their operating conditions. Wear hinders service life estimations which rely on the assumption of surface fatigue as a failure mode. Wear mechanisms of oscillating bearings depend on the specific conditions endured by the ball-raceway contacts. The oscillation amplitude and its relation to the contact size are of particular importance in determining the resulting wear mechanisms. For example, the wear phenomena provoked by micro-vibrations differ significantly from those induced with relatively large oscillating movements, such as routinely used in laboratory tests according to ASTM D4170 or SNR-FEB2. These macroscopic oscillating movements cause “real” rolling processes to dominate. This is in complete contrast to the effects of micro-movements that are particularly critical when encountered by nominally stationary bearings. Hence, using “False Brinelling” as a general term, irrespective of the prevailing contact conditions, is too ambiguous. By using the parameter $x/2b$, the conditions in different experimental set-ups across different bearing sizes can be unambiguously characterized in terms of what the tribologically relevant parameters are that the bearing contacts are actually subjected to and what the possible wear mechanisms are that can be expected. Seemingly contradictory results can often be attributed to dissimilar contact conditions and wear characteristics, variously labeled with umbrella terms like Fretting, Fretting Corrosion, or False Brinelling. Grouping the contact conditions and wear characteristics with $x/2b$ can resolve what initially appears as contradictory results. The Research Summary Table (supplementary material) lists all the relevant published sources and classifies them in terms of this non-dimensional parameter and all other relevant operational parameters, thus providing a concise and unified perspective on the research field.

In addition to the description of the contact kinematic processes, the review categorizes and summarizes numerous experimental results, showing the influence of individual factors such as the load spectrum and the lubrication condition. An emerging consensus appears to be that no separating lubricating film can build up during oscillating motions with relatively small angles ($x/2b < 1$) or vibrations. Instead, the oscillating motion actively displaces the lubricant from the contact point. This quickly leads to localized starvation, boundary friction and damage because of tribochemical corrosion. In addition, the energy input is so low that chemically triggered additives cannot react, with the usual lubricant additive sacrificial chemistry often failing to protect the surfaces. If the oscillation amplitude is increased, re-lubrication can occur and antiwear additives become increasingly effective. At values larger than $x/2b \approx 18$, accelerated raceway wear is no longer reported. A clear distinction in the functional role of the lubricant is observed.

The general insistence on using grease as a lubricant in oscillating bearings is a testament to the substantial advantages of grease lubrication. The consensus within the research field is clear: low viscosity base oils should be used together with low concentrations of thickeners. Thus the grease increasingly attains properties similar to oil. The latter is by itself superior in terms of wear protection. No other grease rheology parameters have been correlated as strongly to oscillating bearing wear performance. The fact that grease bleeding rate is measured in a way that does not reflect the actual workings of grease in an oscillating bearing (i.e. releasing oil within the crevices that surround the contacts) strongly suggests that future developments may most likely rely on achieving higher “effective” bleed rates (in-situ or on-demand) by means of new grease formulations and/or bearing designs. Future developments should also include life cycle assessments of grease-bearing systems to minimize their carbon footprint.

Analytical and numerical calculations are invaluable to fully understand the kinematic processes occurring at the contact area between the rolling element and the raceway. Energy dissipation, due to frictional work, spatially correlates with material removal. Nevertheless, all models rely on a singular, time and location-invariant value of the friction coefficient. The ongoing chemical processes have hitherto been completely ignored. It will require a great deal of experimental trials to develop future lubrication strategies for practical applications. Furthermore, different oscillating bearing applications display drastic differences in failure criteria, damage mechanisms, application environment, and overall bearing requirements. Wind power and space are extreme examples of applications of research areas with a substantial amount of experimental literature, yet lubrication strategies, contact pressures, and design requirements are so different, that bearing deterioration, failures, and even damage mechanisms can be widely divergent. This divergence makes the creation of widely representative standard testing methods a very challenging proposition. This is particularly the case for methods aiming to emulate larger bearings through scaling. Nevertheless, more sophisticated standardized procedures for scaling and testing may prove beneficial, particularly if bearing tests continue to be the primary means for investigation. Such procedures would ideally rely on statistical backing of application-specific field data correlations, in such a way that a range of conditions of load, amplitude, and frequency can be represented within the framework of the standard. It would, in all likelihood, involve purpose-built standard-issued bearing samples, upon which a baseline for widespread lubricant development can be streamlined.

Our understanding of the wear inflicted on bearings by oscillations has substantially progressed since it was first detected. In the coming decades, progress in several key technological sectors will continue to rely on further improvements in the service life of oscillating bearings. This reliance will result in the exploration of numerous development pathways. One of these could be bearing designs that exploit cage replenishment effects, in conjunction with optimized kinematics for oscillating motions, which reduce differential slip while also facilitating replenishment of base oil. Many developments will also take place within the realm of lubrication technology. Examples of these could be some of the following technologies: (i) surface-active additives that do not rely on a running-in process to protect the surface; (ii) development of diverse vacuum-compatible lubricants with ionic liquid chemistry; (iii) low-wearing hard coatings; (iv) synergistic effects of coatings and "wet" lubricants, and (v) greases that provide extremely large effective bleeding rates in-situ while maintaining their overall consistency.

CRediT authorship contribution statement

Román de la Presilla: Conceptualization, Methodology, Investigation, Data curation, Writing – original draft, Writing – review & editing, Visualization. **Sebastian Wandel:** Conceptualization, Methodology, Investigation, Data curation, Writing – original draft, Writing – review & editing, Visualization. **Matthias Stammler:** Conceptualization, Methodology, Investigation, Data curation, Writing – original draft, Writing – review & editing, Visualization. **Markus Grebe:** Conceptualization, Methodology, Investigation, Data curation, Writing – original draft, Writing – review & editing, Visualization. **Gerhard Poll:** Writing – review & editing. **Sergei Glavatskih:** Supervision, Conceptualization, Writing – review & editing.

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.

Data availability

Data will be made available on request.

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Appendix A. Supplementary data

Supplementary material related to this article can be found online at https://doi.org/10.1016/j.triboint.2023.108805.

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