Acta Montanistica Slovaca
ISSN 1335-1788
actamont.tuke.sk

Maintenance management of large-size rolling bearings in heavy-duty machinery

Marek KRYNKE[1], Krzysztof KNOP[2] and Magdalena MAZUR[3]**

Authors' affiliations and addresses:
[1] Department of Production Engineering and Safety, Faculty of Management, Czestochowa University of Technology, Al. Armii Krajowej 19b, 42-218 Czestochowa, Poland
e-mail: marek.krynke@pcz.pl

[2] Department of Production Engineering and Safety, Faculty of Management, Czestochowa University of Technology, Al. Armii Krajowej 19b, 42-218 Czestochowa, Poland
e-mail: krzysztof.knop@pcz.pl

[3] Department of Production Engineering and Safety, Faculty of Management, Czestochowa University of Technology, Al. Armii Krajowej 19b, 42-218 Czestochowa, Poland
e-mail: magdalena.mazur@pcz.pl

***Correspondence:**
Magdalena Mazur, Department of Production Engineering and Safety, Faculty of Management, Czestochowa University of Technology, Al. Armii Krajowej 19b, 42-218 Czestochowa, Poland
tel.: +48 34 3250 367
e-mail: magdalena.mazur@pcz.pl

How to cite this article:
Krynke, M., Knop, K. and Mazur, M. (2022). Maintenance management of large-size rolling bearings in heavy-duty machinery. *Acta Montanistica Slovaca,* Volume 27 (2), 327-341.

DOI:
https://doi.org/10.46544/AMS.v27i2.04

Abstract
Slewing bearings are one of the most important elements in the vast majority of large-size machines.
They are widely used in the mining industry: tunnel cutters, bucket excavators and many other devices. In a Bucket-wheel excavator, continuous rotation of the body is most advantageous due to the technique of digging the input or coal. The rotational movement of the machine is then the basic cutting movement, and the delivery movement, carried out by driving the machine, is only an auxiliary movement. A similar kinematics occurs in tunnel cutters. Therefore, these bearings have played such a significant role and have been the subject of extensive research and continuous improvement over the years.
High demands are placed on them in terms of load capacity, friction, accuracy, durability and reliability. It happens, however, that despite careful design and manufacture, the bearings do not achieve the required durability. Failures usually result in economic losses due to loss of production, damage to adjacent parts and repair costs. Premature bearing failure can occur for a variety of reasons. Each failure leaves its own special mark on the bearing. Consequently, by examining the damaged bearing, it is in most cases possible to find the root cause and define corrective actions, thus preventing further failures.
This publication aims to provide basic knowledge about the factors determining the load capacity and durability of large-size slewing ring bearings and the analysis of their damage. The result of the considerations is finding the sources of errors in determining the load-bearing capacity characteristics of roller slewing bearings. For this purpose, the ISHIKAWA and FMEA methods were used and the risk level for errors was determined. Moreover, the article presents some forms of damage to raceways of slewing bearings and indicates their causes. Changes in the so-called angle of action of the rolling elements in the ball bearing due to the transferred loads. The influence of changes in this angle on the geometry of the contact zone of the rolling elements and raceways was investigated. It has been shown that the contact angles of some rolling elements increase significantly. This can damage the raceway by chipping or rolling the edge of the bearing ring. With the knowledge presented in this article, it is possible to evaluate various emergency situations and start their proper analysis.

Keywords
maintenance management, FMEA, slewing bearing, bearing damage.

Introduction

The swivel mechanisms in heavy-duty work machines are usually designed using large-dimension roller bearings. These are machine components that transfer the entire load from one machine element to another, e.g. the body to the chassis. Their special features, which they owe to their high load-carrying capacity with a relatively compact design and relatively small dimensions, make them used not only in classical machines and equipment such as excavators, all kinds of cranes and other construction machines, military vehicles, but they are also widely used in the mining industry: tunnel heading machines, excavators, stackers and many other machines (Mazanek & Kania, 2012; Kovanič et al. 2020; Pamukcu, 2015; Pivarčiová, 2019).

The main feature of these bearings, which distinguishes them from standard bearings, apart from their large diameters, is the design of their bearing rings. These rings are shaped in such a way that they can be fixed by means of bolts directly to two supporting parts of the device (machine body) – one rotating, the other fixed. Very often, one of the bearing rings is also a toothed rim, hence the name "slewing ring bearings". The toothed wheel can be external or internal (Fig. 1) depending on the location of the mating drive gear.

The construction of slewing rings is very diverse. Due to the design features, numerous classification criteria can be distinguished (Smolnicki, 2013):

- the shape of the toothed rim – external, internal,
- treadmill construction – soft, hard, monolithic, wire,
- type of rolling elements – ball, roller, pinion and roller,
- number of tracks in a row – single-row, multi-row,
- number of races cooperating with the rolling element – two-track, four-track.

The basic criterion for selecting slewing rings for machines is their static load capacity. The load capacity of slewing bearings usually limits the maximum value of external loads of the designed device, and its correct determination is an important part of the working machine's calculations. Slewing bearings are usually selected to the extreme, that is, their operation is at the breaking point. This requires careful and accurate calculations of their operational parameters, the most important of which is the static load capacity, and more and more often supplemented with additional criteria from the assembly, durability to the determination of resistance to motion. (Aguirrebeitia, Abasolo, Avilés, & Fernández de Bustos, 2012; Kania, Krynke, & Mazanek, 2012). It should be borne in mind that on many machines, damage or destruction of a slewing bearing can cause catastrophic accidents. Furthermore, it requires long periods of machine downtime and results in high repair costs (Jin, Chen, Wang, Han, & Chen, 2021). Therefore, slewing ring bearings in the vast majority of machines can be classified as their most responsible components. High demands are also placed on the load capacity and reliability of these bearings. For this reason, rolling bearings have been the subject of intensive research for many years, and over time, bearing technology has developed into a separate branch of knowledge.

Among the benefits of these tests is the ability to calculate bearing life with a high degree of accuracy, which enables the bearing life to be matched to the machine's operating life (Vicen, Bokůvka, Nikolić, & Brönček, 2020). Due to the specific nature of the operation of slewing ring bearings – slow rotating or oscillating movements, gradual wear of the raceway (most often by peeling), or occasional pitting chipping, do not affect the smoothness and accuracy of movement for a long time. Therefore, when calculating and designing slewing bearings, the concept of service life was introduced, defining the operating time of the bearing, after which the wear becomes so high that it causes a sharp increase in the moment of resistance to movement of the bearing, at which further operation becomes impossible (El Laithy et al., 2019).

Unfortunately, there are occasions when a bearing does not achieve design life. This is due to various factors: greater than expected load, insufficient or inadequate lubrication, poor handling, ineffective sealing, or incorrect assembly. Each of these causes generates a specific, typical type of damage and reveals itself in the form of a characteristic "damage pattern". Consequently, in most cases, it is possible to see the cause of the damage by examining the damaged bearing and taking measures to prevent it from failing again (Biały & Fries, 2019; Biały, 2022; Yakout, Nassef, & Backar, 2019).

This article analyzes the causes of typical failures of slewing rings. In addition, considerations were made regarding the formation of errors in determining the load capacity characteristics for such bearings, and guidelines for designers of such devices were given.

Literature review

A slewing bearing that is properly designed, manufactured, installed and operated will eventually be damaged by pitting or fretting. The phenomenon of pitting is fatigue wear of the surfaces of rolling elements or raceways as a result of cyclic loading with contact stress and manifests itself by the formation of cracks on the surface or at a small depth under the surface. The propagation of these cracks leads to the chipping of fragments of material in the form of scales from the surface. In turn, the phenomenon of fretting consists in the formation of micro-cracks in the area of contact of the rolling element with the raceway, as a result of which cavities are formed (Caesarendra, Kosasih, Tieu, Moodie, & Choi, 2013; Turis, Beňo, & Biały, 2018).

During machine operation, knowledge of the forces transmitted by individual rolling parts of a bearing is the basis for assessing the bearing's ability to perform its function. Pallini (Pallini, 1979) lists 5 criteria for the evaluation of slewing rings:

1. durability,
2. stopping capacity,
3. properties of the hardened layer of the raceway and the core,
4. strength of other elements of the rotation mechanism (screw connections, girth gear),
5. lubrication conditions.

Rolling elements are rarely subject to surface fatigue first. They are made of bearing steel, which is a material with much better properties in terms of fatigue wear of the surface than the material from which the rings of slewing bearings are often made. Moreover, due to the more favorable distribution of the variable load on their surface and the smaller number of contacts in a given area (contact in a given area of the ball is random), they are more resistant to wear (Wang, Chen, Wang, & Zhang, 2019).

The basic usable quantity that characterizes the suitability of a bearing to fulfill specific tasks is its load capacity. In the bearing technology, it commonly distinguishes between static and dynamic load capacity, determined according to standards (ISO, 2006, 2007). Regarding plain bearings (ISO, 2017), the nominal static load rating is defined by the external static load, which corresponds to the permissible contact stress in the center of the contact of the most loaded rolling element or the total permissible relative (depending on the diameter of the rolling element) deformation of the rolling element and the bearing raceway. On the other hand, the nominal dynamic load capacity is defined as the external load of a rotating bearing, which will allow obtaining its nominal life up to 1 million turnovers. It should be noted that when determining the static and dynamic load capacity of ordinary bearings, the external loads are strictly assumed as radial for radial bearings and axial for thrust bearings. In the case of slewing ring bearings (as opposed to plain bearings), due to the static nature of work, the static load capacity is of the greatest practical importance, which is widely described, among others, in the papers: (Kania, 2012; Kania & Krynke, 2013; Li, Wang, & Li, 2022; Mazanek, 2005; Ojo, & Obasha, 2022; Abramov et al., 2015). It should be noted that the static load capacity of plain bearings cannot be directly compared with the static load capacity of slewing bearings because the strength assumptions and the nature of taking over the external load are different for both types of bearings. Slewing bearings used in cranes, bucket excavators, and many other heavy-duty machines tend to operate under a static load. Their work does not end, as in the case of conventional bearings, when the first symptoms of pitting appear. Slewing bearings can work for a longer time in conditions of progression of pitting and fretting damage (Glodež, Potočnik, & Flašker, 2012; Siwiec & Pacana, 2021).

The durability of typical rolling bearings is defined as the time of their operation until the first microcracks appear on the surface of the raceway or the rolling element, or actually the first chipping or cavity. In the case of large-size bearings, the service life cannot be treated as an absolute value due to the variety of loads during operation of the bearing. It can be and is treated as an element of the definition and a comparative value. All slewing bearings do not achieve theoretical durability, but a significant part of them exceeds this durability, in some cases reaching values several times higher. The durability criterion does not have to be met by bearings operating at low speed or performing oscillating movements. In these cases, the circumferential speeds are low, so the raceway wear and pitting do not have a large impact on the stability and precision of movement, which is important in ordinary rolling bearings (Raadnui & Kleesuwan, 2011).

In the technical documentation of slewing ring bearings, the static load capacity is presented in the form of the so-called static load capacity characteristic. It contains the relation of the mutual dependencies of the transferred overturning moment M on the axial force Q for the assumed value of the radial force H (Fig. 1). These characteristics normalize the ranges of permissible loads that a given bearing is able to safely carry, assuming the durability specified by the manufacturer. Thus, durability is an indispensable parameter accompanying the static load-bearing capacity of slewing ring bearings. Due to the operational variety, including the randomness of the external load value, precise determination of the bearing life becomes impossible. Therefore, an approximate definition of the life of slewing bearings is used, related to the so-called equivalent number of full revolutions of the bearing operating under the maximum allowable load (Gibczyńska & Pytko, 1999).

In determining the durability of large-size bearings and the corresponding dynamic load capacity, two approaches can be distinguished. One represented by the manufacturer of such bearings – Rothe Erde (Rothe Erde, 2018), the other by SKF (SKF, 2019). The Rothe Erde company provides the so-called service life of the bearing, defining it as the service life of the bearing until the raceway wear, which no longer allows for its safe functioning. For the estimated useful life defined in this way, which corresponds to 30,000 rotation speed, the catalogs show load capacity diagrams for bearings operating at full load consisting of an axial force and a tipping moment (curve 2 in Fig. 2). Curve 1 (Fig. 2) defines the nominal static load capacity characteristics of a new slewing ring bearing. These curves can be used to determine the service life of the bearing, with load values other than the catalog values, and to select a bearing with a given load spectrum. They cannot be used when radial forces are applied and when the bearing operates at high revolutions and high precision of movement is required (Mazanek & Kania, 2012).

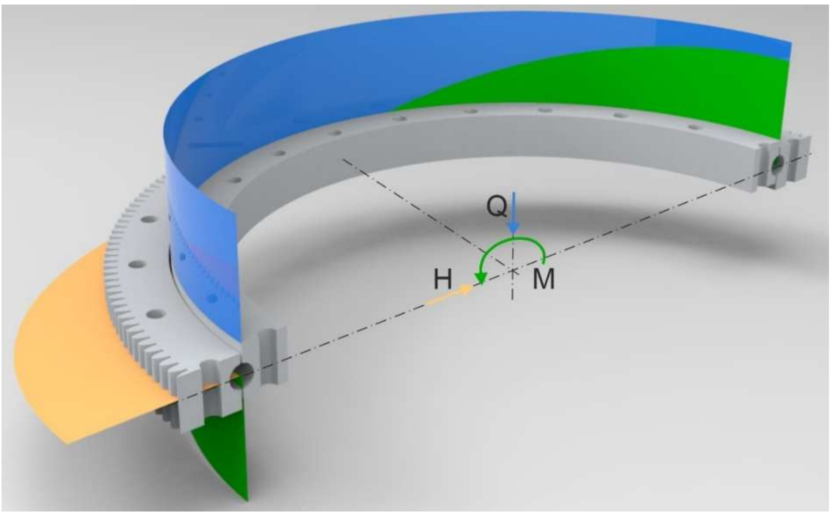

Fig. 1. External load on a slewing ring bearing

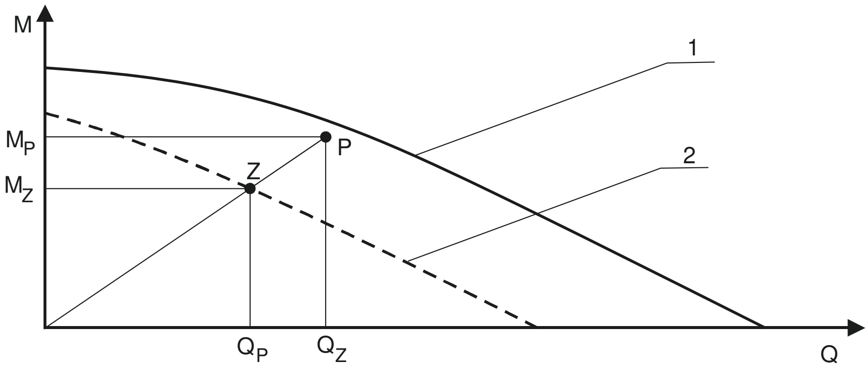

Fig. 2. Illustrative characteristics of the bearing static load capacity in the nominal (line 1) and after-service (line 2) condition

As a result of wear, the course of the load-bearing characteristics has changed. The obtained relationship was called the characteristic of the bearing static load capacity in the after-service condition. It is mainly used to determine the number of equivalent rotations n_{ek} (related to the bearing life) for loads smaller than the limit loads contained in the working field (area under the curve no. 1 – Fig. 2). The number of n_{ek} is determined from the formula 1:

$$n_{ek} = (f)^m \cdot 30000 \tag{1}$$

where the exponent m takes the values: $m = 3$ for ball bearings and $m = 10/3$ for roller bearings (Kania, 2012). The parameter f is taken as the smaller of the quotient values defined below:

$$f = \frac{M_Z}{M_P} \; or \; f = \frac{Q_Z}{Q_P} \tag{2}$$

The essence of the quantities included in the formulas (2) is explained in Fig. 2. For example, with the assumed load of the slewing bearing in accordance with the marked work point $P(Q_P, M_P)$, the point $Z(Q_Z, M_Z)$ is located at the intersection of the line formed by the operating point and the origin of the system coordinates with the characteristic of the static load capacity in the after-service state. If different load variants are anticipated (several operating points in the number of i) and it is possible to determine the percentage share of time T_i of operation of individual loads during operation – the number of equivalent revolutions is determined in accordance with the formula (3):

$$n_{ek} = \frac{100}{\Sigma_i \left(\frac{T}{n_{ek}}\right)_i} \tag{3}$$

According to the suggestion of the company, the presented post-service characteristics of the static load capacity of the bearings cannot be used to predict the range of cyclic operation of the bearings subjected to high values of radial forces and rotational speeds greater than provided for. However, they take into account the angular path (smaller than the full angle) that the bearing ring traverses when the direction of rotation is changed, which can be reduced to the total number of equivalent turns n_{ek} (Rothe Erde, 2018).

For SKF (SKF, 2019) life is defined as the equivalent number of full load revolutions that the bearing can make before the first signs of fatigue appear, i.e. pitting or peeling of the raceways or the rolling elements. As in the case of commonly used bearings, the nominal service life is defined as the service life achieved by 90% of the large batch of bearings tested under the same conditions. The average life is 3-4 times greater than the nominal one and it is demonstrated by 50% of the bearings.

The expected bearing life is determined depending on the nominal service life, the number of device cycles, the bearing rotation angle, and the load spectrum parameter. The load rating corresponds to a life of 10^6 full revolutions of the bearing under full load of axial force and tipping torque. It is also possible to take into account the radial force. However, if the expected number of bearing operating cycles is lower, or the bearing oscillates with an angle less than 360º, or the spectrum parameter is different from 1, the equivalent number of load cycle changes should be determined. For this purpose, appropriate nomograms have been included in the catalogs to determine the equivalent number of cycles for devices operating in oscillating motion, such as cranes and excavators. Having determined the equivalent number of cycles, the effective life of the bearing can be determined from the appropriate diagrams. If the bearing is subject to a three-dimensional system of forces or the structure does not meet the manufacturer's requirements, i.e. when the distribution of forces in the bearing is different from the classical cosine distribution, the effective durability of the pre-selected bearing should be determined by the precise method that determines the load on each rolling element and then on the raceway.

Kunc (Kunc & Prebil, 2003), Smolnicki and Stańco (Smolnicki, 2013; Stańco, 2008), among others, dealt with the issues of deformation wear of the raceway in slewing rings, and included the results of numerous experiments and calculations obtained by numerical modeling in their works.

Material and Methods

Slewing bearings usually carry both the axial force Q, the tipping moment M and the radial load H (Fig. 1). They are sets of working machines with a high cost of production as well as replacement or renovation, they are often a critical element of the device. Bearing failure can cause large losses due to unplanned production downtime. Moreover, its replacement, due to its large dimensions, may take several months.

The basic quantity determining the operating conditions of slewing rings is their static load capacity determined by the appropriate characteristic built in the coordinate system $M = f(Q)$. Therefore, the determination of the load capacity of these bearings requires the use of precise calculation methods. The load capacity characteristics can be obtained by analytical calculations assuming the non-deformability of the bearing rings (Mazanek, 2005). However, in many cases, when calculating the load capacity of slewing bearings, it is necessary to take into account the deformation of the bearing rings and the machine components to which they are bolted. The bearing capacity is particularly strongly affected by deformations of the supporting structures of the carrier, causing deflections of the bearing rings screwed with an appropriate initial clamp to the seat of the supporting subassembly. The deflections of the bearing rings cause significant changes in the load distribution of individual rolling parts in relation to the distributions assumed in the analytical calculations (Kania & Krynke, 2013). The best solution to this problem is computational models using the finite element method (Kania, 2006).

The load capacity of slewing bearings depends on a number of factors, which were grouped according to the Ishikawa method (Fig. 3) (Burduk, Więcek, Tlach, Ságová, & Kochańska, 2021; Kania, 2005; Krynke & Mielczarek, 2016; Smolnicki & Rusiński, 2007; Vaněk, Valverde, Černý, & Hudeček, 2020):

- susceptibility of bearing rings,
- susceptibility of bolts securing bearing rings in the structure of the carrier,
- sizes of the contact zones of the rolling elements with raceways,
- nominal angle of action of the forces transmitted by the rolling elements, and its change when the bearing is loaded,
- coefficient of adhesion of the balls to the raceway,
- fill factor of the rolling circumference of the bearing raceway,
- bearing clearance (including preload in the case of a triple row roller bearing with split rings).

Moreover, the following assumptions are made when calculating the load capacity of slewing ring bearings. (Smolnicki, 2013):

- due to the low rotational speed of these bearings, the centrifugal forces of the rolling balls or rollers are ignored,
- perfect shapes of rings and rolling elements,
- the same diameter of all rolling parts,
- equal hardness of all bearing races,

- materials of the rings and rolling parts are homogeneous and isotropic,
- the limit load on the rolling element is the force causing the relative plastic deformation $\delta_{pl\ dop}/d = 0.0002$.

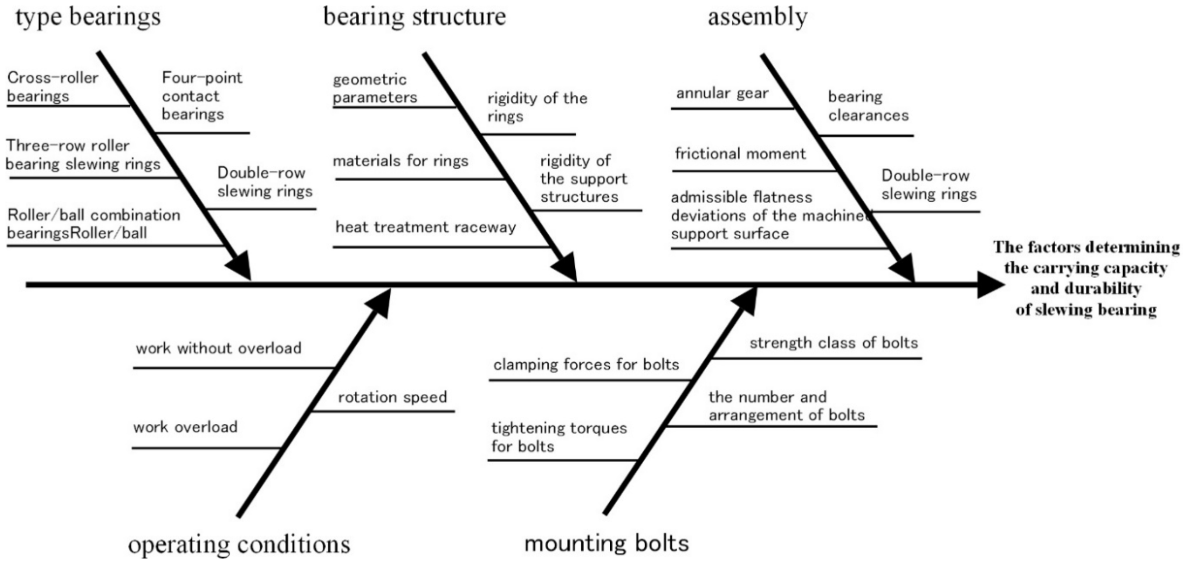

Fig. 3. Factors determining the carrying capacity and durability of slewing ring bearings

The static load capacity of slewing ring bearings is presented in the form of diagrams called characteristics, which are described by the function $M(Q, H)$ where M is the maximum value of the tipping moment, Q is the maximum axial force, and H is the component of the radial load (Fig. 4). The H component is most often taken as a constant value at which the function $M(Q)$ is determined.

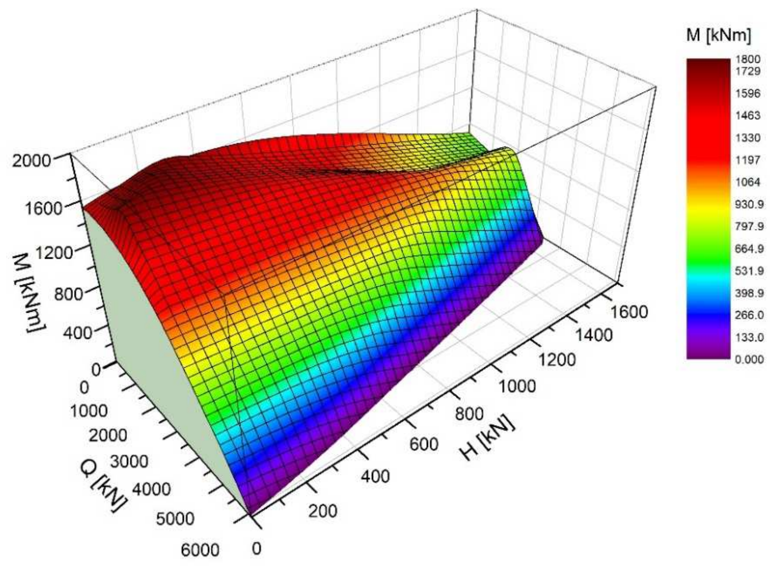

Fig. 4. Spatial diagram of the load capacity of a slewing ring bearing M = f(Q,H)

The load capacity of slewing ring roller bearings is determined by various calculation methods. The simplest calculation models allow determining the bearing characteristics assuming a number of simplifications. A consequence of the simplifications used is an inaccurate estimation of the actual bearing capacity (Kania et al., 2012).

As part of this study, the FMEA method was used to determine the risk associated with underestimating the load capacity of slewing slewing bearings (Wolniak, 2019). This method is an analysis of the causes and effects of emerging errors (Ennouri, 2015). The purpose of this analysis is to find potential causes and effects of errors in the design process phase and to eliminate them before the finished product is created. It is also widely used at the stage of product operation, where there are already failures caused by errors in the implementation of the product, i.e. in the production process. This method is mainly used in design, research and development, and production activities (Knop, 2017; Nedeliaková, Hranický, & Valla, 2022).

In the research, the risk was determined by defining the criteria for selecting the coefficients, i.e .: OCC – risk of error occurrence, SEV – meaning of the error and DET – the difficulty of including the error in the calculations. On the other hand, the priority number of risk was determined using the dependence: according to the formula:

$$RPN = OCC \cdot SEV \cdot DET \tag{4}$$

When carrying out the FMEA analysis, it was assumed that the causes of the occurrence of the load estimation error, in the case when the priority number 1 <RPN <100, no preventive action is required. However, if the priority number $RPN \geq 100$, corrective actions should be taken to remove the causes of errors in determining the load capacity. Table 1 shows the individual factors and the risks that arise if they are not taken into account in the calculation of the load capacity of slewing ring roller bearings..

Tab. 1. Analysis of the causes and effects of errors in calculating the load capacity of slewing ring bearings

CAUSE AND EFFECT ANALYSIS OF ERRORS FOR THE CALCULATION OF THE LOAD CAPACITY OF SLEEVE BEARINGS								
			CURRENT STATE					CONTRACTED ACTIVITIES
Potential error	Effects of Error	The causes of the error	Research measures / methods	*OCC*	*SEV*	*DET*	*RPN*	Recommended preventive actions
Failure to take into account the susceptibility of supporting and load-bearing structures for the machine	Revaluation of the static load capacity	No knowledge of the geometry in the supporting structures at the stage of calculating the catalog bearing	Classic calculation models of bearings	9	6	6	324	It is necessary to get to know the bearing installation structure and prepare a calculation model taking into account the entire structure of the machine related to the rotation node
The change in the nominal contact angle of the balls is not taken into account	Risk of rolling the edge of the bearing raceway, The static load capacity is underestimated	Performing calculations of ball bearings on calculation models for roller bearings where the contact angles do not change	calculation models of bearings in classical and numerical terms	7	4	6	168	In calculation models, balls should be replaced with a special rolling super-element, which allows, inter alia, take into account the changes in the operating angles of the rolling parts due to the transferred loads.
Failure to take into account the compliance of the bearing rings	Incorrect estimation of the static load capacity	The assumptions made in the computational model	Classic calculation models of bearings	5	4	7	140	Preparation of numerical models using MES
Failure to take into account the bolts securing the bearing rings	Incorrect estimation of the static load capacity	Simplification of the calculation model regarding the omission of fastening bolts	Classic calculation models of bearings, without the possibility of introducing fixing screws	6	3	6	108	Model bolts with preload beam elements in numerical MES calculations
Bearing clearance not taken into account	Incorrect estimation of the static load capacity, Incorrect geometry of the gear ring and pinion	Simplification of the calculation model without the possibility of introducing a bearing clearance	Calculation of the equivalent characteristics of rolling elements	7	5	3	105	Introduce the clearance into the calculation model by shifting the equivalent material characteristic for rolling parts
Omission of the parameter deviation from the flatness of the sedimentary surfaces in the calculations	Incorrect estimation of the static load capacity	The assumptions made for the computational model	Classic calculation models of bearings	6	4	4	96	The deviation from the flatness parameter should be included in the geometry of the calculation model for the bearing seats
Pre-clamp is not taken into account	Underestimating the rolling friction moment during bearing rotation	Simplification of the calculation model without the possibility of introducing the initial clamp	Calculation of equivalent characteristics for rolling elements	8	2	3	48	Pre-clamp can be taken into account by shifting the equivalent material characteristic for the rolling parts

The greatest risk that is associated with incorrect estimation of the load capacity of slewing bearings is related to the failure to take into account the susceptibility of the supporting and load-bearing structures for the machine used. In this case, it is necessary to take into account the flexibility not only of the bearing rings and fastening bolts, but also that of the entire supporting structures of the carrier. It should also be remembered that during the operation of the bearing, depending on the intensity of the raceway wear process, the clearances in the bearing increase. Excessive bearing clearances contribute to the deterioration of the meshing conditions of the pinion with the gear ring as a result of uneven load distribution along the tooth line, and also reduce the bearing capacity. The contact angle of the rolling elements also has a great influence on the load capacity of ball bearings. This parameter increases with both increasing loads and increasing bearing clearances.

Results

Slewing bearings, like all rolling bearings, are subject to wear which results from the contact of the rolling parts with the bearing raceways. Such wear, in which the bearing is damaged after working a certain number of hours, i.e. reaching the planned service life, is called natural wear in the literature (Krzemiński-Freda, 1985). Slewing bearings operate in incomparably more difficult conditions than usual rolling bearings, which is correlated with: high load of the rolling parts, plastic deformation of the raceway, uneven load distribution on the bearing circumference, difficulties in maintaining optimal lubrication conditions. This is usually accompanied by the high cost of replacing the bearing, as well as the high cost of the bearing itself. Therefore, in their operation, the criteria useful in assessing the state of wear for plain bearings are rather unnecessary, especially in the case of large-size bearings (Gibczyńska & Pytko, 1999). Wear of slewing ring bearings is allowed beyond these criteria, e.g. for bearings with non-hardened raceways it is allowed to expand significantly, which in a certain period of operation is even beneficial (Stańco, 2008).

Due to the condition presented above, apart from natural wear, there are also a number of examples of wear typical only for slewing ring bearings. They are presented below, at the same time an attempt was made to analyze the causes of this wear and proposed solutions to some problems.

As already mentioned, in slewing bearings there is a natural wear of the bearings caused mainly by high loads on the rolling parts, which leads to pitting (Fig. 5) or spalling (Raadnui & Kleesuwan, 2011).

a) b)

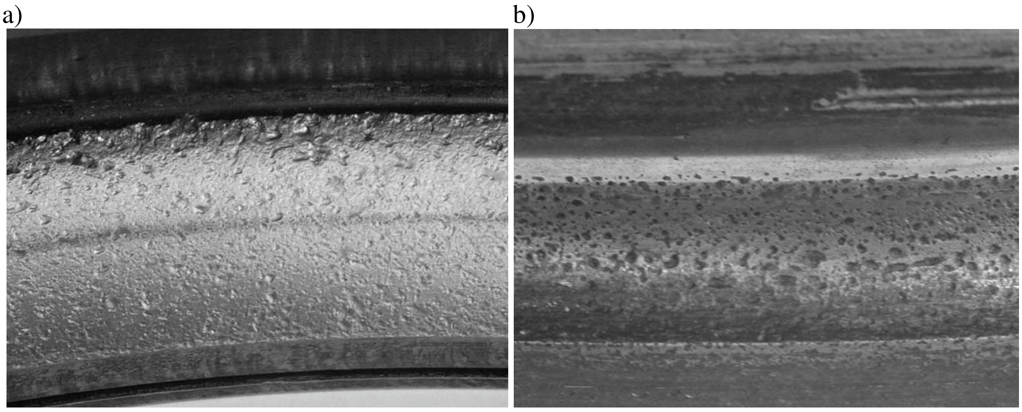

Fig. 5. Traces of pitting on the raceway of a cross roller bearing (a), Traces of corrosion and pitting on the raceway of the ball bearing (b)

The type of fatigue wear is partly related to the conditions of bearing lubrication. Inadequate lubrication conditions, which can occur with some equipment before pitting occurs, may cause the surface to peel off. In this case, premature wear occurs, which is directly related to the fact that the estimated life of the bearing is usually related to the criteria for pitting. Additionally, in the event of inadequate lubrication, the bearing raceways are not protected against corrosion. Corrosive wear is premature wear, and corrosion damage to the raceway is the initiation sites for accelerated fatigue wear. Figure 5b shows an example such of corrosion wear.

The basic lubricating medium in slewing bearings are plastic greases. There are two design solutions for supplying grease to the bearing: through holes made directly in the bearing rings, in which lubricators are mounted (the more frequently used method) and through lines supplying grease from grease nipples installed outside the bearing. O-ring lubricators usually require manual handling, but the use of automatic lubricant dispensers is now recommended, especially in continuous or semi-continuous operation equipment.

Non-lubricated bearings can be used only where their intended use requires it, e.g. in the food industry or in medical devices. In this case, the bearing should be selected in such a way that the load of the rolling parts does not cause plastic deformation in the zone of the greatest stress under the raceway surface (at the Bielajewa point), then the main cause of rapid pitting is avoided. In practice, this means that the operation of the bearings is below their catalog load capacity.

Slewing bearings with surface hardened raceways may have a relatively low service life under certain operating conditions. Immediately after the first signs of pitting appear on treadmills, large chippings of the top layer may appear and spread over long stretches of the treadmill. As a result, it is necessary to withdraw the bearing from service, otherwise the rolling parts may jam and, as a result, damage the bearing, in extreme cases the rings may break (Fig. 6).

Fig. 6. Damage to large parts of the raceway and fractures of the bearing ring

Fig. 7 shows damage to the bearing raceway on large surfaces. The main cause of such phenomena is the overloading of the rolling parts. It may be caused by an excessive external load, it is a clear error in the operation of the machine. It can also result from the formation of excessive play in the bearing, which reduces the number of rolling parts involved in the load and, consequently, an increase in the maximum load (Krynke, Borkowski, & Selejdak, 2014). A common cause of this type of bearing damage is the fact that it is mounted on the carrier girders that are not rigid enough. Due to their uneven deformation, the so-called "hard points of support" are formed around the circumference of the bearing, in which the permissible loads can be significantly exceeded.

Another cause of damage to the raceway may be the insufficient thickness of the hardened layer. This situation causes the zone of maximum stress of the treadmill to extend beyond the surface of the treadmill that has been hardened. Therefore, its thickness should exceed the distance of the Bielajew point from the bearing raceway surface several times. This can lead to cracks in the hardened layer (Fig. 7b).

a) b)

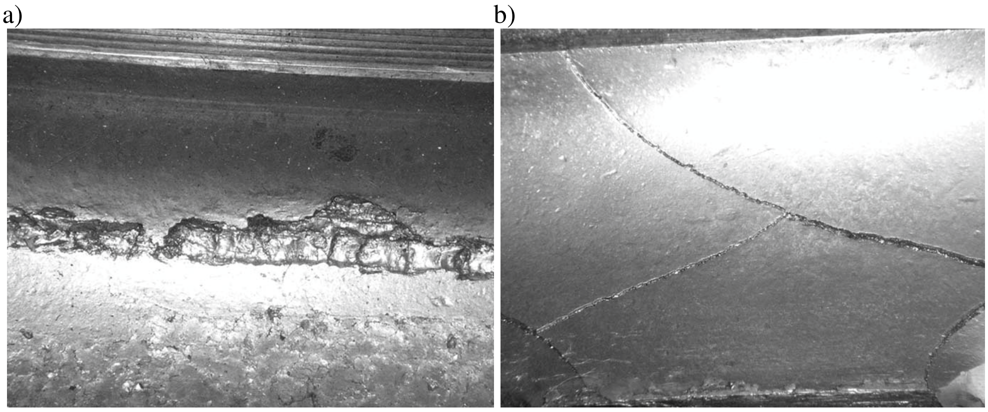

Fig. 7. Damage to large parts of the bearing raceway (a), cracks in the hardened raceway layer (b)

The cause of damage to hardened raceways may also be errors in the heat treatment of the raceway (microcracks), but they occur very rarely due to the careful control of the hardened layer at the bearing manufacturers. This type of damage is more common at critical points due to technology (Kania, 2012) (narrow non-hardened zone).

One of the cases where the raceways of slewing ball bearings are subject to excessive wear is the rolling of the raceway. It can take on a variety of characters. It is the high load on the rolling element that causes the plasticization of the raceway material, which takes place in the contact zone. As a result of the load, the material is dented and some parts of material flows out. After the load is stabilized under the surface of the raceway, a hardened zone can be distinguished, in which the raceway material has a greater hardness than initially. This state of affairs is the result of burnishing, caused by the cyclic turning of the rolling elements and the hardening of the raceway material. The remainder of the bearing raceway remains elastic. The diagram showing the mechanism of deformation wear formation is shown in Figure 8. Relatively large plastic deformations occur in bearings with non-hardened raceways, which are often used as ball beds in mining machines (Smolnicki, 2013). In the initial phase, these deformations improve the adjustment of the raceway shape to the working conditions, and only large expanding becomes the cause of excessive increase in clearances. Such a situation, in turn, entails an unfavorable reduction of the active section of the raceway, thus increasing the load on the rolling parts (the load is transferred by a significantly reduced number of balls). This issue is extensively discussed in the works (Smolnicki, 2013; Stańco, 2008).

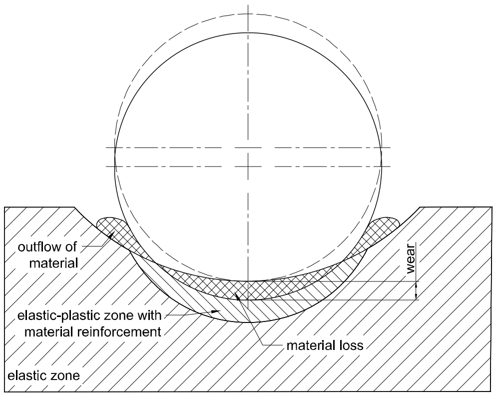

Fig. 8. The mechanism of deformation wear as a result of rolling the ball along the bearing raceway

Another character is the rolling of the raceways in small and medium size single row ball bearings (up to 3000 mm), which usually have hardened raceways. In this case, we should rather talk about the destruction of the edge of the raceway. An example of such a rolling edge of the raceway is shown in Fig. 9. The reason for this type of raceway wear is the increase in the value of the bearing contact angle, which is caused by the load and possibly increased clearance in the bearing. This situation brings the ball-raceway contact area to the raceway edge.

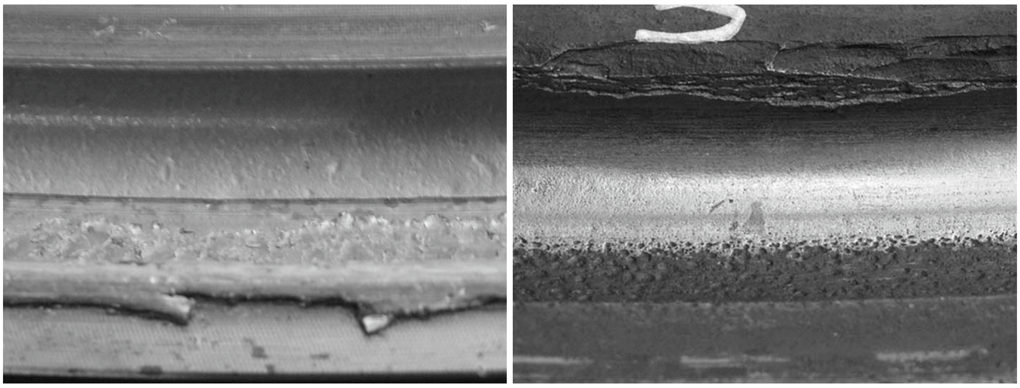

Fig. 9. Strong rolling and chipping of the raceway edge of the ball bearing

Due to the high values of the coefficient of ball adhesion to the raceway k_p (it is the quotient of the ball radius and the radius of the raceway rounding profile) and high ball loads, the size of the contact zone between the ball and the bearing race is significant. This zone will have the shape of an ellipse with a large axis (reference 2a). As a result, the bearing contact angle of the ball with the raceway γ is also large (Fig. 10). Taking into account the factor related to the improvement of the bearing design, the most important value is the maximum angle of the active raceway profile, which is marked in Fig. 10 as α_{max}. Its value can be calculated using the formula:

$$\alpha_{max} = \alpha^* + \frac{\gamma}{2} \approx \alpha^* + \frac{2a}{d} \tag{5}$$

where α^* is the actual operating angle of the bearing, d is the ball diameter.

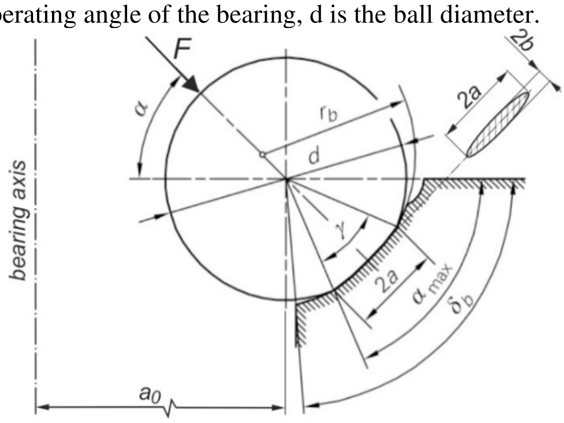

Fig. 10. The dimensions of the contact area between the balls and the bearing raceway for a slewing ball bearing

Figure 10 shows that the raceway profile angle δ_b should meet the condition $\delta_b > \alpha_{max}$, otherwise, during the maximum load of the ball, the ball may cooperate with the edge of the raceway, which in turn causes damage such as shown in Figure 9. Additionally, it should be taken into account that due to the load on the balls, the angle of the bearing's operation changes. The increase of the contact angle reaches the value of approx. 11° with the nominal contact angle $\alpha = 45°$ and the contact coefficient $k_p = 0.97$. It is slightly lower at higher nominal contact angles and decreases with a decrease in the k_p factor. Fig. 11 shows the graphs of changes in the maximum contact angle α_{max} as a function of the adhesion coefficient.

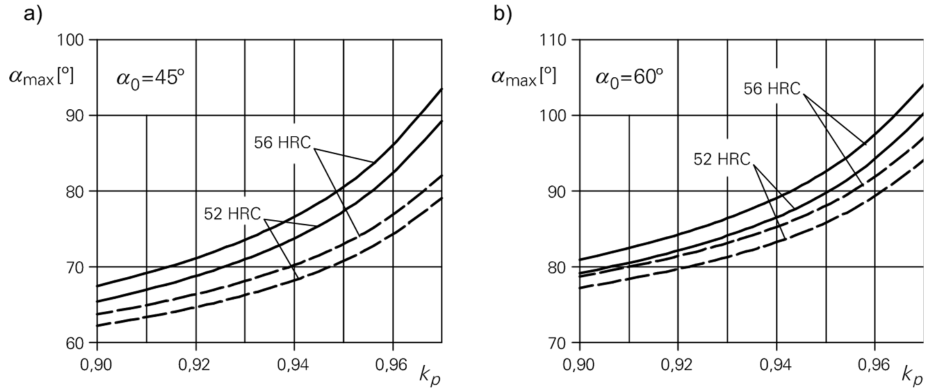

Fig. 11. The relationship between the change in the angle α_{max} as a function of the contact coefficient with the ball load allowed for the raceway hardness of 52 HRC and 56 HRC: a) for α_0 = 45º, b) for α_0 = 60º, dashed line - for the nominal contact angle, continuous line - for the actual contact angle

An important element of this graph is the fact that the angle α_{max} increases to values exceeding 90°. Such angle values can only be obtained in the design of double row ball bearings, in single row bearings smaller contact factors should be used, which, however, reduces the bearing capacity. The most endangered in this respect are wire bearings, where the opening angle of the raceway is small. These bearings cannot bear excessive force loads. Changes in the contact angle should be used as a criterion for calculating the load capacity. The graphs in Figure 11 do not take into account changes in the contact angle of the bearing, which are caused by the play in the bearing, which increases the actual contact angle (Krynke et al., 2014). Therefore, in this respect, it is necessary to carefully control the clearance, which is used in this type of bearings.

Discussion

Identification of changes in the contact angle and load of individual balls in a bearing with flexible rings was carried out on the example of a catalog single-row ball bearing with 86 balls with a diameter of d = 40 mm distributed over a rolling diameter of d_t = 1400 mm (Rothe Erde, 2018). The adhesion factor of the ball to the raceway was k_p = 0.96 and the raceway hardness was 54 HRC. The nominal contact angle for this bearing is 45°. Using the previously developed methods for determining the load capacity characteristics of slewing ring bearings based on MES, the load characteristics were prepared (Krynke, 2010). The load capacity characteristics were prepared for a flexible slewing ring bearing, taking into account the change in the angle of operation of the rolling elements (Fig. 12). The curves that define the ultimate bearing capacity assuming the non-deformability of its rings and taking into account the changes in the angle of operation of the rolling elements under load are shown in the diagram. Differences in load capacity between a bearing with the influence of the flexibility of the rings and the change in the contact angle taken into account, and a bearing with non-deformable rings, and assuming that the contact angle is invariable, are significant. At high loads with tipping moment, the difference amounts to approx. 9%, while for loads with high axial forces Q, there is even more than 30% increase in the static load capacity of the bearing.

Fig. 13 shows the nature of changes in the maximum and minimum contact angles at individual operating points on the load capacity characteristic (points 1-10).

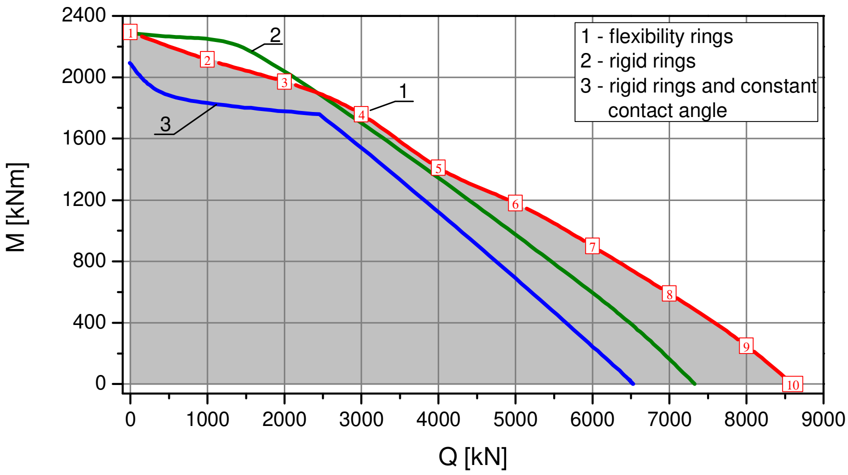

Fig. 12. Load characteristics for single row ball bearings.

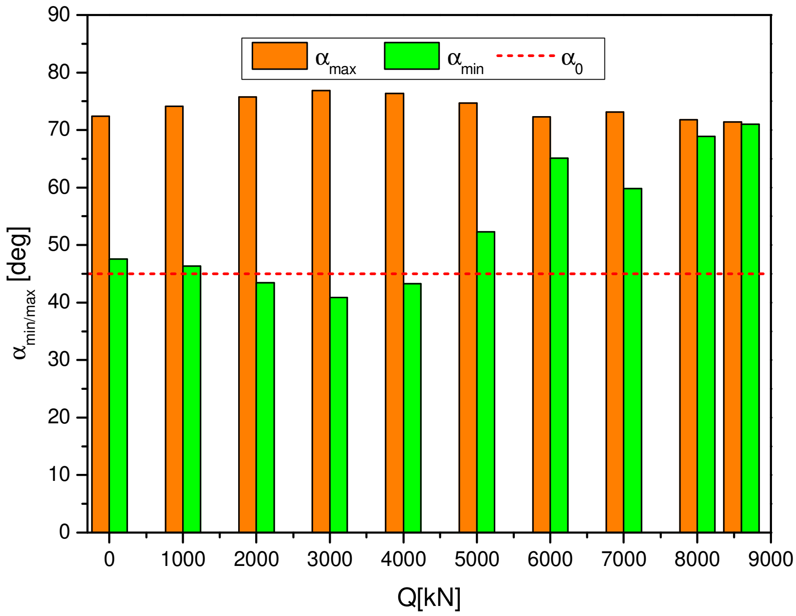

Fig. 13. Minimum and maximum contact angles of the rolling elements for load limit values according to the load capacity characteristics

In a bearing with flexible rings, a significant increase in the contact angles of the rolling elements is visible. This increase is up to 25% in relation to the operating angles of the most loaded rolling elements in a bearing with non-deformable rings. The increase in bearing capacity is caused by an increase in the contact angle. However, at the same time, a too large contact angle in the four-point contact ball bearing may lead to the rolling edge of the raceway, i.e. the contact area of the ball with the raceway is brought to the raceway edge. It should also be anticipated that increasing clearances in the bearing during use cause an additional increase in the operating angles of the rolling elements. In extreme cases, such a situation may lead to accelerated wear of the bearing raceway. In such a case, smaller values of the coefficients of adhesion of the ball to the raceway should be used. This phenomenon does not occur in double row slewing ring bearings.

Conclusions

Decisions already made at the design stage of the bearing and the machine have a significant impact on the durability of large diameter bearings. In addition to design and construction factors, also technological and operational factors are important, in particular the correctness of assembly, machine monitoring and ensuring adequate lubrication.

The reasons for the wear of the slewing bearings, which have been presented, are different from the typical forms of wear of deep groove roller bearings. Large-size rolling bearings are used under a wide variety of operating conditions. Their modes of operation can be completely different, such as rotating at different angles, different load cycles, oscillating movements or continuous rotation. Therefore, the already well-known life criteria for deep groove bearings cannot be applied directly to large diameter bearings, in particular to bearings with intermittent rotation or slow rotation..

The main cause of damage to large-size bearings is usually the excessive load on the rolling parts caused by:

- a significant increase in the bearing clearance and, consequently, a reduction in the active number of rolling parts,
- due to the large and uneven flexibility of the support structures, which in turn leads to a significant excess of the expected load of the rolling parts, which are used on the bearing circumference,
- by exceeding the maximum external load in operation.

In the design and construction process, it is also important to take into account the phenomenon of changes in the operating angles of the rolling elements under the influence of loads transferred through the bearing. Changes in the contact angles of the rolling elements occur mainly in ball bearings. However, the greatest risk from damage to the raceway as a result of this phenomenon is in: single row ball bearing with four-point contact, and with a high coefficient of adhesion the ball to the raceway. The analysis of these changes showed that the angles of interaction of some rolling elements during the limit loads of the ball bearing change even by more than 20 °. In the case of large bearing clearances, which cause an additional increase in the angle of operation of the rolling elements, it may lead to the displacement of the ball. Such a displacement of the ball to a large extent causes that the contact area of the ball and raceway covers the edge of the bearing ring. As a result of this, the edge may be damaged by rolling.

Summarizing these considerations, it should be stated that there is a need for extremely careful design and selection of slewing rings, taking into account the conditions of their foundation. The selection of bearings based solely on the catalog load capacity may in many cases be insufficient. Additionally, particular attention should be paid to the following issues:

- in ball bearings, appropriate angles of the raceway profile should be used in order to avoid degradation of the raceway edges; in justified cases, a bearing larger than it would appear from the analysis of the catalog bearing capacity should be selected; this is very important in relation to wire ball bearings, where the opening angle of the raceway is usually small,
- during the bearing assembly process, the critical points that weaken the locally hardened raceway must be correctly positioned,
- it is necessary to properly lubricate the bearings, it is recommended to use automatic lubricant dispensers instead of manually operated lubricators; in the case of non-lubricated bearings, the bearing should be selected in such a way as to reduce or eliminate plastic deformation at the point of maximum raceway stress.

The discussed causes of wear of slewing ring bearings do not cover all cases, especially the problems of design and operation of large slewing bearings with non-hardened raceways, the reader will find relevant information in the quoted literature.

References

Abramov, I., Bozek, P., Nikitin, Y., Abramov, A., Sosnovich, E., Stollmann, V. (2015.) Diagnostics of electrical drives. In The 18th International Conference on Electrical Drives and Power Electronics . EDPE 2015. The High Tatras, Slovakia, 21 - 23. September 2015. IEEE, 2015, pp. 364-367.

Aguirrebeitia, J., Abasolo, M., Avilés, R., & Fernández de Bustos, I. (2012). General static load-carrying capacity for the design and selection of four contact point slewing bearings: Finite element calculations and theoretical model validation. *Finite Elements in Analysis and Design*, *55*, 23–30. https://doi.org/10.1016/j.finel.2012.02.002

Biały, W., & Fries, J. (2019). Computer Systems Supporting the Management of Machines/Equipment in Hard Coal Mines. Case Study. *Management Systems in Production Engineering*, *27*(3), 138–143. https://doi.org/10.1515/mspe-2019-0022

Biały, W. (2022). Revitalization of Mining Dumps. Assessment of Possibilities, Production Engineering Archives 2022, 28(2), 201-211. https://doi.org/10.30657/pea.2022.28.24

Burduk, A., Więcek, D., Tlach, V., Ságová, Z., & Kochańska, J. (2021). Risk assessment of horizontal transport system in a copper mine. *Acta Montanistica Slovaca*. (26), 303–314. https://doi.org/10.46544/AMS.v26i2.09

Caesarendra, W., Kosasih, P. B., Tieu, A. K., Moodie, C. A. S., & Choi, B.-K. (2013). Condition monitoring of naturally damaged slow speed slewing bearing based on ensemble empirical mode decomposition. *Journal of Mechanical Science and Technology*, *27*(8), 2253–2262. https://doi.org/10.1007/s12206-013-0608-7

El Laithy, M., Wang, L. [Ling], Harvey, T. J., Vierneusel, B., Correns, M., & Blass, T. (2019). Further understanding of rolling contact fatigue in rolling element bearings - A review. *Tribology International*, *140*, 105849. https://doi.org/10.1016/j.triboint.2019.105849

Ennouri, W. (2015). Risk management applying fmea-steg case study. *Polish Journal of Management Studies*, *11*(1), 56–67.

Gibczyńska, T., & Pytko, S. (1999). *Łożyska toczne wieńcowe*. Kraków: Uczelniane Wydawnictwa Naukowo-Dydaktyczne AGH.

Glodež, S., Potočnik, R., & Flašker, J. (2012). Computational model for calculation of static capacity and lifetime of large slewing bearing's raceway. *Mechanism and Machine Theory*, *47*, 16–30. https://doi.org/10.1016/j.mechmachtheory.2011.08.010

ISO (2006). *Rolling bearings - Static load ratings*. (ISO 76:2006).

ISO (2007). *Rolling bearings - Dynamic load ratings and rating life*. (ISO 281:2007).

ISO (2017). *Rolling bearings - Radial bearings - Boundary dimensions, general plan*. (ISO 15:2017).

Jin, X., Chen, Y., Wang, L. [Lei], Han, H., & Chen, P. (2021). Failure prediction, monitoring and diagnosis methods for slewing bearings of large-scale wind turbine: A review. *Measurement*, *172*, 108855. https://doi.org/10.1016/j.measurement.2020.108855

Kania, L. (2005). *Analiza obciążenia wewnętrznego w łożyskach tocznych wieńcowych w aspekcie ich nośności statycznej.* Częstochowa: Wydawnictwa Politechniki Częstochowskiej.

Kania, L. (2006). Modelling of rollers in calculation of slewing bearing with the use of finite elements. *Mechanism and Machine Theory*, *41*(11), 1359–1376. https://doi.org/10.1016/j.mechmachtheory.2005.12.007

Kania, L. (2012). *Zagadnienia lokalne w numerycznym modelowaniu łożysk tocznych wieńcowych*. Częstochowa: Wydawnictwa Politechniki Częstochowskiej.

Kania, L., & Krynke, M. (2013). Computation of the general carrying capacity of slewing bearings. *Engineering Computations*, *30*(7), 1011–1028. https://doi.org/10.1108/EC-02-2012-0023

Kania, L., Krynke, M., & Mazanek, E. (2012). A catalogue capacity of slewing bearings. *Mechanism and Machine Theory*, *58*, 29–45. https://doi.org/10.1016/j.mechmachtheory.2012.07.012

Knop, K. (2017). Analysis of risk of nonconformities and applied quality inspection methods in the process of aluminium profiles coating based on FMEA results. *Production Engineering Archives*, *16*(16), 16–21. https://doi.org/10.30657/pea.2017.16.04

Kovanič, Ľ., Blistan, P., Urban, R., Štroner, M., Pukanská, K., Bartoš, K. and Palková, J. (2020) Analytical determination of geometric parameters of the rotary kiln by novel approach of tls point cloud segmentation. *Applied Sciences (Switzerland)*, 10 (21), art. no. 7652, pp. 1-27. https://doi.org/10.3390/app10217652

Krynke, M. (2010). Modele obliczeniowe i charakterystyki nośności statycznej łożysk tocznych wieńco-wych z uwzględnieniem podatności pierścieni łożyskowych (Praca doktorska). Politechnika Częstochowska, Częstochowa.

Krynke, M., Borkowski, S., & Selejdak, J. (2014). Analysis of influence of bearing clearance on the static carrying capacity of multi-row slewing bearings. *Periodica Polytechnica Transportation Engineering*, *42*(1), 43–48. https://doi.org/10.3311/PPtr.7261

Krynke, M., & Mielczarek, K. (2016). Analysis of causes and effects errors in calculation of rolling slewing bearings capacity. *Production Engineering Archives*, *12*, 38–41. https://doi.org/10.30657/pea.2016.12.09

Krzemiński-Freda, H. (1985). Rolling bearings. *PWN, Warsaw*.

Kunc, R., & Prebil, I. (2003). Low-cycle fatigue properties of steel 42CrMo4. *Materials Science and Engineering: A*, *345*(1-2), 278–285. https://doi.org/10.1016/S0921-5093(02)00464-1

Li, Y., Wang, R., & Li, J. (2022). *Calculation method of static carrying curve for double-row different-diameter ball slewing bearing.* https://doi.org/10.21203/rs.3.rs-1336981/v1

Mazanek, E. (2005). *Zagadnienia konstrukcyjne i wytrzymałościowe w wielkogabarytowych łożyskach tocznych wieńcowych*. Częstochowa: Wydawnictwa Politechniki Częstochowskiej.

Mazanek, E., & Kania, L. (Eds.) (2012). *Problemy modelowania i obliczania łożysk wieńcowych jako podatnych węzłów obrotu maszyn i urządzeń*. Częstochowa: Wydawnictwo Politechniki Częstochowskiej.

Nedeliaková, E., Hranický, M. P., & Valla, M. (2022). Risk identification methodology regarding the safety and quality of railway services. *Production Engineering Archives*, *28*(1), 21–29. https://doi.org/10.30657/pea.2022.28.03

Ojo, O.O., & Obasha, I. O. (2022). Modeling and optimization of friction stir stitching of AISI 201 stainless steel via Box-Behnken design methodology, *Production Engineering Archives 2022, 28*(2), 132-140. https://doi.org/10.30657/pea.2022.28.15

Pallini, R. (1979). Failure analysis and redesign of large-diameter stacker/reclaimer slew bearings. *Lubric. Eng.*

Pamukcu, C. (2015). Analysis and management of risks experienced in tunnel construction. *Acta Montanistica Slovaca*, *20*(4), 271–281.
Pivarčiová, E., Domnina, K., Ságová, Z. (2019). Design of the construction and research of vibrations and heat transfer of mine workings. In *Acta Montanistica Slovaca*, Vol. 24, no. 1 (2019), pp. 15-24.
Raadnui, S., & Kleesuwan, S. (2011). Electrical pitting wear debris analysis of grease-lubricated rolling element bearings. *Wear*, *271*(9-10), 1707–1718. https://doi.org/10.1016/j.wear.2011.01.040
Rothe Erde (2018). *Slewing Bearings – Customer-specific solutions for individual requirements.*
Siwiec, D., & Pacana, A. (2021). Method of improve the level of product quality. *Production Engineering Archives*, *27*(1), 1–7. https://doi.org/10.30657/pea.2021.27.1
SKF (2019). *Slewing bearings.*
Smolnicki, T. (2013). *Wielkogabarytowe toczne węzły obrotowe, zagadnienia globalne i lokalne*. Wrocław: Oficyna Wydawn. Politechniki Wrocławskiej.
Smolnicki, T., & Rusiński, E. (2007). Superelement-Based Modeling of Load Distribution in Large-Size Slewing Bearings. *Journal of Mechanical Design*, *129*(4), 459–463. https://doi.org/10.1115/1.2437784
Stańco, M. (2008). *Modele analityczno – numeryczne zużycia odkształceniowego wielkogaba-rytowych łożysk tocznych* (Rozprawa doktorska). Politechnika Wrocławska, Wrocław.
Turis, J., Beňo, P., & Biały, W. (2018). The Optimal Tribotechnical Factors in the Design of Machines – Environmental Element in the Production Systems. *Management Systems in Production Engineering*, *26*(4), 207–211. https://doi.org/10.1515/mspe-2018-0033
Vaněk, M., Valverde, G. F., Černý, I., & Hudeček, V. (2020). Coal Handling Operational Risk Management: Stripped Overburden Transport in Brown Coal Open Pit Mines. *Acta Montanistica Slovaca*, *25*, 170–181. https://doi.org/10.46544/AMS.v25i2.4
Vicen, M., Bokůvka, O., Nikolić, R., & Brônček, J. (2020). Tribological behavior of low-alloyed steel after nitriding. *Production Engineering Archives*, *26*(3), 78–83. https://doi.org/10.30657/pea.2020.26.16
Wang, S., Chen, J., Wang, H., & Zhang, D. (2019). Degradation evaluation of slewing bearing using HMM and improved GRU. *Measurement*, *146*, 385–395. https://doi.org/10.1016/j.measurement.2019.06.038
Wolniak, R. (2019). Problems of use of FMEA method in industrial enterprise. *Production Engineering Archives*, *23*(23), 12–17. https://doi.org/10.30657/pea.2019.23.02
Yakout, M., Nassef, M. G. A., & Backar, S. (2019). Effect of clearances in rolling element bearings on their dynamic performance, quality and operating life. *Journal of Mechanical Science and Technology*, *33*(5), 2037–2042. https://doi.org/10.1007/s12206-019-0406-y