DAMAGE RISK OF THE RACEWAY EDGE ARISE IN FOUR-POINT SLEWING BEARING

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Abstract: In the article some forms of damage to raceway of slewing bearings for single-row ball bearing slewing ring with four-point contact and their causes were shown. Changes of the contact angle and its influence on the geometry for contact zone of the rolling elements raceway were analyzed. An identification of changes for contact angle of individual balls for different parameters of the contact was received. It was showed that contact angles of some rolling elements were increasing. It can cause damage to the raceway by spalling or rolling out of edge of the bearing ring. Ways of avoiding too early damage to the raceway at the stage of the design and the selection of coronary bearings were suggested.
Keywords: slewing bearing, contact angle, contact zone

1. INTRODUCTION

Slewing bearings, as all rolling bearings, are break down that resulting from the contact of rolling parts with raceway of bearing. Such a wear and tear, where the bearing is undergoing determined number of hours, after achieving the planned durability is called in literature usual wear and tear (Mazanek, 2005). Slewing bearings are working in incomparably more difficult conditions than usually rolling bearings. It is caused by great loading of rolling parts, plastic deformations of the raceway, the uneven disintegration of loading on the circumference of the bearing, the difficulty in preserving optimal conditions of lubrication. The higher estimate connected with the exchange of the bearing, as well as a higher estimate of bearing most often reach it. Therefore than are not in force in their exploitation, especially in the case of big sizes bearings, useful criteria in the assessment of the state of wear for ordinary bearings (Smolnicki, 2013).

Owning to the presented above state, besides the natural wear and tear a number of examples of the typical wear and tear is only appearing for slewing bearings. Damage to the raceway edge is one of such kinds of damage to the slewing bearing. It is caused by the changing contact angle of the of rolling elements under the load (F). In

this article was made an attempt of analysis this causes and proposals to solve some problems were presented.

2. CONSTRUCTION OF THE SLEWING BEARING

The slewing bearing is integrated with the toothed wheel rim on the internal or external ring with placed set of great loading. They usually transfer axial power, the unstable moment and the radial load at the same time. Slewing bearings are often a critical element of the device. Damage of the bearing can cause heavy losses on account of unscheduled stoppages in production. Moreover, its exchange, from the attention to great dimensions, perhaps to last for a few months (Smolnicki and Stańco, 2014).

There is many different types of slewing bearings, depending on the number of rows and the kind of rolling elements. So, bearings can be one, two or three row, and rolling elements can be balls or bearing roller (Mazanek, 2005).

Fig. 1 shows a diameter of the single-row ball slewing bearing with four-point contact. In this type of bearing there is one row of balls and every ball ball is cooperating with two pairs of the raceway.
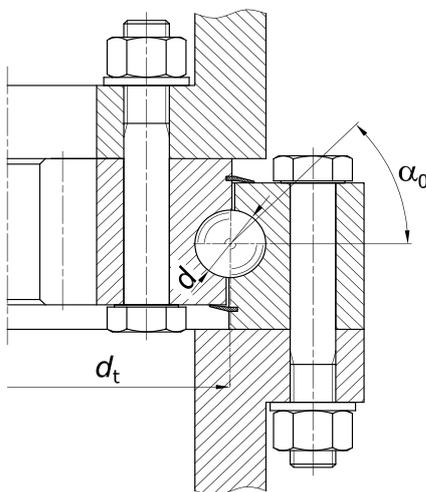

Fig. 1. Rolling knot of a single-row ball bearing with four-point contact

In this type of bearing there is characteristic phenomena resulting from geometry of the contact zone, different than in ordinary roller bearings. It concerns tensing in contact elements, and pressures in the contact zone, as well as sizes of contact zone and changes of the acting angle which are under the influence of external loading.

Curvature diameters of the raceway d_b and rolling elements d are determined by adhesion coefficient ball to raceway k_p:

$$k_p = \frac{d}{d_b} \tag{1}$$

Producers of bearings are recommending to accept value of this parameter between 0.92 and 0.98. In the nominal state, when the bearing is not loaded, between the rolling element and the raceway a contact has a pointed character. The contact angle α is defined as the angle between the straight joining contact points of ball and raceway with plane perpendicular to the axis bearings (Fig. 1).

2. THE WEAR AND TEAR OF SLEWING BEARING CAUSED BY THE EXCESSIVE CONTACT ANGLE

The slewing bearing which is actually designed, produced, installed and utilized, will undergo after all damaging as a result of pitting corrosion or fretting. Pitting phenomenon relies on fatigue wear and tear surface of rolling elements or raceway. It is a result of the cyclical loaded of contact tension and is manifesting itself through coming into existence on the surface or on the little depth under the surface cracks. The propagation of these cracks is leading for shelling scales from the surface of fragments material in the form of teaming lap. Fretting phenomenon depends on micro-crack existence in the area of the rolling element contact with raceway. As a result of this there are cavities (Smolnicki and Stańco, 2014; Ulewicz and Novy, 2016). The permanence of standard roller bearings is defined as the time of their work for the moment of the appearance of the first micro-cracks on the raceway surface or running element and first element of caving or spalling (Ulewicz, 2016).

In the case of large-size bearings on account of the diversity of loading in the working time the durability cannot be treated by bearings as absolute value. It can be treated as an element of definition and comparative value. All slewing bearings are not achieving the theoretical durability, however considerable their part exceeds this durability, achieving in some cases bigger value. This criterion is not valid for bearings that working with low rotation or with oscillatory move. In these cases district speeds are small and wear and tear of the raceway and pitting corrosion are not considerable influence to the sedateness and the precision of the move what is essential in usual roller bearings (Śpiewak, 2016; Korzekwa et al., 2016).

Besides the typical tiredness wear and tear of slewing bearings they are appearing different, characteristic mainly for slewing bearings of damage of raceway (Krynke and Mielczarek, 2016; Knop and Mielczarek, 2018). One of such kinds of damage is raceway rolling in single-row balls about small and average sizes (up to 3000 mm) that usually have hardened raceway. In this case it should tell rather about damage to the edge of raceway. The example of such raceway damage was shown in Fig. 2. An increase of the bearing contact angle is a cause of the raceway wear and tear. It is caused by loaded and enlarged or bearing slackness. It causes leading the field of the contact ball with raceway to raceway edge.
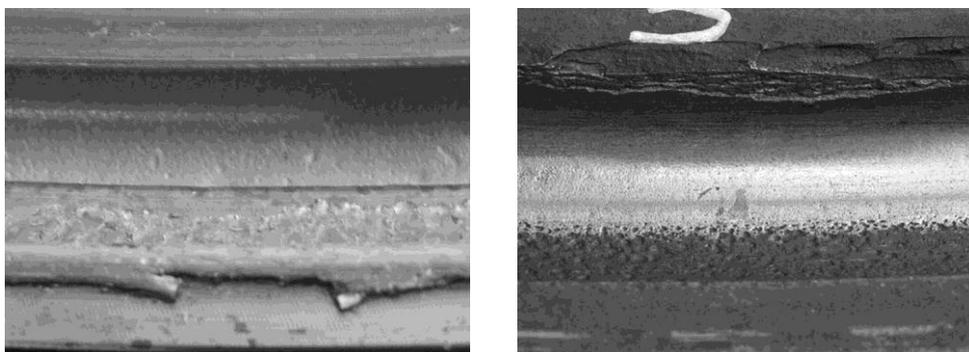

Fig. 2. Strong rolling out and spalling of the raceway edge a ball bearing

In Fig. 3 were presented schematically basic parameters of the contact zone of ball and the raceway. Geometry of pointed contact are characteristic in main curvatures contact. In roller bearings one main plain marked with indicator 2 are outlining axes of the bearing and the rolling part, and second perpendicular to it was marked with

indicator 1. It contains the straight joining points of the contact on both raceway (Kania, 2012). In the slewing bearing ball about the nominal contact angle α curves of the ball and the raceway are determined according to formula in Fig. 3:

$$\rho_{11} = \rho_{12} = \frac{2}{d}, \qquad \rho_{21} = -\frac{1}{r_b}, \qquad \rho_{22} = \pm \frac{1}{\frac{a_0}{\cos\alpha} \pm \frac{d}{2}} \tag{2}$$

where the first index of the curve is identifying the body, and second is identifying the main plain.
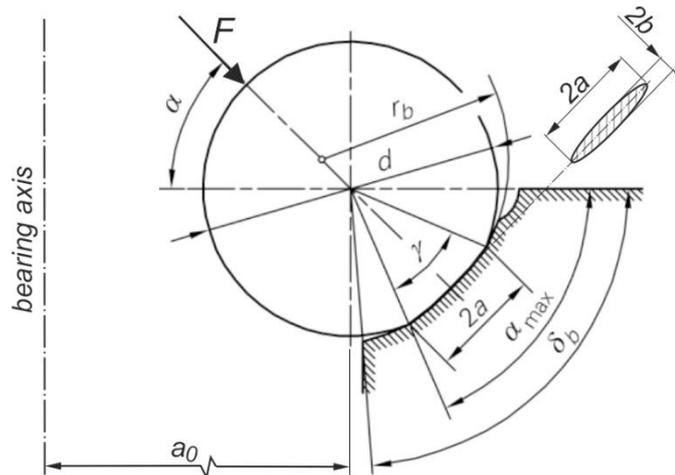

Fig. 3. Comparison of curvature and sizes of the contact zone of ball with raceway slewing bearing ball

The curvature convex is positive, the curvature concave – negative. So curvature $\rho 22$ is positive for the internal ring and negative for the external ring. The sum of curvatures of contact bodies is determined by the formula:

$$\Sigma\rho = \rho_{11} + \rho_{12} + \rho_{21} + \rho_{22} \tag{3}$$

The contact zone is an ellipse (Fig. 3), lengths of its axis *2a* and *2b* is from formulas (Kania, 2012):

$$2a = 2\mu_H \sqrt[3]{\frac{3(1-\nu^2)}{E\,\Sigma\rho}F}, \qquad 2b = 2\nu_H \sqrt[3]{\frac{3(1-\nu^2)}{E\,\Sigma\rho}F} \tag{4}$$

where: μ_H and ν_H are coefficients of contact zone, usually selected from the tables (Eschmann et al., 1987), *E* –modulus Young, ν – Poisson ratio.
From the attention to great value of coefficient adjoining the ball to the raceway k_p (it is a quotient of the ray of the ball and the ray of the profile of rounding raceway) and great loading of balls. A size of the contact zone of ball with raceway is considerable, more closely big contact ellipse *2a*. It causes also that carrying angle of raceway profile is big γ (Kania et al., 2016). From a point of view of the correct construction of the bearing in this case a maximum angle of the dynamic profile of the raceway was marked in Fig. 3 as the most important size as αmax. It is possible to calculate from the formula (Kania, 2012):

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

where α^* is a real action angle of bearing action.
It results from Fig. 3, that angle of the raceway profile δ_b he should fulfil condition $\delta_b > \alpha_{max}$, differently during the maximum load of the ball it can reach for the cooperation of the ball with the edge of the raceway what is a cause of such damage

as in Fig. 2. Additionally it should take into consideration that as a result of the loading of balls is reaching for the change of the action angle of the bearing. The increase in the action angle reaches value about 15° by the nominal action angle $\alpha = 45°$ and rate of adhesion $k_p = 0.96$. It is a little smaller by bigger nominal action angles and is decreasing with the fall in the coefficient k_p (Kania, 2012). Fig. 4 shows graphs of changes for the maximum angle of the raceway profile α_{max}, real action of the bearing α^* and angle of the raceway profile o γ in the function of the rate of adhesion.
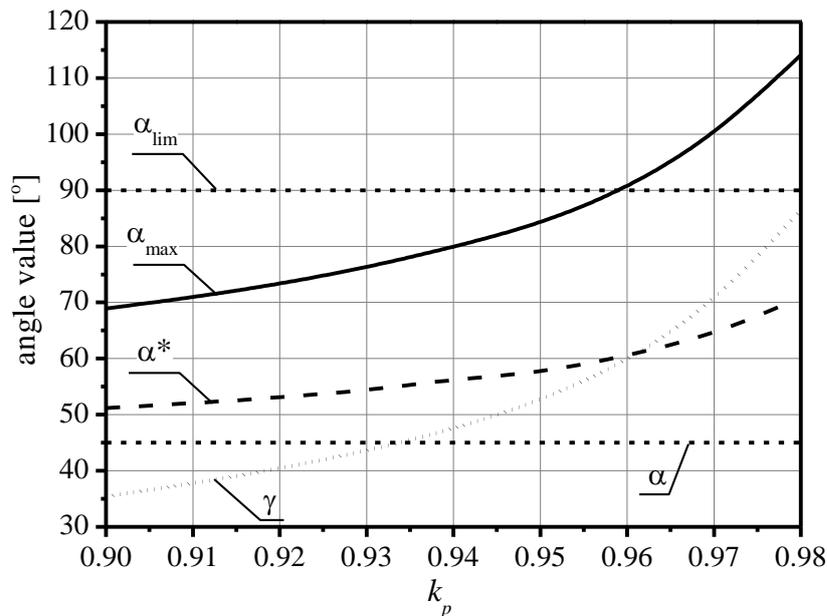

Fig. 4. Course of changes for the maximum angle of the raceway profile αmax, real contact angle of the bearing α^* and angle of the raceway profile γ in the function the rate of adhesion at acceptable loading the ball for the raceway hardness 54 HRC and $\alpha = 45°$

It should be pay attention to the height of the angle α_{max} to value 90°. It is possible to reach that value in construction of double-row ball baring. In single-row should apply smaller rates of adhesion what is forming with the fall in the load capacity of the bearing. The most throated are wire bearings where the angle of flare is small. These bearings cannot transfer too great loads. Changes of the contact angle should constitute the criterion of calculating the carrying capacity for them. Graphs in Fig. 4 are not taking into consideration changes of the contact of the bearing caused by bearing slackness. It cause that real action angle is increasing. Hence the necessity of particularly a careful control of slackness in this type of bearing.

This type damage of the raceway can be a source of disastrous destruction of the bearing through deep spalling of raceway. They cause the necessity of analyses of the disintegration of the contact tension in the contact zone, peculiarly at great changes of the contact angle. In spite of recommendations of producers as for maximum value of baring slackness which in fact are not too great. The utilization practice many times is not warning these principles, then exaggerated slackness can change real contact angle of zone even much over 30 ° what results in the very strong loading for the raceway edge.

3. SUMMARY

Changes contact angles of rolling elements are appearing above all in balls bearing. The greatest risk of raceway damage for the sake of the increase in contact appears in ball bearing single-row with four-point contact at the big rate of adhesion of the ball to the raceway. Analysis of these changes showed that action angles of some rolling elements loading for a ball bearing were changing even about over 20°. In the case of appearing of great bearing slackness which causes the additional height of the contact angle. It can result in great ball transfer, that the field of the contact the ball and raceway includes the edge of ring bearings. As a result of this occurrence can appear damage of the edge through its spalling. In a double-row ball bearing, on account of the different geometry of raceway mating with rolling elements, it is not reaching this phenomenon. Therefore double-row balls bearing can correctly work by bigger axial clearance.

Summing up, it should be stated that there is necessity exceptionally careful design and the selection of slewing bearings considering conditions of their foundation. Additionally, in balls bearing, should be applied appropriate obtuse angles of raceway profile, in the purpose of avoiding the degradation in the edge of raceway. In justified cases it should be chosen the bigger bearing than it would result from catalogue bearing capacity. In the special way it is links to wire ball bearings, where obtuse angle of raceway is rather not large.

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