CONTACT STRESS STUDY AND FME ANALYSIS OF LARGE SIZE THRUST BALL BEARINGS

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Abstract. Thrust ball bearings with extremely large size and capacity, are not standardized and could not be found in catalogues. In aim to investigate possible failure causes of thrust ball bearings in heavy machinery excavators, this paper presents stress and deformations calculation supported with FME analysis in balls-raceway interfaces because of carrying whole upper construction and allow rotation movement. In this paper it was performed numerical calculations and their comparison for two, conceptually different types of large size thrust bearings. First one is classical pivot, such as load distribution in modified new bearing solution. Contact stress calculation of those bearings and FME analysis show that stress and deformation values for ball are near to the limit of static load capacity. Presented study supported by images of balls and raceways damage appearance, with or without grease as a lubricant, could be very useful in the bearing failure analysis.

Keywords: *Thrust ball bearing, contact stress, contact deformation, FEM analysis, bearing failure.*

I. INTRODUCTION

For machines such are cranes, excavators, turntables, concrete pumps, cement mixers or carrousels, large size thrust ball bearings are common applied. This kind of thrust bearing, can perform both slewing (oscillating) movements as well as rotational movements and because of that "slewing bearing" is their common name. Basically, a slewing bearing consists of an inner and outer ring with rolling elements (balls or cylindrical rollers) separated by spacers (sages). Rings, one of which usually incorporates a gear, are provided with holes to accommodate attachment bolts. There are appropriate recommendations for design according to the required dimensions, load capacity, conditions for manufacturing or installing, maintenance or repair [1]. In "Kolubara mining pool in Serbia" three different thrust ball bearings are mounted and exploited in bucket wheel excavators (Fig. 1, 2).

Fig. 1 Bucket wheel excavator in "Kolubara " mining pool

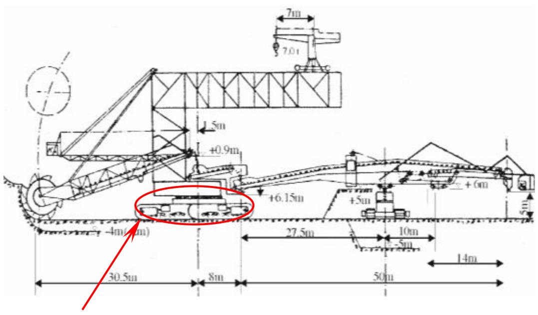

Fig. 2 Thrust bearing in bucket wheel excavator

In aim to investigate potential causes of failure of thrust bearings, here will be presented the analysis of stress and deformations in balls-washers interface, starting from classical pivot arrangement of the bearing, as shown in Fig. 3.

Fig. 3 Thrust ball bearing with classical pivot arrangement

Slewing bearing as an object of the study in this paper carries mass of 1500 tons, has about 10000mm outside diameter and because of dimensions consists of 30 segments, each is 230 mm high. Each of 120 balls in the bearing has 200mm diameter and weight of respectable 33kg! The circular motion of the excavator is done through an axial ball bearing and the drive via toothed collar. Toothed outer ring has 33 teeth, with module of m=3mm, assembled into 30 identical segments (cross section in Fig. 4).

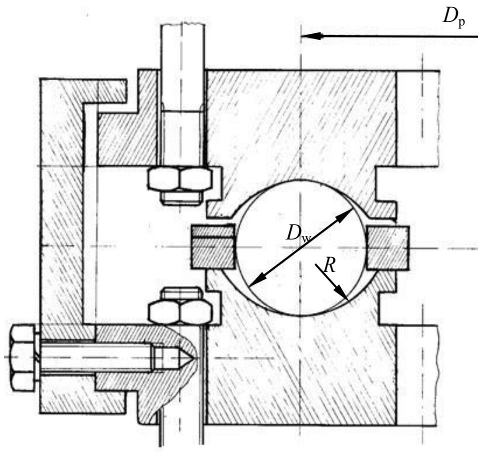

Fig. 4 Cross section of classical pivot arrangement

II. CALCULATION OF STRESS AND DEFORMATIONS

Under the load, contact point of every ball with upper and lower raceway become a small surface of ellipse shape. Stress and deformations values on those places can be determined by using known Hertz's theory of contact stress [3].

In this case of thrust ball bearing loaded by central external axial force F_A, static balance equation of the bearing could be represented by relation:

$$F_A = \sum_{i=0}^{z} F_i \, , \qquad (1)$$

where F_i is partial load of a single i-ball, and z is number of balls in the bearing (Fig. 5).

Because of the load F_i in ball - groove interface of upper and lower washers, local contact deformations are occur. Every contact point become small area with ellipse shape, where magnitude of the contact ellipse is determined by geometry of contact surface (ball and washers groove). The value of contact pressure (*Hertz stress*) depends of ellipse size and normal load. Stress and deformations on those places can be determined using Hertz's theory of contact stress [3, 4]. Total contact deformation of i-ball and corresponding raceway is calculated by known relation:

$$\delta_i = C_F F_i^{2/3} . \tag{2}$$

Loading constant C_F in the relation (2) depends on material and geometry of contact surfaces:

$$C_F = \frac{3}{2}\left(\frac{2K}{\pi n_a}\right)\sqrt[3]{\frac{1}{3}\left(\frac{1-\nu^2}{E}\right)\frac{2}{D\mathrm{w}}\left(2-\xi\right)}, \tag{3}$$

where E is Young module of materials in interface; ν is *Poisson* coefficient of contact surfaces materials; D_w ball diameter; $\xi = D_w/2R$ represents contact geometrical parameter; where R is groove washer radius (Fig. 3); $2K/\pi n_a$ plays a role of bearing constant, which depends on the geometrical parameter ξ and is determined on the basis of appropriate tables or diagrams [4].

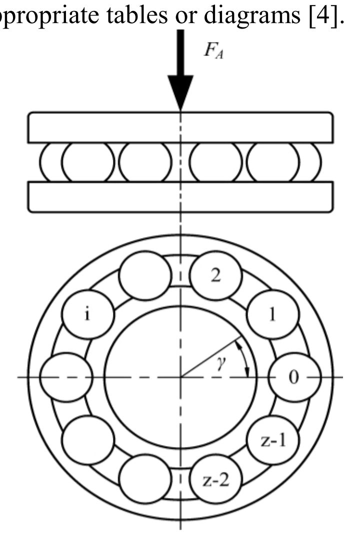

Fig. 5 Centrically loaded bearing with *z* number of balls

In aim to determinate contact deformation on corresponding ball in the bearing, it is necessary to know single ball load F_i. In this case of uniform load distribution, according to relation (1), load of every ball is calculated as:

$$F_i = \frac{F_A}{z}, \quad i = 0 \dots z-1 . \tag{4}$$

Since the load is uniform distributed on all balls, than contact deformation is equal on every single ball in the bearing. Using relation (2) for deformation values, we could write:

$$\begin{aligned} F_0 = F_1 = F_2 = \dots = F_i = \dots = F_{z\text{-}1} \\ \delta_0 = \delta_1 = \delta_2 = \dots = \delta_i = \dots = \delta_{z\text{-}1} \end{aligned} . \tag{5}$$

Value of *Hertz* stress used to be calculated [5, 6] as:

$$\sigma_i = \frac{3}{2}\frac{F_i}{a_i b_i \pi}, \tag{10}$$

where a_i and b_i are semi minor axis of contact ellipse on ball and raceway interface, as shown in Fig. 6.

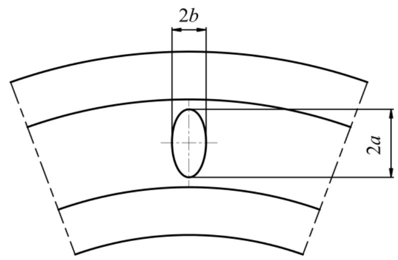

Fig. 5 Ball-raceway contact ellipse

Dimensions of contact ellipse, shown in Fig. 4, could be calculated by using following relations:

$$\begin{aligned} a_i &= n_a \sqrt[3]{\frac{3}{2}\frac{(1-\nu^2)D_w F_i}{E(2-\xi)}} \\ b_i &= n_b \sqrt[3]{\frac{3}{2}\frac{(1-\nu^2)D_w F_i}{E(2-\xi)}} \end{aligned}, \tag{11}$$

where n_a i n_b are numbers those are depending on geometrical parameter ξ and could be determinated by tables and corresponding diagrams [4].

Results of above described calculations are presented in Table I. Based on data for observed variant of thrust ball bearing, with using dimensions of their internal geometry (Fig. 3), loading constant C_F and deformation constant C_δ have been calculated, such as contact ellipse dimensions.

TABLE I
Stress and Deformation Calculation Results

Quantity, unit	**Bearing performance**	**Quantity, unit**	**Bearing performance**
z	120	C_F	$4.654 \cdot 10^{-5}$
D_w, mm	200	C_δ	$3.149 \cdot 10^{6}$
ξ, mm	0.8	m_o, t	1500

m_w, kg	33	F_A, kN	14715
E, N/mm^2	$2.1 \cdot 10^5$	F_i, kN	122.625
v	0.3	δ_i, mm	0.115
n_a	1.818	a_i, mm	9.276
n_b	0.628	b_i, mm	3.204
$2K/\pi n_a$	0.876	**σ_i, MPa**	**1970**

III. FEM ANALYSIS OF STRESS AND DEFORMATIONS

In aim to compare above mentioned calculation results, also FEM analysis is conducted. The first analysis is done for thrust ball bearing with classical pivot arrangement, which means that inner and outer ring are horizontally positioned (Fig. 4), and ball-raceway contact areas with elliptical shape are positioned on top and bottom of the ball (Fig. 6). All those analysis for selected trust ball bearing segment are conducted in ANSYS software tool.

The ANSYS structural analysis software suite enables solutions for complex structural engineering problems. With this finite element analysis tool, it is possible to make better and faster design decisions, because it allows us to analyse multiple design scenarios. ANSYS Structural Mechanics software easily connects to other physics analysis tools, providing even greater realism in predicting the behaviour and performance of complex products. ANSYS FEA software is used throughout industry to enable engineers to optimize their product design.

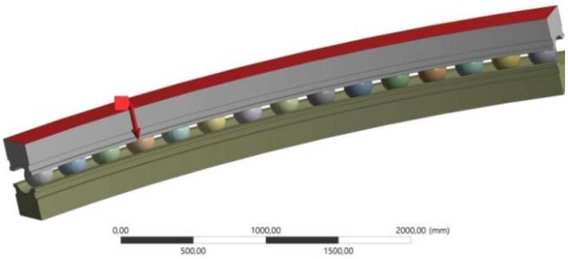

Fig. 7 Model of classical pivot solution with axial load

According to FEM analysis results for model of classical pivot solution (Fig.7), we could see the place and maximum stress value on ball-raceway contact such as ellipse shaped stress distribution (Fig. 8).

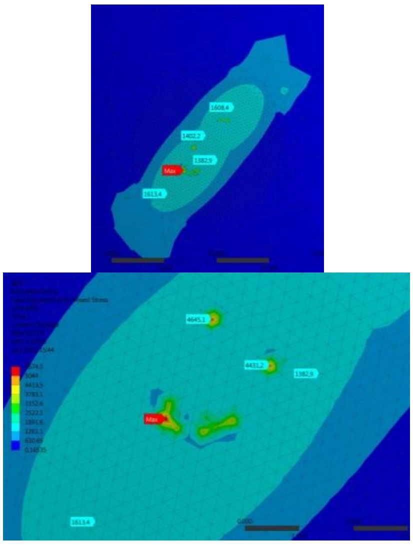

Fig. 8 Ellipse of contact with contact stress values

For this step in calculation and analysis, contact stress and deformations have been determinated, where bearing load is centrically applied and thus load distribution is uniform. Calculation results for uniform stress distribution on ball-raceway contact have enough small values compared with static load rating. According to ISO standard [5] static load capacity is defined by contact stress value of 4200 MPa, which represents plastic deformation in the ball-raceway interface of $10^{-4}D_w$. Calculated values in Table I are calculated for bearing variant where in case of uniform load distribution contact stress value rich only 45% of static load ratings. Those are expected and common values in the exploitation of this kind of roller bearings. Similar results are obtained also for two other solutions with different number of balls (159 till 203), weight (5 till 14kg) and diameter (110 till 150mm). It those cases stress values could rich 54% till 58% of the bearing static load capacity [11].

In results obtained by FEM analysis we could observe a few places with extremely maximum stress values (red colour points in Fig.8). If we take those points as singularities of the method, which means calculation error, we could except them from real results. Contact area shows similar ellipse shape as calculated, where stress values are in a line with numerical obtained results, a bit under the static load rating.

IV. COMPARATION WITH NEW BEARING SOLUTION

The authors of the paper try also to compare this classical pivot solution with new thrust ball bearing design, that is in recent time more in exploitation for large size machines.

FEM analysis for two different thrust bearing solutions is conducted in aim to compare results for both bearing design which common name is slewing bearing, because they carry axial and also rotation load of the whole upper structure. The main characteristic of new slewing bearing solution are horizontally positioned rings, where line of ball-raceways contact points is horizontal (Fig. 9).

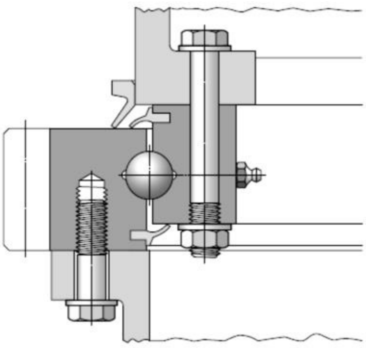

Fig. 9 New slewing bearing solution

FEM analysis for this new bearing solution model with load application in axial direction on inner ring (Fig. 10), is conducted under the equal load value as for classical pivot solution, which allow us to compare their load capacities.

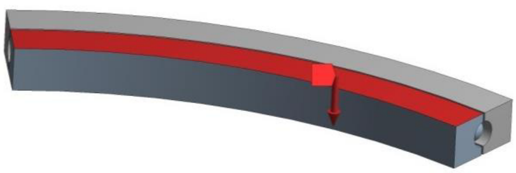

Fig. 10 Model of new slewing bearing solution for FEM

Results of contact stress show us places with their maximum values on every ball under the corresponding load for this kind of bearing design Fig. 11.

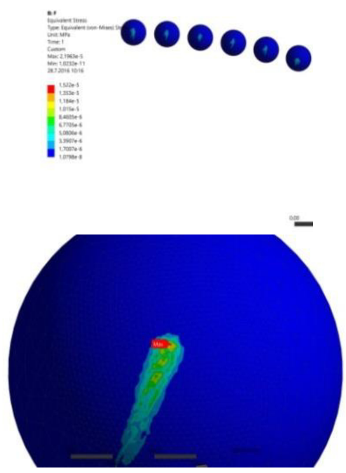

Fig. 11 FEM analysis results for new bearing solution

Obtained results of maximal contact stress values for this solution done by FEM analysis allow us to conclude that new solution for thrust bearing is much better under pure axial load, because maximal stress values are even 10 times less than for classical pivot design. Doubtless advantage of this new bearing solution could be also identified in case of simultaneous action of centrically load in addition of tilting moment, which is common situation during the exploitation period. Obviously this new design for non-uniform load distribution is less sensitive under the complex load acting. This could be explained by respectable tilting angle for classical pivot solution, which could lead to extremely high stress level of particular balls in contact [11].

V. FAILURE ANALYSIS AND MAINTENANCE

Presented results could be a perfect background for failure explanations of classical pivot solution applied in bucket wheel excavators for mining or similar high loaded machines [8]. In aim to describe given calculation results and show effects of bearing operating conditions, several examples of ball and raceway failures should be shown and explained. Those damage samples and images are taken during regular lifetime maintenance intervals and reparations needed to be done in "Kolubara mining pool", Serbia, where wheel excavators where engaged in exploitation.

The phenomenon of loosing operating capacity is most pronounced in the case of unplanned excessive shock loads, as well as large angles of relative tilting motion of the bearing washers. Then, due to large imbalances of load, stress and deformation of some particular balls greatly exceed the allowable value, determined by static load ratings (Fig. 12).

Fig. 12 Lost operation capacity of highest loaded ball

Extremely heavy burden, static overload or load impact, could lead to the so-called "false brineling" as imprints on the ball-raceway interface (Fig. 13).

Fig. 13 False brineling and flaking of loaded balls

In case of further rolling over those "imprint areas", it could leading to further intensive progressive destruction of the surface - the separation of the surface layers of metal (flaking contact surfaces). Contamination of bearing contact surfaces are different possible origin, for example metal particles as products of previous surfaces destruction of rolling bearing such as hard mineral abrasive particles from environment, as sand, stones or minerals [9]. Due to the penetration of contaminants into the bearing interior, additional imprints and abrasive wear of ball-raceway interface can occur (Figure 14).

Fig. 14 Abrasive wear of ball on contact surface

Inadequate lubrication of the washers raceways leads to friction increasing and the reduction of hardness of the contact surfaces and further adhesive to intensive wear (Figure 15).

The primary function of a lubricant is to create enough thick film between the rolling elements and raceways as well as between the gears, to prevent metal-to-metal contact. As thrust ball bearings normally operate at slow speeds, the free space between the rings could be filled entirely with grease (Fig. 16). Unless otherwise specified bearings are filled with an NLGI class 2 mineral oil based EP grease containing a lithium soap thickener [1,10].

Fig. 15 Wear of bearing upper surface raceway

The first degreasing and first lubrication of the gear should be carried out immediately after mounting the bearing, until the grease exudes from the seals around the whole circumference. This grease provides extremely good corrosion inhibiting properties and excellent mechanical stability. The standard seals used in slewing bearings provide good protection against moisture and contaminants and also provide reliable retention of the lubricant.

Fig. 16 Lubricated thrust bearing ready for exploitation

VI. CONCLUSIONS

Slewing bearings were originally designed to be mounted only on horizontal support structures, but can now be used successfully in vertical bearing arrangements. The forces and load distribution in slewing bearings are subjected to axial, radial load and tilting moment. Adequate lubrication and regular maintenance intervals are of essential importance for every thrust bearing operating and maximum service life. Those bearings are generally lubricated with grease, that provides added protection against the ingress of water and contaminants, where seal selection is also vital for their performances. This paper represents only a continuous attempt to give some calculations and discussion aimed to contribute in better explanation and analysis of thrust ball bearing failures. Those combined loaded bearings with extreme dimensions and specific way of maintenance are common application in mining machines such are bucket wheel excavators, as an example of machinery employed in the thermo energetic sector of Serbia [10]. Even this shows an extraordinary application; analysis and discussion made in this paper is step forward of complex numerical calculations [11] and failure study. Conducted FEM analysis of standard pivot solution and compared with new thrust ball bearing design show doubtless advantages of new modified solution for this kind of bearings. Here to be comment that also additional numerical calculations used to be done, may be also in wear predicting process, where ANSYS software tool make it possible based on above mentioned stress and deformation analysis. Investigation in this field is still in progress with hope that authors of the paper with their contributions in combination with other research results could leads to some general conclusions and recommendations in proper maintenance of this complex bearings and machinery.

ACKNOWLEDGMENT

This work has been performed within the project TR-35011 Republic of Serbia. This project is supported by the Ministry of Education, Science and Technological Development Republic of Serbia, which financial help is gratefully acknowledged.

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