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[1.]Mirjana Bojanić ŠEJAT, [1.]Aleksandar ŽIVKOVIĆ, [1.]Ivan KNEŽEVIĆ,
[1.]Milan RACKOV, [1.]Cvijetin MLAĐENOVIĆ

MODELING OF STATIC BEHAVIOR OF FOUR POINT CONTACT BALL BEARING

[1.]University of Novi Sad, Faculty of Tehnical Sciences, Trg Dositeja Obradovića 6, Novi Sad, SERBIA

Abstract: Within this paper is described an analytical model for the analysis of quasistatic load distribution in four contact ball bearings. The quasistatic model was developed on the basis of the static model, applying Hertz's theory of contact and John-Harris's load distribution on ball. In this paper, the model is extended by introducing parameters such as positive/negative clearances into static equilibrium equations. Behavior analysis of the four point contact ball bearing (FKL LSQFR 308) was performed for different operating conditions. Within certain analyzes, the analysis of the influence of conceptual parameters (positive/negative clearances) on the operational characteristics of bearings was performed. The change in external load also varied. Verification of the static behavior of the ball bearing LSQFR 308 was performed by comparing the results obtained by quasistatic modeling and the results obtained using the finite element method.

Keywords: bearing, clearance, load, mathematical model, four point contact

1. INTRODUCTION

In agriculture, as in all areas of industry, the demands placed on machines and equipment are growing. In today's agricultural industry, we are moving towards the development of bearings based on the reduction of weight and dimensions, while increasing the operational characteristics of bearings. A typical bearing that is increasingly used in agricultural mechanization is the four-point contact bearing, which is used, among other things, to support the seeder disc.

Hui-Yuan et al. (Hui-yuan, Chun-xi et al. 2012) are presents a contact analysis method for large negative clearance four-point contact ball bearing based on the theory of Hertz Contact. The results show that contact loads of four contact points of the bearing increase with the absolute value of negative clearance, tiny changes of the negative clearance have a great effect on contact stresses. The same conclusion was reached by *Wang, Wu and Zhu* (Wang, Wu et al. 2013) in the analysis of bearings used for windmills. The values of the nominal contact loads between the ball and the inner/outer ring of the bearing were calculated at different axial preload, without external loads. Distribution of loads by quasi-static analysis of axial ball bearing with four-point contact was shown by *Wang* (Wang and Zhu 2013).

Based on *Hertz's* theory of contact, the contact stress can be determined at anyone point. The results show that the ball comes into contact with the inner/outer raceway at four point whose contact surfaces and contact stress distribution are approximately the same. *Chen* (Chen, Zhang et al. 2010) has shown experimentally that preload significantly increases the contact forces between the balls and the raceway and that these forces create elastic deformations on the bearing rings. The mathematical model established by *Chen* takes into account the elastic deformations of the ring, without external load, and it represents the basis for calculating the load distribution, bearing capacity and bearing life. *Aguirrebeitia* et *al*. (Aguirrebeitia, Plaza et al. 2014) considered the effect of preload on ball. They adapted the theoretical model and the finite element model (FEM) to consider the effect of preload on static bearing capacity and total bearing stiffness of four-point contact ball bearing.

Within this paper, an analytical model for the analysis of quasi - static load distribution in four - point contact ball bearings is described. The quasi-static model was developed on the basis of the static model, applying *Hertz's* theory of contact and *John-Harris's (Jones 1960), (Harris 2001)* load distribution on balls. In this paper, the model is extended by introducing influence, positive/negative clearances into static equilibrium equations. In addition to the quasi - static model, a mechanical model of ball bearings was developed, modeled in a general purpose software system based on the finite element method (FEM). Within certain analyzes, the analysis of the influence of conceptual parameters (positive/negative clearances) on the operational characteristics of bearings was performed. Verification of the static behavior of the bearing LSQFR 308 was performed by comparing the results obtained by quasi-static modeling and the results obtained using the finite element method.

2. MATERIALS AND METHODS

Figure 1 shows the internal kinematics of four-point contact ball bearing, i.e. the position of the center of the ball and the position of the center of curvature of the inner raceway with and without the combined load on the balls relative to the center of curvature of the outer raceway, assumes in this case it is fixed. Shown positions of the center of the ball in Figure 1, marked with P W, V, α_{ir}, α_{il}, α_{or}, α_{ol}, Δ_{ir}, Δ_{il}, Δ_{or} and Δ_{ol} are the axial and radial components of the position of the center of the ball, the angle of contact with the inner right and left raceway, as well as the outer right and left raceway and the distance between the center of the ball and the center of curvature with the inner right and left and outer right and left raceway. When the external combined load acts on the balls, due to different angles of contact between the ball and the outer or inner raceway, the line of action of the load will not be collinear with the distance between the centers.

All bearing elements are deformed under the action of the external combined load, that is, there is a relative movement of the inner ring in relation to the outer ring which is fixed. As a result of these changes, the distance between the center of curvature of the inner and outer raceway and the new position of the center of the ball changes. Therefore, under the action of the combined load, there is a relative displacement of the inner ring in relation to the outer one by the magnitude of the axial and radial displacement u_a, u_r as well as the angular displacement θ (Figure 1). In this case, the center of curvature of the right raceway (C_{ir0}) is moved to a new position (C_{ir1}) and the center of curvature of the left raceway is moved from the starting position (C_{il0}) to a new position (C_{il1}). In this case, the rolling elements will make contact with the raceway at three or four points, depending on the size of the axial load and the size of the positive/negative clearance in the bearing.

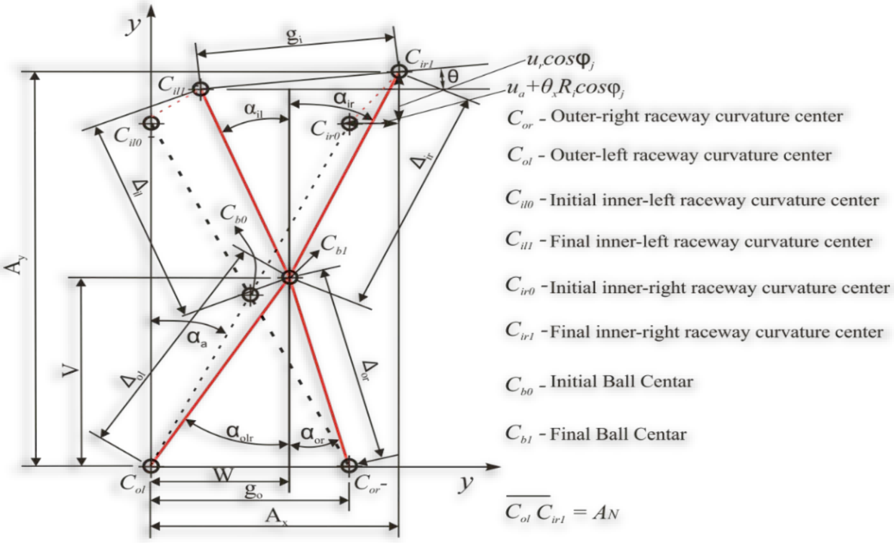

Figure 1. Position of ball center and raceway curvature centers before and after load

The equations of equilibrium of kinematic constraints between the ball and the raceway from Figure 1 are:

$$\begin{aligned}
&\left[A_{yj}-V_j\right]^2+\left[A_{yj}-W_j\right]^2-\Delta_{il,j}^2=0\\
&\left[A_{yj}-V_j-g_i\sin\theta\right]^2+\left[g_i\cos\theta-A_{xj}+W_j\right]^2-\Delta_{ir,j}^2=0\\
&V_j^2+W_j^2-\Delta_{or,j}^2=0\\
&V_j^2+\left(g_o-W\right)_j^2-\Delta_{ol,j}^2=0
\end{aligned} \tag{1}$$

Equilibrium force equations for each ball from Figure 1are:

$$\begin{aligned}
&Q_{il}sin\alpha_{il}+Q_{ol}sin\alpha_{ol}-Q_{ir}sin\alpha_{ir}-Q_{or}sin\alpha_{or}=0\\
&Q_{il}cos\alpha_{il}+Q_{ir}cos\alpha_{ir}-Q_{ol}cos\alpha_{ol}-Q_{or}cos\alpha_{or}+F_c=0
\end{aligned} \tag{2}$$

For each contact between the ball and the raceway, the contact load can be expressed as in the equation:

$$Q_{i,o(j)}=K_{i,o}\left(\delta_{i,o}^{3/2}\right)_j \qquad \text{(Valid for } \delta\geq 0\text{)} \tag{3}$$

When the balls are outside the load zone, the angle of contact with the outer right and left raceway is equal to zero. In this case, relation (3) becomes:

$$\begin{aligned}
&Q_{i,l(j)}=Q_{i,r(j)}=0\\
&Q_{o,l(j)}=Q_{i,r(j)}=F_{cj}
\end{aligned} \tag{4}$$

In order for all balls to be in contact with the raceway, the following condition must be filled:

$$\begin{aligned}
&\sqrt{\left(A_{xj}^2+\left(A_{yj}-\Delta_{o,rj}\right)^2\right)}\leq\left(f_i-0{,}5\right)d_b\\
&\sqrt{V_j^2+\left(g_o-W\right)_j^2-\Delta_{ol,j}^2}\leq\left(f_i-0{,}5\right)d_b
\end{aligned} \tag{5}$$

Further, the three bearing reaction forces are defined:

$$\begin{aligned} F_x &= \sum_{j=1}^{z}\left[Q_{il,j}\sin\alpha_{il,j} - Q_{ir,j}\sin\alpha_{ir,j} \right] \\ F_y &= \sum_{j=1}^{z}\left[Q_{il,j}\cos\alpha_{il,j} + Q_{ir,j}\cos\alpha_{ir,j} \right]\cos\varphi \\ M_y &= \sum_{j=1}^{z}\left[Q_{il,j}\cos\alpha_{il,j} + Q_{ir,j}\cos\alpha_{ir,j} \right]\sin\varphi \end{aligned} \tag{6}$$

3. RESULTS

The analysis of the static behavior of the bearing LSQFR 308 was performed for external axial and radial load, where bearings with radial positive/negative clearance from -20 to 30 μm are observed. The analysis includes the effects of contact deformations, contact loads, displacement of the bearing center and stiffness, under the action of given loads. Analyzes were performed at speed $n = 200$ rpm. The distribution of maximum equivalent stresses with inner left and outer right raceway for $F_a = 2000$ N is shown in Figure 2. Table 1 shows a comparison of maximum contact pressure and maximum contact deformations determined by quasi-static and FEM modeling for different axial loads at zero clearance. The values obtained by the quasi-static model were taken as reference.

a) b)

Figure 2. Distribution of maximum equivalent stress on: a) inner left; b) outer right raceway at $F_a = 2000$ N and $G_r = 0$ μm

Figure 3 shows the number of rolling elements involved in load transfer depending on the positive/negative clearance for different values of the external radial load.

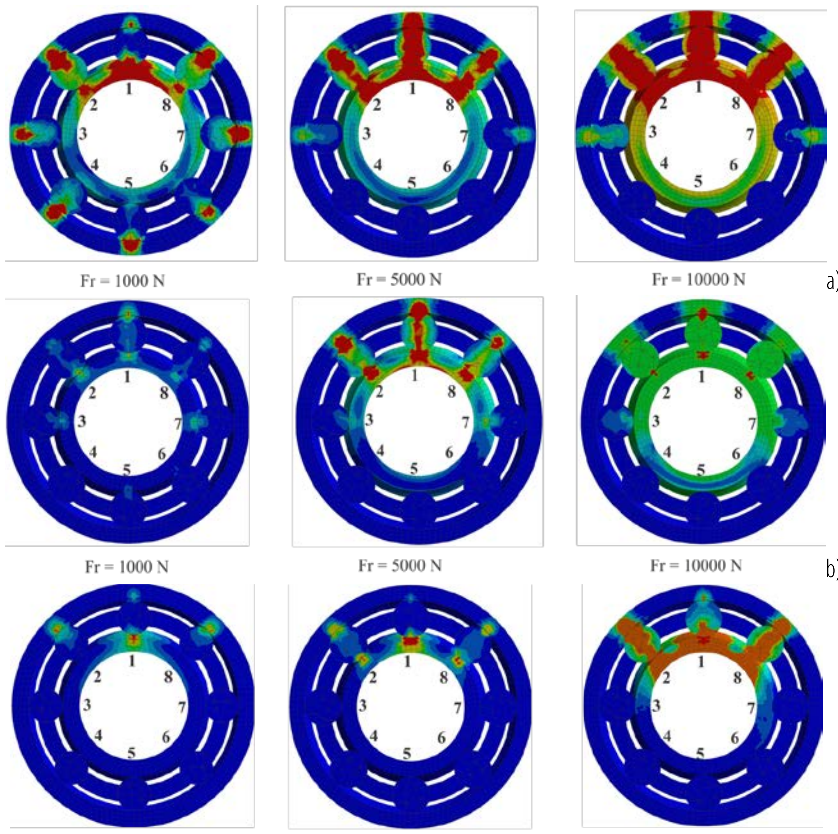

Figure 3. Load distribution for: a) $G_r = -20$ μm, b) $G_r = 0$ μm and c) $G_r = 20$ μm depending on the external radial load for the bearing LSQFR 308

Table 1. Comparison of max. contact pressure and deformation determined by quasi-static and FEM modeling

Load F_a [N]	Quasi-static model		MKE model		Deviation [%]	
	Contact pressure [MPa]	Deformation [µm]	Contact pressure [MPa]	Deformation [µm]	Contact pressure	Deformation
400	1395	5,9	1467	6,25	-4,91	-5,60
800	1493	6,8	1618	7,37	-7,73	-7,73
1200	1575	7,6	1630	7,98	-3,37	-4,76
1600	1655	8,4	1697	8,83	-2,47	-7,87
2000	1729	9,2	1830	9,82	-5,52	-6,31

4. CONCLUSIONS AND DISCUSSION

When the radial load acts on a four-point contact ball bearing, there is an uneven load distribution on the ball. The number of ball that participate in load transfer depends on the size of the positive/negative clearances, and on the other hand, whether the contact will be in two, three or four points. Increasing the negative clearances or reducing the positive clearance (or increasing the axial load) increases the number of balls that transmit radial load and the number of contact points between the balls and the raceway, which significantly reduces the degree of uneven distribution in external load transfer (Fig. 3).

If the results of contact pressure and deformation obtained by the analytical model and the FEM model are compared, the maximum deviations are below 8%, which is at a satisfactory level.

Finally, it should be noted that the development of four-point contact ball bearings is still a current and insufficiently researched area, therefore, analysis and examination of the impact of any parameter of this bearing on its static and dynamic behavior may be the direction of future research.

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