Influence of Clearance on Load Distribution of Crossed Cylindrical Roller Slewing Bearings

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***Abstract*- Clearance is an important parameter influencing on the performance of the slewing bearings. The influence law of clearance on load distribution of crossed cylindrical roller slewing bearings was researched. Firstly, the statics model of the bearing with external combined loads was established and a set of nonlinear equilibrium equations were obtained; then, the roller load distribution was calculated by using Newton-Raphson iterative procedure. By using this model and method, the relation of internal bearing clearances and the roller load distribution was analyzed. Results show that with the decrease of the clearance, the number of rollers undertaking the loads begins to increase, and the roller load distribution tends to be homogenized. As the clearance decreases from positive value to negative value, the maximum roller load decreases firstly and then increases gradually. There is an optimal clearance value which is propitious to improve the carrying capacity of the bearing.**

Keywords- Slewing Bearing; Clearance; Load Distribution; Carrying Capacity

I. INTRODUCTION

Slewing bearings are machine elements that enable relative rotation of two structural parts, they can accommodate axial load, radial load and tilting moment, acting either single or in combination and in any direction. Crossed cylindrical roller slewing bearings have excellent rotational accuracy and stiffness. They are widely used in machining centers, medical equipment, precise turntable, measuring devices, IC card manufacturing equipment and other fields of mechanical engineering. The crossed cylindrical roller slewing bearings are also used in mining machinery, construction machinery and other large turning table.

Clearance is an important parameter of the slewing bearing, the value of which impacts the rotating accuracy and the load distribution of the bearing. Meanwhile load distribution of the slewing bearing determines its performance parameters such as deformation, contact stress, stiffness and fatigue life. So, it is necessary to investigate the influence law of clearance on load distribution.

In recent years, the mechanical analysis of slewing bearings has attracted much attention by the researchers. S. Zupan and I. Prebil [1] researched the influence of the geometry parameters of the rolling contact, such as the actual play, the shift direction of raceway center and the osculation on the carrying capacity of the four-point ball slewing bearings. Furthermore, S. Zupan and I. Prebil also take into account deformation of the supporting struction, which is defined by its stiffness matrix obtained on the base of the finite element method analysis. I. Prebil, P. Kaiba [2] presented a program environment to help the designer with the design process of four-point slewing bearings, the designer can dimension and optimize the composing bearing parts by the experts system built in the program. Robert Kunc, et al [3] presented the material model, numerical analysis of the actual carrying capacity of the rolling contact in single row ball bearings and the verification of the numerical material model with experimental results of low cycle carrying capacity. Srečko Glodež [4] proposed a comprehensive computational model for determination of the static capacity and the fatigue life of raceway of large slewing bearing, the model consider most of the specificity, such as deformation of bearing rings and raceway hardness-dependent material properties. L.Kania [5] analyzed the load-deformation characteristics of the roller-raceway contact for three-row roller slewing bearing, it was performed by using the FEM and the substituting of bearing rollers with truss elements. Since there is no need to calculate the contact force between the raceway and the rollers, this significantly simplifies the finite element analysis. Amasorrain J.I., et al [6] established the statics equilibrium equations of the single row four-point contact ball slewing bearing according to the bearing geometrical parameters and the relationship between the ring displacement and the approach of two raceway groove curvature centers, the rolling element forces were obtained by solving the equations. The analysis is on the assumption that the ring is rigid and the influence of the surface flatness error of the bearing supporting platform was not considered. Potonik P, et al [7] established the statics equilibrium equations of the double row four-point contact ball slewing bearing by using vectors to represent the bearing geometrical structure. The maximum rolling element load was obtained by solving the equations, and the finite element analysis of the contact between the rolling element and the raceway was performed. Ludwik Kania, et al [8] established the simplified model of the slewing bearing by using finite element analysis software ADINA, the rolling element is simplified as nonlinear truss elements, connecting bolt was simplified as beam element, rolling element load distribution and static carrying capacity curve were calculated by using this model.

In this paper, static models of the crossed cylindrical slewing bearing considering the clearances were established, by solving the nonlinear equilibrium equations of the models, the internal roller loads distribution was obtained. Based on these calculations, the loaded roller number and the maximum roller contact force of the bearing were determined, and the influence laws of the clearance on them were researched further.

II. STATICS MODEL

Crossed cylindrical roller slewing bearing has two set of rollers and raceways, all the rollers are arranged in one row and the adjacent rollers are orthogonal. This allows two set of rollers to fit into single-row space for conserving space and saving material costs. As shown in Fig. 1, the bearing is composed of inner ring (1), seal (2), roller (3) and outer ring (4). For the purpose of analysis, a polar coordinate system is set with its origin at the bearing center in the radial plane of the bearing, the polar angle is designated by φ_i, the region of φ_i is $[0,2\pi]$. At any angular position the contact force between the rolling element and the inner upper and outer lower raceway is defined as $Q_{1\varphi}$, the contact force between the rolling element and the inner lower and outer upper raceway is defined as $Q_{2\varphi}$.

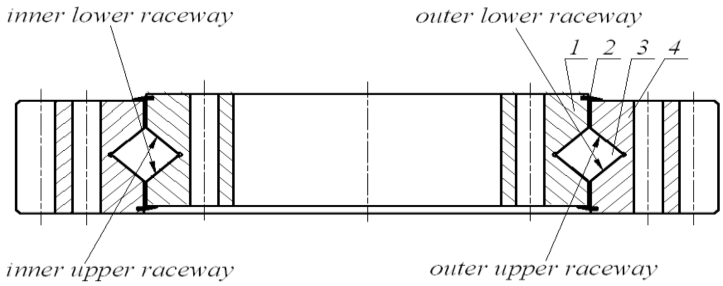


Fig. 1 Polar coordinate system of the slewing bearing

According to the direction of the roller contact load, the rolling element will be divided into two groups, the angular position φ_1 corresponding to $Q_{1\varphi}$ can be expressed as

$$\varphi_{1i} = 4\pi(i-1)/Z \tag{1}$$

the angular position φ_2 corresponding to $Q_{2\varphi}$ can be expressed as

$$\varphi_{2i} = 2\pi/Z + 4\pi(i-1)/Z \tag{2}$$

where Z is the total roller number of the bearing, and $i = 1,2,\cdots,Z/2$.

Under most working condition, slewing bearings rotate with very low speed, hence the static model is valid to calculate the internal contact force. Suppose the outer ring of the bearing is fixed, under the combined action of radial load F_r, axial load F_a and tilting moment load M, the corresponding radial displacement, axial displacement and angular displacement of the inner ring is δ_a, δ_r and θ, as shown in Fig. 2.

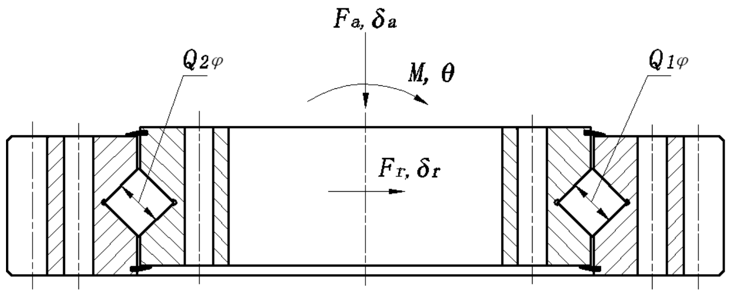


Fig. 2 Displacements of inner ring under combined loads

Fig. 3 (a) shows the initial normal distance between the inner ring raceway and outer ring raceway at any roller angular position along the bearing circumference before the bearing is loaded.

$$A = D_w + \frac{1}{2} G_a \cos\alpha \tag{3}$$

where D_w is roller diameter, G_a is the axial clearance of the bearing, and α is the nominal contact angle of the rollers.

When the bearing is loaded, the position of the inner ring raceway, outer ring raceway and the rolling element are shown in Fig. 3 (b). According to deformation compatibility condition, the relative displacement of the inner upper raceway at any angular position of rolling element can be written as:

$$S_1 = (\delta_a + \frac{1}{2}\theta d_m \cos\varphi_i)\sin\alpha + \delta_r \cos\varphi_i \cos\alpha \tag{4}$$

(a) Before the bearing is loaded (b) After the bearing is loaded

Fig. 3 Distance between two raceway way

According to Fig. 3, after the bearing is loaded, the normal distance between the inner ring raceway and the outer ring raceway at any angular position of rolling element is

$$A_1 = S_1 - A \tag{5}$$

So, the elastic contact deformation betweent inner upper and outer lower raceways and the roller is

$$\delta_{1\varphi} = D_w - A_1 \tag{6}$$

Substituting Eq. (5) into Eq. (6), the following equation is obtained for the contact deformation between the rolling element and the inner upper and outer lower raceway.

$$\delta_{1\varphi} = \delta_r \cos\varphi_i \cos\alpha + (\delta_a + \frac{1}{2}\theta d_m \cos\varphi_i)\sin\alpha \tag{7}$$

In the same way, the contact deformation between the rolling element and the inner lower and outer upper raceway can be written as:

$$\delta_{2\varphi} = \delta_r \cos\varphi_i \cos\alpha - (\delta_a + \frac{1}{2}\theta d_m \cos\varphi_i)\sin\alpha \tag{8}$$

According to hertz contact theory, the contact force can be expressed in terms of contact deformation and contact stiffness. Hence, the contact force at any angular position can be calculated by the following equation:

$$Q_{j\varphi} = \begin{cases} K_n \delta_{j\varphi}{}^{\mathrm{n}} & \delta_{j\varphi} > 0 \\ 0 & \delta_{j\varphi} \le 0 \end{cases} \quad j = 1,2 \tag{9}$$

where the exponent $n = 1.11$ is valid for the line contacts. K_n is the total contact stiffness of the contact between the rolling element and the raceway, which depends on the property of the geometry and elasticity material of raceway and rolling elements, and can be calculate as:

$$K_n = \frac{1}{[(1/K_i)^{1/n} + (1/K_e)^{1/n}]^n} \tag{10}$$

where, for steel and the raceway contact:

$$K_j = 7.86 \times 10^4 L_w{}^{8/9}, \quad j = i, e \tag{11}$$

The force and moment equilibriums of the bearing inner ring can now be written as follows:

$$F_a = \sum_{i=1}^{Z} Q_{1\varphi} \sin\alpha - \sum_{i=1}^{Z} Q_{2\varphi} \sin\alpha , \tag{12}$$

$$F_r = \sum_{i=1}^{Z} Q_{1\varphi} \cos\alpha \cos\varphi_i + \sum_{i=1}^{Z} Q_{2\varphi} \cos\alpha \cos\varphi_i \tag{13}$$

$$M = \sum_{i=1}^{Z} Q_{1\varphi} \sin\alpha(0.5d_m \cos\varphi_i) - \sum_{i=1}^{Z} Q_{2\varphi} \sin\alpha(0.5d_m \cos\varphi_i) \tag{14}$$

Substituting formulas (7) and (8) into the above three equilibrium equations, a system of non-linear equations can be obtained. If the geometrical parameters of the bearing are given, the values of unknown variables δ_a, δ_r and θ can be solved by Newton-Raphson method. The contact force $Q_{1\varphi}$ and $Q_{2\varphi}$ can be calculated according to formula (9) further.

The maximum contact force is obtained from the contact load distribution, as follows: $Q_{\max} = \max(Q_{j\varphi})$, where $j = 1,2$.

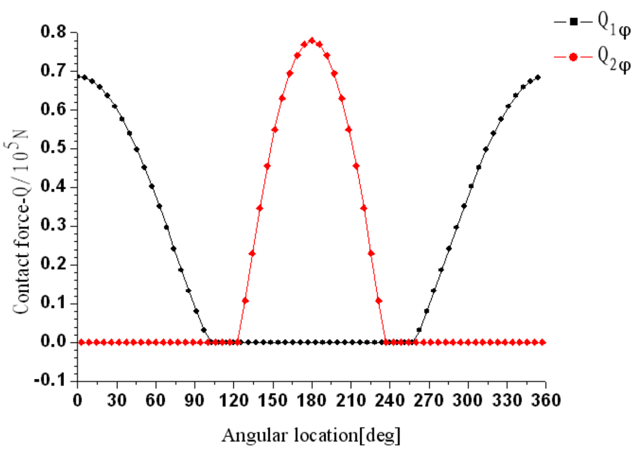
III. RESULT AND DICUSSION

A practical example was done on a crossed slewing bearing, the geometry and working parameters are shown in Table I.

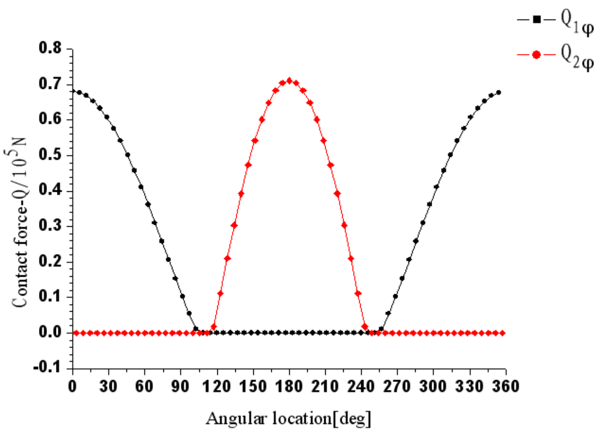
TABLE I PARAMETERS OF SLEWING BEARING

Parameter item	**Value**
Dw (mm)	70
Lw (mm)	69.5
α (°)	45
Z	126
Fa (kN)	350
Fr (kN)	150
M (kNm)	2000

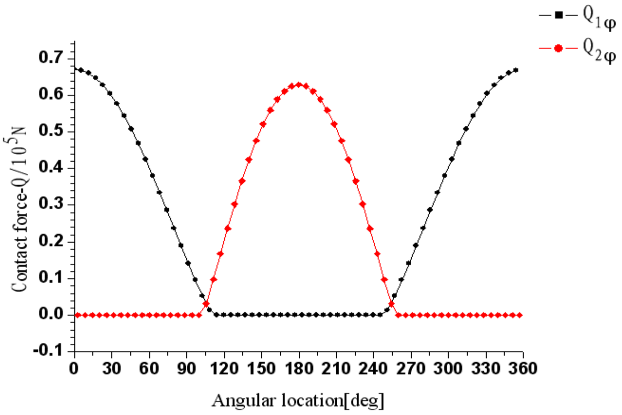
Apply Newton-Rapson algorithm to find the unknown variables, we could get the contact force distribution along the raceway for different clearance. As shown in Fig. 4, when the clearance is 0.1, 0.05, 0, -0.05, -0.06 and -0.07, the number of rolling elements contact the raceway is 54, 60, 66, 86, 92 and 104 respectively. As it can be seen from this figure, we could find that with the clearance decrease from 0.1 to -0.07 towards the negative direction, the rolling elements which could contact the raceway increase gradually, and the contact force at any angular position have different values and directions. Furthermore, the curve of the load distribution along the raceway is basically in the same trend with different clearance, with the clearance decrease toward to negative direction, more and more rolling elements began to undertaker external load, and the load distribution curve is not so steep.



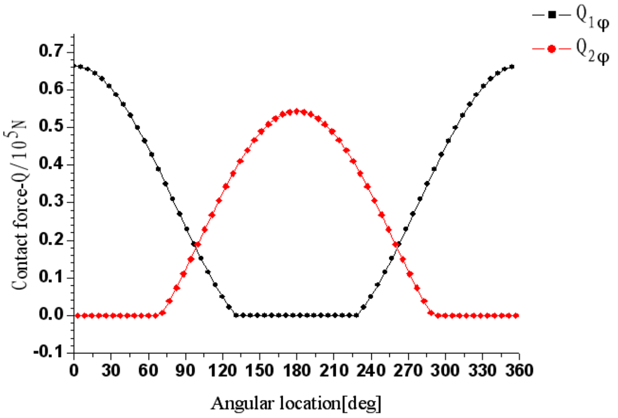
(a) Bearing clearance: 0.10mm



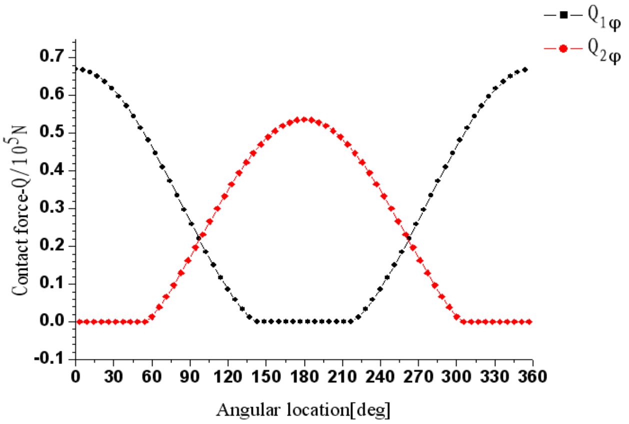
(b) Bearing clearance: 0.05mm



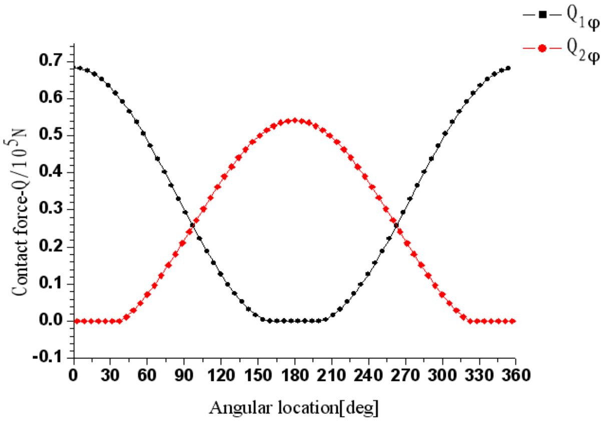
(c) Bearing clearance: 0 mm



(d) Bearing clearance: -0.05mm



(e) Bearing clearance: -0.06mm



(f) Bearing clearance: -0.07mm

Fig. 4 Load distributions for different clearance

Fig. 5 shows the maximum roller contact force changes with the clearance. As can be seen from the figure, under the certain external load, when the bearing clearance in the range [0.1, 0.15], with the reduction of the clearance, the maximum contact force of the roller decreases. While it is unexpected that contact force reduced to a certain value then increases drastically. This is due to usually crossed cylindrical roller bearing undertaker external force mainly by tilting moment (alone or in any combination with axial and radial forces). Influence of clearance on the maximum contact force impacted by the relation of clearance and the tilting moment. As the clearance is in the range [0.15, -0.06], the roller loads are impacted mainly by the tilting moment. As the bearing with bigger clearance, contact force caused by tilting moment will be more concentrated in a smaller number of the rolling elements, then less and less of the rolling element with the given external load to achieve a balance, the maximum contact force would increase undoubtedly. While the clearance is in the range [-0.06, -0.1], the roller loads are impacted lightly by the tilting moment. When nearly all of the rolling element contact the raceway, it is unexpected that maximum contact force instead of increasing with decreasing clearance. In this condition, clearance of -0.06 reaches the minimum contact force.

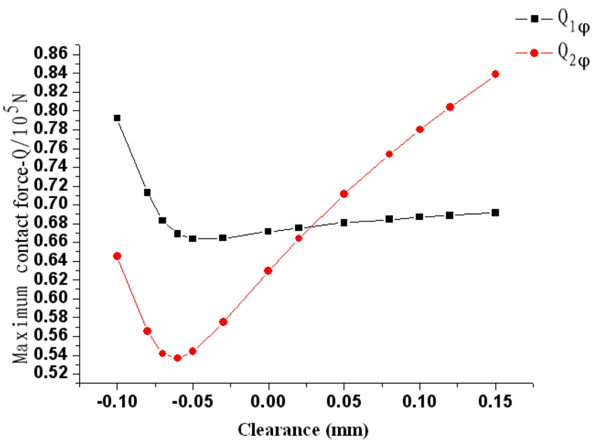


Fig. 5 Influence of clearance on maximum contact force

IV. CONCLUSIONS

With the application of slewing bearing in more and more new fields, accurately and deeply understand the performance parameters and their influencing factors becomes more and more important, while this problem can not be solved well by using traditional rolling bearing analysis theory. In this paper, the influence of bearing clearance on the contact force of raceway was researched. This was realized by building the static model considering clearance of the bearing firstly; then the internal loads distribution of the bearing was obtained by solving this static problems and the influence of the clearance on maximum contact force was researched further. Based on the analyzed results, we can conclude that the curve of the load distribution along the raceway is basically in the same trend with different clearance, with the clearance decrease toward to negative direction, more and more rolling element began to undertaker external load and the load distribution curve is not so steep. What is more, with the clearance reduction, the maximum contact force of the bearing is reduced, while it is unexpected that contact force reduced to a certain value then increases drastically. If an optimums clearance is selected, it is propitious to improve the carrying capacity of the bearing.

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