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Structure Design and Performance Analysis of High-Speed Miniature Ball Bearing

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Abstract. The working performances of miniature ball bearings are obviously affected by its' geometric structure parameters. In this paper, quasi-static analysis theory is applied in the design of miniature ball bearings. Firstly, it is studied the influence of geometry structure, preload and rotating speed on the dynamic performance of bearing. Secondly, bearing dynamic characteristics are analyzed which include the bearing stiffness and Spin to roll Ratio. Lastly, the contact stress and bearing life are calculated. The results indicate that structure parameters play an importance role in bearing's dynamic performances. Miniature ball bearings which have lager ball number, bigger ball diameter and smaller inner race groove radius can get better performances while velocity and preload have great impact on the bearing life. So that parameters of miniature bearing should be chosen cautiously.

1. Introduction

High speed miniature bearings are widely used in high speed instruments, which require high precision and long service life. In order to adapt to a variety of working conditions and satisfy interchangeability, the angular contact miniature ball bearings are often chose in these applications. The design of its geometric structure parameters and the selection of axial preload have close relationship with contract load, contract stress, service life and stiffness. The bearing performance can be affected by the subtle change of structure greatly due to its small geometry [1-2].

In structure design, machining method, performance analysis and measurement method of miniature bearing, many scholars have done a lot of researches. For calculating the critical speed of rotating machinery supported by miniature bearing, the stiffness of miniature bearings under various loads must be predicted first [3-5]. The friction torque is a very important performance of the miniature bearing, and the measurement of the torque is a difficult task in bearing test field Chen [6] has studied the miniature bearing and proposed that the better test data gathered must use more sophisticated test platform. In Gao's [7] paper the factors which cause the friction torque and the measuring methods for the friction torque of the miniature bearing are summarized and analyzed by him. Li [8] established an analysis modal for calculating high speed miniature ball bearing friction torque based on thermal elastohydrodynamic lubrication (TEHL), which has proved by using high precision miniature bearing testing machine. Zhang [9] and his team mainly analysis the design and manufacture of miniature air bearing, and explore the influence of microscale effect and slip flow effect on lubrication characteristics respectively. A new type of grinding process is proposed in Gui's [10] paper, which can effectively improve the wear resistant property of miniature bearing. A new kind of high strength alloy material is introduced by Tomasello [11], which can greatly improve the

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corrosion resistance and low temperature properties of miniature bearing. The damages due to contamination and high handling forces and preload options which are used to remove axial play and boost axial and radial stiffness are also presented [12]. According to Zhu's [13] paper the structure and design scheme of a closed–type miniature bearing are introduced, and the causes of the vibration of the bearing are also analyzed. Based on the quasi-dynamics of high-speed miniature ball bearing and the principle of energy conservation and taking the characteristics of lubrication oil and friction mechanism into account, a mathematic model of friction torque is established in Li's [14] paper. In this paper, based on the quasi-static model established by A B Jones [15], the calculation equations of miniature bearing parameters such as stiffness, life and contact stress are established.

In conclusion, it's difficult to design and test the performance of miniature bearing, and it is of great value to study the influence of miniature bearing geometry parameters on the contact load, contact stress and bearing life, stiffness. In this paper, the effects of miniature ball bearing geometric structure parameters such as ball number, ball diameter, contact angle, groove radius on bearing working performances are analyzed. Moreover, some suggestions are put forward on bearing parameter selection. Undoubtedly, it can provide guidance for the design of miniature bearings.

2. Analytical calculation model

According to the quasi-static model established by A B Jones, it is known that combine bearing's quasi-static equilibrium equation with displacement deformation compatibility equation, then iterate those equations can get many dynamic performance parameters, such as contact deformation, contact load, contact stress, ratio of revolution and rolling and bearing stiffness.

2.1. Contact stress and contact load of ball with ring

Using the iterative method to solve the quasi-static equilibrium equation and displacement-deformation compatibility equation of the bearing can get the contact deformation δ_{ij} of every ball. According to Hertz contact theory, the contact load between ball and bearing ring is

$$Q_{ij} = K\delta_{ij}^{\frac{3}{2}} \tag{1}$$

$$K = \frac{1}{\left(1/K_1\right)^{\frac{2}{3}} + \left(1/K_2\right)^{\frac{2}{3}}} \tag{2}$$

where Q_{ij} is contact load between the ball number j and raceway, for inner ring i =1, for outer ring i =2; K is Load displacement coefficient; K_1 is outer ring stiffness; K_2 is inner ring stiffness; δ_{ij} is contact deformation between the ball number j and rings.

The max contact stress between ball and bearing ring is

$$\sigma_{ij} = \frac{3Q_{ij}}{2\pi ab} \tag{3}$$

where a, b is long and short radius of Hertz contact ellipse respectively, the calculation process provided by Wang [16] and Harris [17].

2.2. Bearing stiffness

Putting the contact load Q_{ij}, gyro torque M_{gi} of balls and centrifugal force F_{ci} into force balance equation (forma4-6), can get the total bearing loads, which include axial load F_a,, radial load F_r and angular bending moment load M.

$$F_a = \sum_{j=1}^{z}\left[Q_{2j}\sin\beta_{2j} - \frac{2(1-\lambda)M_{gi}}{d}\cos\beta_{2j}\right] \tag{4}$$

$$F_r = \sum_{j=1}^{z}\left[Q_{2j}\cos\beta_{2j} + \frac{2(1-\lambda)M_{gi}}{d}\sin\beta_{2j}\right] \tag{5}$$

$$M = \sum_{j=1}^{z} \left[\begin{array}{l} Q_{2j} \sin \beta_{2j} \\ -\frac{2(1-\lambda) M_{gi}}{d} \left(R \cos \beta_{2j} - f_2 D_b \right) \cos \phi_j \end{array} \right] \tag{6}$$

where F_a, F_r is axial and radial load of bearing respectively; M_{gj} is gyro torque of j ball; β_{2j} is the contact angle between the j ball and the inner ring; φ_j is position angle of j ball; D_b is ball diameter; for outer ring λ=1, for inner ring λ=2.

$$[K] = \begin{bmatrix} \frac{\partial F_x}{\partial x} & \frac{\partial F_x}{\partial y} & \frac{\partial F_x}{\partial z} & \frac{\partial F_x}{\partial \theta_x} & \frac{\partial F_x}{\partial \theta y} \\ \frac{\partial F_y}{\partial x} & \frac{\partial F_y}{\partial y} & \frac{\partial F_y}{\partial z} & \frac{\partial F_y}{\partial \theta_x} & \frac{\partial F_y}{\partial \theta y} \\ \frac{\partial F_z}{\partial x} & \frac{\partial F_z}{\partial y} & \frac{\partial F_z}{\partial z} & \frac{\partial F_z}{\partial \theta_x} & \frac{\partial F_z}{\partial \theta y} \\ \frac{\partial M_x}{\partial x} & \frac{\partial M_x}{\partial y} & \frac{\partial M_x}{\partial z} & \frac{\partial M_x}{\partial \theta_x} & \frac{\partial M_x}{\partial \theta y} \\ \frac{\partial M_y}{\partial x} & \frac{\partial M_y}{\partial y} & \frac{\partial M_y}{\partial z} & \frac{\partial M_y}{\partial \theta_x} & \frac{\partial M_y}{\partial \theta y} \end{bmatrix} \tag{7}$$

where x, y, z is the three coordinate axes of the bearing.

Using the variables in formula (3-5) to taking the derivative of the displacement (δ_x、δ_y、δ_z、θ_x、θ_y) of the bearings on the five degrees of freedom, can get the relationship between bearing loads and displacement. The stiffness matrix of bearing on five degrees of freedom can be transformed into the above equation.

2.3. Spin to roll ratio

Spin to roll ratio on behalf the size of spin motion of ball on the ring. With the increasing of the ratio, the spin becomes more violent, the bearing generates more thermal, and also deteriorate the lubrication. According to A B Jones raceway control theory [18], the spin motion of the ball occurs only in the non-controlled raceway, so spin to rolling ratio of the ball in controlled raceway is 0, and in non-controlled raceway the ratio can be got through formula (8) and (9):

$$\omega_{si} = \left[-\omega_i \sin \beta_i + \omega_b \sin\left(\beta_i - \alpha \right) \right] \tag{8}$$

$$\omega_{roll} = -\omega_i \frac{D_m}{d} \tag{9}$$

where α is the angle between the ball spin axis and the central axis of the bearing; β_i is the contact angle of the ball; D_m is bearing pitch circle diameter; ω_i is the relative rotating speed between ring and cage, if the control ring is inner ring i = 1, if not i = 2.

2.4. Bearing fatigue life

The fatigue life of bearing is discrete, according to literature [3], it can be got from the following formula:

$$L_i = \left(\frac{Q_{ci}}{Q_{ei}} \right)^3 \tag{10}$$

$$Q_{ci} = 98.1 \left(\frac{f_i}{f_i - 1} \right)^{0.41} \frac{(1\mp}{(1\mp} \quad \frac{\gamma}{\mathrm{os}\, \beta_i} \Bigg)^{0.3} D_b^{1.8} Z^{-\frac{1}{3}} \tag{11}$$

$$Q_{ei} = \left(\frac{1}{z} \sum_{j=1}^{j=z} Q_j^{\frac{1}{e}} \right)^e \tag{12}$$

where Q_{ci} is the rated static load of the ring; Q_{ei} is working load of the ring; f_i is curvature radius of the ring; Z is the number of the ball; for the rotating ring e=1/3, for static ring e=0.3.

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2.5. Model verification
In order to verify the correctness of the model, it is used to calculate the bearing parameters in Li's paper and compare the result with it. Bearing parameters shown in table 1, bearing structure shows in figure 1 and the results are as table 2.

Table 1. Bearing parameters in Li's [19] paper

Parameters	Value
Number of balls	11
Diameter of ball (D_b/mm)	3.175
Pitch diameter of bearing (d_m/mm)	16
Contact angle (β/°)	15
Outer curvature coefficient ($f_1{}^{a}$)	0.55
Inner curvature coefficient ($f_2{}^{b}$)	0.53
Material	GCr15
Axil force P_r (N)	70

[a] f_1=R_1/D_b, [b] f_2=R_2/D_b
where R_1, R_2 is the bearing outer and inner raceway curvature radius respectively.

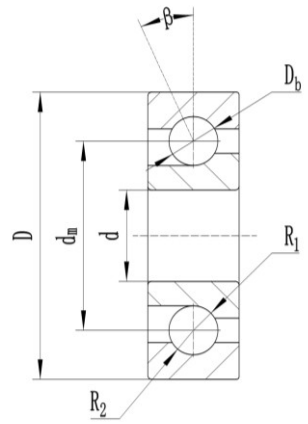


Figure 1. Schematic diagram of parameters

Table 2. Compare results of two papers

Parameters	Result of Li's paper	Result of this paper
Contact stress σ/(MPa)	1200	1372
Stiffness K/(N·mm)	22	21
Spin to roll Ratio	0.27	0.25
Life L/(h)	6000	6600

It can be seen from table 2 that the results of this modal are close to Li's study. So, it indicates that the modal in this paper is credibility.

3. Design calculation and result analysis
Using computer programmes above formula, take a miniature bearing for example whose parameters are shown in table 3. Material of the bearing is GCr15. When it's working, the rotate speed of outer ring is 36000r/min and the load is 4N, using constant pressure preload. Because of the radial load the stress condition of each ball is inconsistent, and the contact load, contact deformation, contact stress, Spin to roll ratio changes with ball position angle, so it is necessary to analyze and calculate the force of all balls. According to paper [20] the reasonable range of bearing parameters is shown in table 4.

Table 3. Geometric structures parameters of this example.

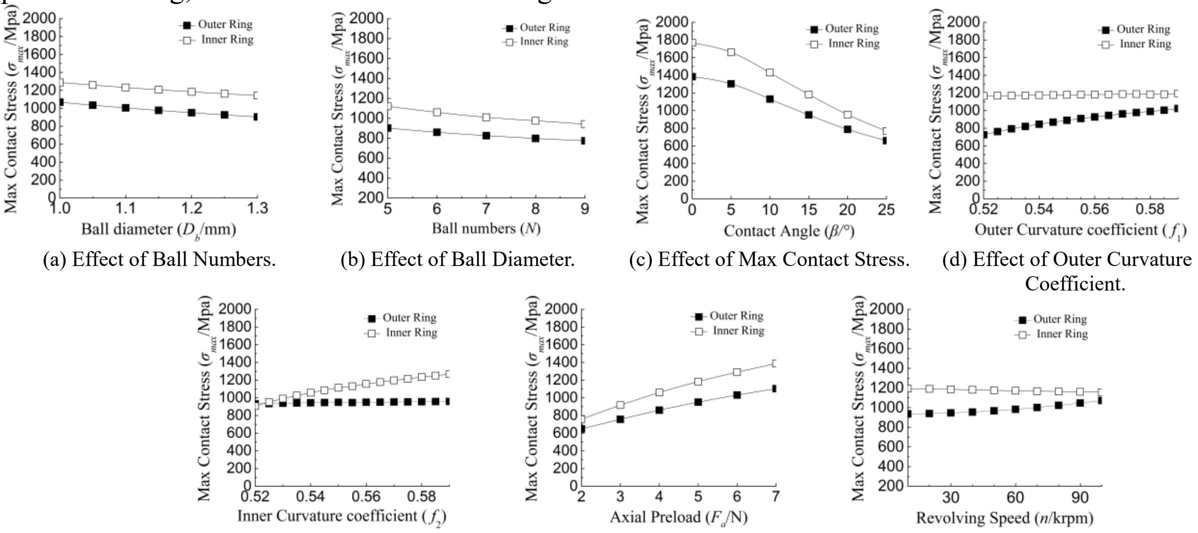
Parameter	Value
Number of balls	6
Diameter of bal (D_b/mm)	1.2
Pitch diameter of bearing (d_m/mm)	4.17
Contact angle (β/°)	15
Outer curvature coefficient (f_1)	0.5337
Inner curvature coefficient (f_2)	0.5667

Table 4. Reasonable range of each bearing parameter

Parameter	Value
Number of balls	5~9
Diameter of bal (D_b/mm)	1.0~1.3
Contact angle (β/°)	0~45
Outer curvature coefficient (f_1, f_2)	0.52~0.59
Rotational sp6eed (n/rpm)	10^5~10^4
Axial preload (F_a/N)	2~7

3.1. The max contact stress between ball and outer inner raceway
When the bearing in a high speed, centrifugal force makes the contact stress in outer raceway much bigger then it in inner raceway. But because the curvature coefficient in inner and outer raceway is different, the contact stress is related to rotate speed and geometric construction. Figure 2 shows the max contact stress change trend, out ring of the bearing rotate speed is 36000r/min while inner ring is static. As shown in Figure 1a and Figure 1b, when the pitch circle of bearing is constant, increasing the number of ball properly can reduce the contact stress, but if too many balls increased will decrease the ball diameter. It is not conducive to reducing ring contact stress. According to Figure 2c, the contact stress decreases with the increasing of contact angle. According to Figure 2d and Figure 2e, contact

stress in raceway is increased with the increase of its curvature radius. As shown in Figure 2f and Figure 2g, contact stress is increased with the increasing of axial preload. Meanwhile, as the rotate speed increasing, the contact stress in inner ring is reduced.



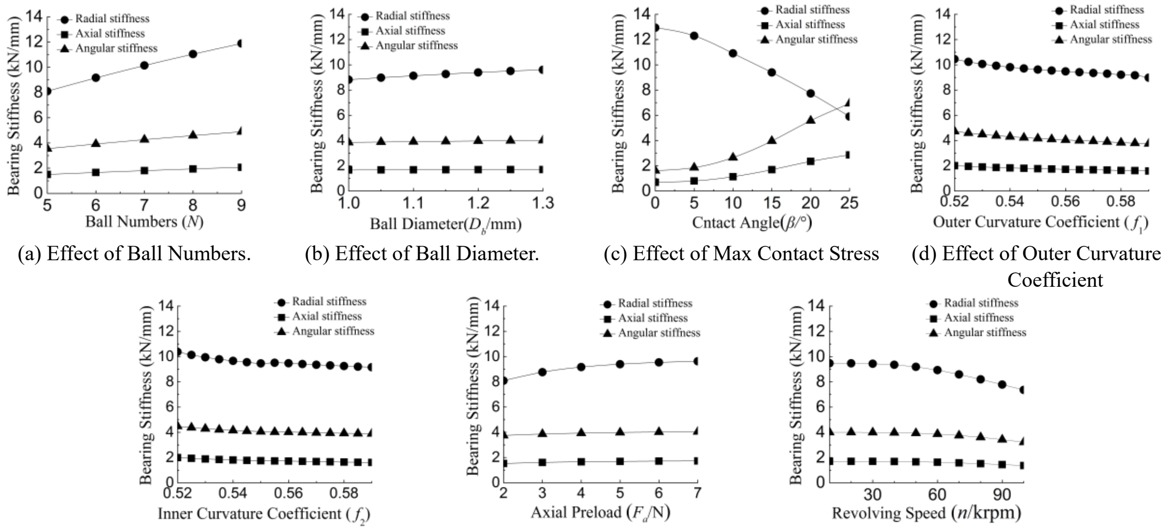
(a) Effect of Ball Numbers. (b) Effect of Ball Diameter. (c) Effect of Max Contact Stress. (d) Effect of Outer Curvature Coefficient.

(e) Effect of Inner Curvature Coefficient. (f) Effect of Axial Preload. (g) Effect of Revolving Speed.

Figure 2. The Max Contact Stress Change Trend Between Ball and Rings.

3.2. Bearing stiffness

According to Figure 3a and Figure 3b, stiffness increases with the increasing of ball numbers and diameter, especially radius stiffness. That because the more balls, the stronger bearing capacity. From Figure 3c, it can be known that with the increasing of contact angle, K_a and K_θ increase obviously, but K_r showed a downward trend. According to Figure 3d and Figure 3e, the bigger curvature radius coefficient is, the smaller the bearing stiffness becomes. According to Figure 3f and 3g, the bearing stiffness increases with the increasing of axial preload and the stiffness decreases with the increasing of rotational speed.



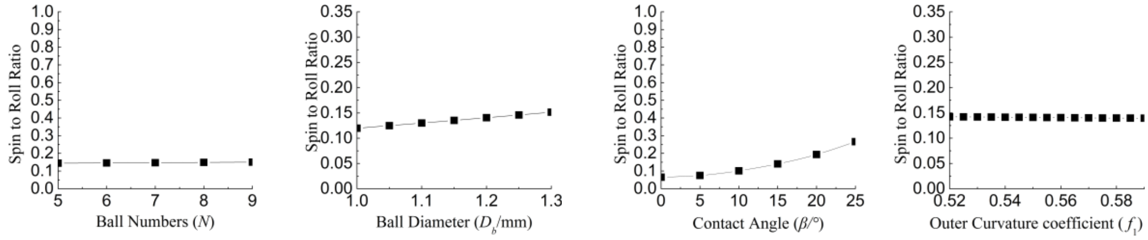
(a) Effect of Ball Numbers. (b) Effect of Ball Diameter. (c) Effect of Max Contact Stress (d) Effect of Outer Curvature Coefficient

(e) Effect of Inner Curvature Coefficient. (f) Effect of Axial Preload (g) Effect of Revolving Speed

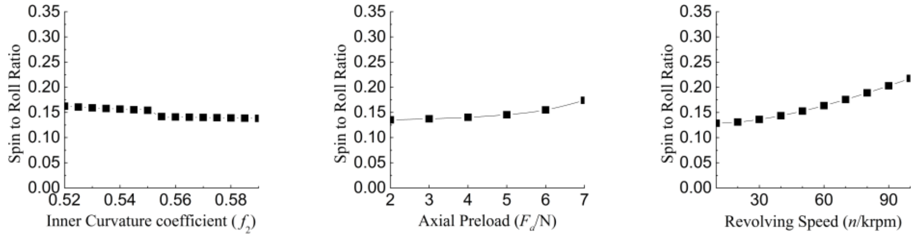
Figure 3. Stiffness Change Trend.

3.3. Spin to roll ratio

The influence of bearing's parameters to spin to roll ratio is showed in Figure 4. According to raceway control theory, spin exists only on the non-control raceway. Figure 4a and Figure 4b shows that ratio increases with ball numbers and ball diameter. According to the Figure 4c, ratio increases with the contact angle significantly. According to the Figure 4d and 4e, f_1 has little effect on the ratio, and with the increasing of f_2 ratio will decrease. According to the Figure 4f and 4g, decrease radial force can lower the ratio.



(a) Effect of Ball Numbers. (b) Effect of Ball Diameter. (c) Effect of Max Contact Stress. (d) Effect of Outer Curvature Coefficient.

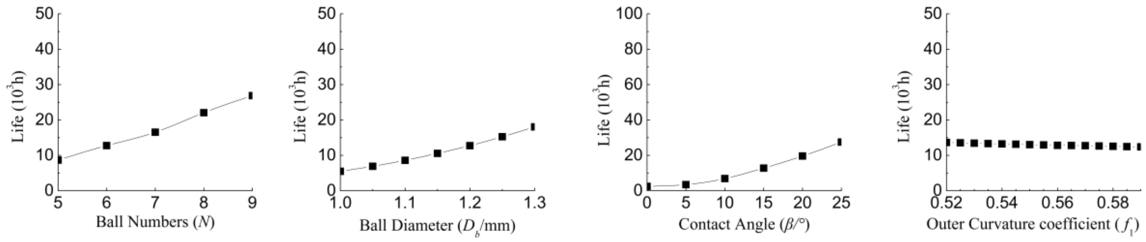


(e) Effect of Inner Curvature Coefficient. (f) Effect of Axial Preload. (g) Effect of Revolving Speed.

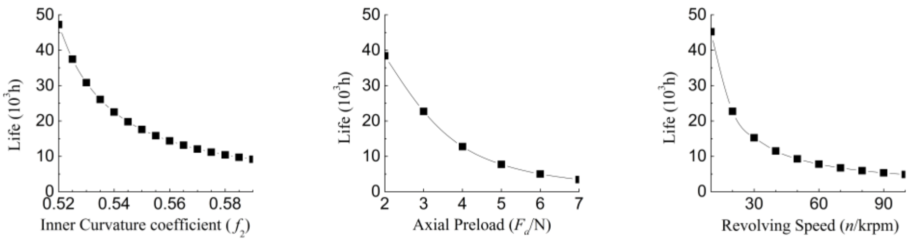
Figure 4. Spin to Rolling Ratio Change Trend.

3.4. Bearing fatigue life

Figure 5 is the influence of the variation of parameters on the fatigue life of bearing L. According to Figure 5a and 5b, increasing ball numbers and ball diameter can prolong the fatigue life of the bearing. According to Figure 5c, increasing contact angle appropriately bearing life will be extended. According to Figure 5d and 5e, increasing f_2 will lead to the reduction of bearing life, while f_1 has little effect on fatigue life. According to Figure 5f and 5g, increasing axial preload and rotate speed can reduce the fatigue life of bearing.



(a) Effect of Ball Numbers. (b) Effect of Ball Diameter. (c) Effect of Max Contact Stress. (d) Effect of Outer Curvature Coefficient.



(e) Effect of Inner Curvature Coefficient. (f) Effect of Axial Preload. (g) Effect Of Revolving Speed.

Figure 5. Bearing Fatigue Life Change Trend.

4. Bearing design

(1) It is hoped that a working bearing has small contact load and contact stress, low Spin to roll ratio, high stiffness and longer service life. According to the above analysis, to achieve those targets the ball number and the diameter should be larger, however, Spin to roll ratio will be higher and heat generation will be intensify. Because of the significant influence of ball numbers on the stiffness and life, as well as the ball diameter's significant influence on Spin to roll ratio, it should be appropriate when increases the number of balls and decreases ball diameter.

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(2) The bigger contact angle and the smaller contact stress, the longer bearing life will be. But small angle can reduce the bearing ratio and increase the radial stiffness. According to Shun's [21] paper, when the contact angle is 15 degrees, the friction torque of the bearing is the smallest. According to the study of Yang [22], when the miniature bearing is loaded, the contact angle will increase significantly compared with the normal bearing, so bearing contact angle should be a small value, the range from 10 degrees to 25 degrees will be better.

(3) Raceway curvature radius coefficient changes will lead to the change of control ring, and the increase of the curvature radius will enlarge the contact stress, reduces the stiffness and shortens the life. According to Deng [23] and Wang's [24] research, a smaller radius of curvature of the inner raceway contributes to the formation of a thicker lubricant film, and bearing's friction is small, which contributes to the formation of elastohydrodynamic lubricating film in its interior. Therefore, it is necessary to select a relatively small raceway curvature radius, and curvature radius of the inner raceway should be smaller than the outer.

Based on the above design principle, the improved parameters are shown in table 5. The table 6 shows the comparison of bearing performance parameters before and after improvement. Despite the slight increase of the contact stress and the Spin to roll ratio, the other parameters have been improved considerably.

Table 5. Parameters of improved bearing

Parameters	**Value**	
	Scheme1	**Scheme2**
Number of balls	7	8
Diameter of ball (D_b/mm)	1.2	1.0
Pitch diameter of bearing (d_m/mm)	4.16	4.16
Contact angle (β/°)	15	15
Outer curvature coefficient (f_1)	0.58	0.58
Inner curvature coefficient (f_2)	0.53	0.53

Table 6. Bearing performance comparison before and after improved

Parameters	**Before improved**	**After improved**	
		Scheme 1	**Scheme 2**
Contact stress σ/(MPa)	Outer 859.9 Inner 1060.5	Outer 940.6 Inner 952.8	Outer 1013 Inner 993.9
Stiffness K/(N·mm)	Axial 1662.0 Radial 9166.5 Angle 3916.9	Axial 1972.1 Radial 10832 Angle 4457.7	Axial 2113.7 Radial 11313 Angle 4733.9
Spin to roll Ratio	0.15	0.16	0.13
Life L/(h)	12751	14033	7953.7

5. Conclusions

(1) The performance of miniature bearing is greatly influenced by its internal geometry parameters, so the geometric parameters of each part should be selected reasonably. When the outer ring rotates, the appropriate increase of the number of balls can be considered, and the appropriate contact angle should be chosen.

(2) The stiffness of bearing can be improved by increasing the axial preload. At the same time, the contact load and contact stress will increase, bearing life will reduce. So when design bearings the requirement of bearing stiffness and life must be taken into account.

(3) Because of the ball centrifugal force, high rotating speed makes the bearing contact stress in outer ring increases obviously, and decreases in the inner ring which reduces bearing stiffness and life. The spin motion of the ball on the raceway is aggravated and the heating is serious, which should pay attention to when design bearings.

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