Crown Analysis of Needle Roller Bearing Based on Finite Element Method

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Abstract: This paper is aimed to investigate the crown characteristics of needle roller bearing. The internal load distribution of needle roller bearings under radial load is calculated by empirical formula. Considering the maximum load of the bearing monomer as the initial boundary conditions, the single contact model of needle roller bearing is established via finite element method. A typical needle roller bearing is conducted as an example to validate the present model by comparing the contact stress of straight generatrix shape. The influences of the modified length and depth are discussed. The obtained results indicate that an appropriate selection of modification length and depth, where the modified length is 10% of the roller and the depth is the elastic deformation of the bearing, can obtain desirable modified properties. Meanwhile, the present method can provide an efficient tool for the design of needle roller bearings.

Key words: Needle Roller Bearing; Crown Design; Contact Stress; Finite Element Method

1 Introduction

Needle roller bearing with smaller cross-sectional area of roller is a special cylindrical roller bearing and suitable for the supporting structure with limited size of radial installation [1]. In radical needle roller bearing, the needle rollers are pressed into the raceway under loading, which results in non-uniform pressure distributions in contacts characterized by a pressure increase in the end of the contact line [2], even lead to a decrease of the bearing fatigue life. Therefore, the crowning design of roller to improve the contact condition between roller and rings has attracted substantial attention to increase the fatigue life of needle roller bearing [3].

In recent years, many researchers devote themselves to researches of the roller modification. An early study about the dependence of “edge effect” was presented by Lundberg using rollers with basic crowned profile. Later, Johns [4] extended the Lundberg approach by proposing an improved profile modification. However, it was

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difficult to obtain uniform contact pressure distribution, especially under heavy load and tilting moment conditions. Horng [5] provided a deformation formula for a circular crowned roller and analyzed the comparison between crowned and non-crowned parts. Urata [6] proposed a crowning shape formed by combining two or more circular arcs. Edge stress disappeared in the crowning shape, but misalignment could not be considered in the model. Later, Hiroki [7] proposed a new logarithm profile function. Three design parameters were introduced into the profile, and the profile could prevent edge stress due to misalignment. However, these parameters are not easy to choose and only be optimized in a small range, thus the profile is not used widely in the roller bearing design. So far, the roller crown shapes mainly include arc and logarithmic profiles [8]. In particular, logarithmic profile is considered to the best profile modification method in theory. However, the processing accuracy of various surfaces is higher and even to the micron scale in practical engineering applications, which leads to logarithmic profile modification is limited[9]. Therefore, the main way to crown is arc profile modification. The present researches are mainly focused on the logarithmic profile modification of roller bearings. It is urgent to carry out researches about arc profile modification of needle roller bearing. Finally, the appropriate amount of arc profile modification can be determined.

The paper developed a three-dimensional finite element model and the way of arc profile modification of roller was selected to research the contact stress distribution by the model. Then, the influence of modified length and depth are all discussed. Finally, the desirable arc profile modified values are given by finite element analysis.

2 Load distribution of needle roller bearing

The maximal load of needle roller can be obtained based on the classical mechanics theory [10], which provides the initial load conditions for the following finite element analysis. Supposing that the lowest needle roller is numbered 0, followed by numbers 1, 2, 3 and so on. The load distribution of needle roller bearing under the radial force is shown in Fig.1.

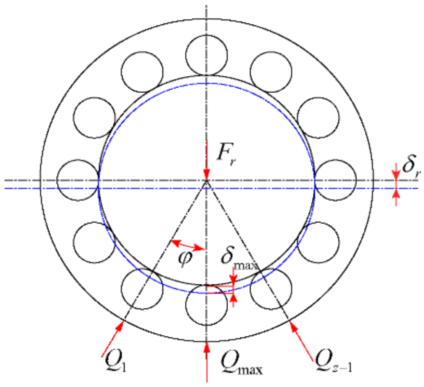

Fig. 1. Force model of needle roller bearing

From the force model of needle roller bearing shown in Fig.1, which can easily obtain the equilibrium equations for the bearing as below:

$$F_r = ZK_n\left(\delta_r - \frac{1}{2}P_d\right)^n J_r(\varepsilon) \tag{1}$$

Where Z is number of needle rollers, n is exponent in the equation determining contact mode, δ_r is elastic deformation, P_d is radial clearance of the bearing, $J_r(\varepsilon)$ is load distribution integral of bearing, K_n is total load deformation coefficient between the roller and ring, which is calculated by K_i and K_e.

$$K_n = \frac{1}{\left[\left(1/K_i\right)^{1/n} + \left(1/K_e\right)^{1/n}\right]^n} \tag{2}$$

$$K_i = K_e = 7.86\times10^4 L^{8/9} \tag{3}$$

The ε is the range parameter of load distribution related to $J_r(\varepsilon)$ in Equation.(1), described as:

$$\varepsilon = \frac{1}{2}\left(1 - \frac{P_d}{2\delta_r}\right) \tag{4}$$

The contact force between the roller and ring is calculated from the Hertz contact formula as follows:

$$Q_{\max} = K_n\delta_{\varphi=0}{}^n = K_n\delta_{\max}^n = K_n\left(\delta_r - \frac{1}{2}P_d\right)^n \tag{5}$$

$$Q_\varphi = Q_{\max}\left[1 - \frac{1}{2\varepsilon}\left(1-\cos\varphi\right)\right]^n \tag{6}$$

The central of contact stress of needle roller and ring can be expressed as:

$$P = \frac{2Q_{\max}}{\pi lb} \tag{7}$$

The overall solution process for the bearing load equations is demonstrated in Fig.2 for the bearing monomer load as inputs.

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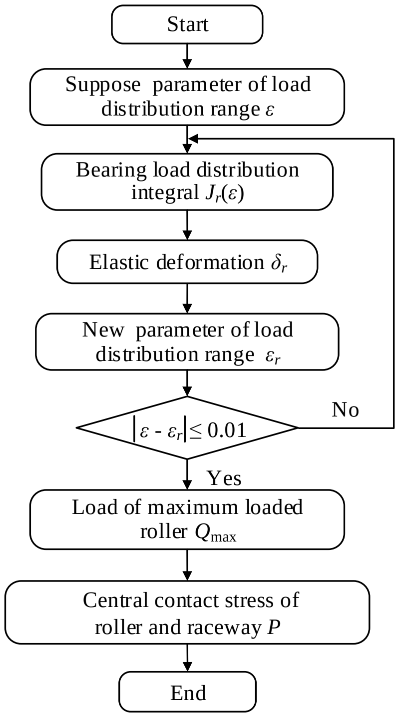

Fig. 2. Flow chart of calculation

The other outputs such as the bearing monomer maximum load and contact forces between roller and races are saved for the later roller contact stress distribution analysis. Fig.3 shows an example of the computational results for a certain type of needle roller bearing with its properties given in Tab.1. The calculated program is established to analyze the load distribution of loaded roller under the limited load 300kN and the referenced load 200kN. Then, the maximum loads of the rollers are 11969N and 8669N, respectively.

Table 1. Structural dimension parameters of bearing

Parameter	Unit	Value
Bore diameter	mm	300
Outer diameter	mm	336
Diameter of inner ring raceway	mm	312
Diameter of outer ring raceway	mm	324
Bearing total width	mm	60
Number of rollers	-	146
Roller length	mm	40
Roller diameter	mm	6
Radial clearance	mm	0.06

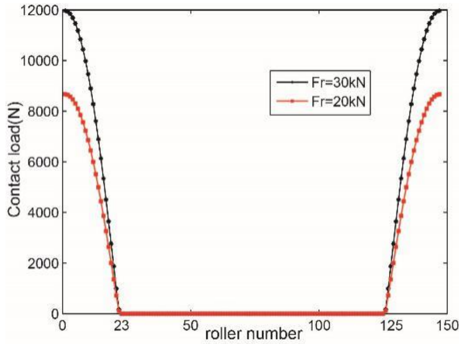

Fig. 3. Load distribution of needle roller bearing

As is seen from Fig.3, the contact loads of roller and raceway are symmetrical distribution. The rollers in the upper part of bearing are no-loaded under the radial loads of 300kN and 200kN, which numbered 23-125. The bearing has a total of 46 loaded rollers, of which the largest loaded roller is the No.0 roller. Then, with the change of No.0 to No.22, the load gradually decreases, otherwise, the load increases with the change of No.126 to No.145.

3 Finite element analysis of needle roller bearing

3.1 Finite element model of needle roller bearing monomer

The finite element model of bearing largest loaded monomer is selected so as to save time and cost, of which sealing structure and chamfer is be ignored. The contact performance of the bearing monomer is analyzed by ANSYS. The specific steps are as follows:

a) Defining the material properties. The Solid45 element is selected for solid modeling, the elastic modulus is 206 GPa, the poisson’s ratio is 0.3.

b) Meshing. The method of grid mesh generation is variable size mesh method using grid encryption. The model of needle bearing and the result of finite element mesh division are shown in Fig.4 by getting 302360 units and 326312 nodes.

c) Setting contact pairs. The contact pairs between roller and raceway are formed by using Contact174 unit and Target170 unit. The friction coefficient is 0.15, and the contact stiffness is 1.0.

d) Applying load and constraint on the bearing monomer. A full constraint is applied to outer surface of the bearing outer ring by considering features of installation between bearing and bearing housing. All nodes on inner surface of the inner ring are coupled and loaded. All nodes on the contact line of roller and ring are constrained by the circumferential and axial direction.

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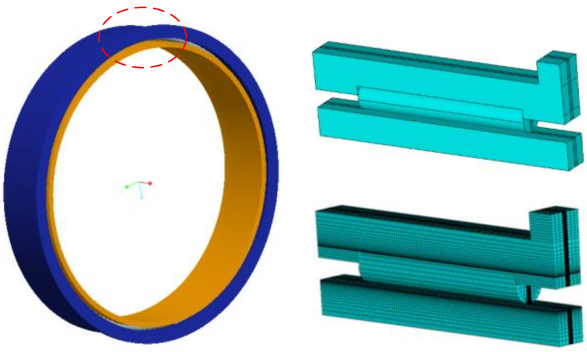
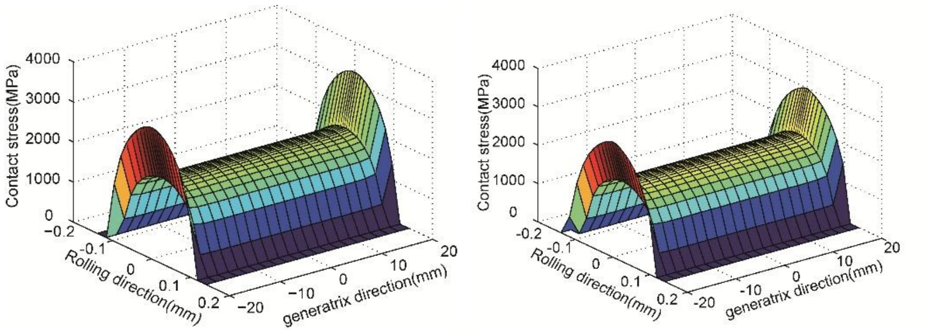


Fig. 4. Finite element model of needle roller bearings

3.2 Verification of finite element model

Based on the finite element model established in the last section, the change of contact stress between roller and raceway is shown in Fig.5. It can be seen that there is obvious stress concentration on both contacted ends of roller and raceway, and uniform stress distribution on contacted center of roller and raceway. Between needle roller and inner raceway, the maximum contact stress is 3290MPa, the central contact stress is 1912MPa. Similarly, between needle roller and outer raceway, the maximum contact stress is 2994MPa, the central contact stress is 1875MPa. Therefore, it is necessary to optimize roller generatrix to reduce the stress concentration effectively.



(a) Inner raceway (b) Outer raceway

Fig. 5. Contact stress of roller and raceway

In order to verify the effectiveness of the bearing monomer finite element model, the contact stresses of roller and raceway are compared by theoretical calculation and finite element method. It can be seen from Tab.2 that the contact stress of roller and outer raceway is less than inner raceway. The results of finite element analysis are consistent with the theoretical calculation with the maximum error is 0.27%. Thus, the finite element model is verified.

Table 2. Contact stress between needle roller and inner or outer raceway

	Theoretical calculation （MPa）	Finite element analysis （MPa）	Error (%)
Contact stress of roller and outer raceway	1880	1875	0.27
Contact stress of roller and inner raceway	1916	1912	0.21

3.3 Verification of finite element model

An obvious stress concentration on both contacted ends of roller and raceway is emerged without modification. In order to reduce “edge effect”, the arc profile modification is adopted in engineering. The modeling process of bearing monomer with arc profile modification is consistent with unmodified monomer. Unlike the unmodified monomer, the location of refine grid is the area of modification and contact. Then, the bearing monomer is divided into 587457 units and 596783 nodes. The finite element model of the bearing monomer with arc profile modification is shown in Fig.6.

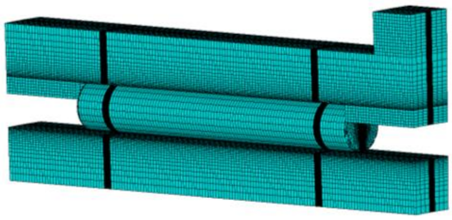


Fig. 6. Finite element model

The contact stress of roller and raceway under arc profile modification is further analyzed. Fig.7 (a) shows that the maximum contact stress is 2352MPa and central contact stress is 2120MPa between roller and inner raceway. Similarly, it’s seen from Fig.7 (b), the maximum contact stress is 2360MPa and the central contact stress is 2060MPa between roller and outer raceway. Compared with the bearing with straight generatrix shape roller, the contact stress of roller and raceway of bearing with arc profile modification decrease significantly.

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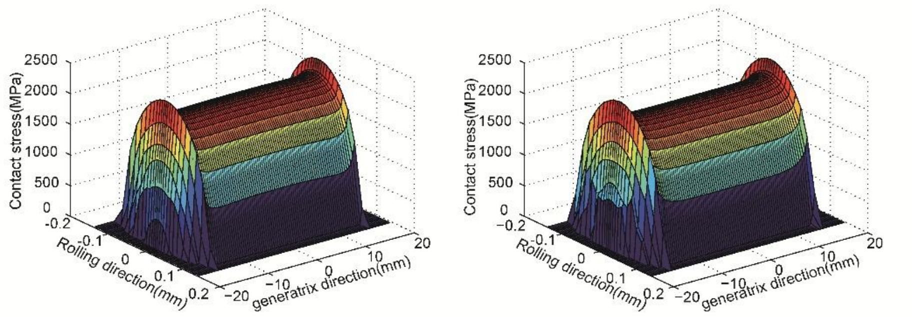


(a) Inner raceway (b) Outer raceway

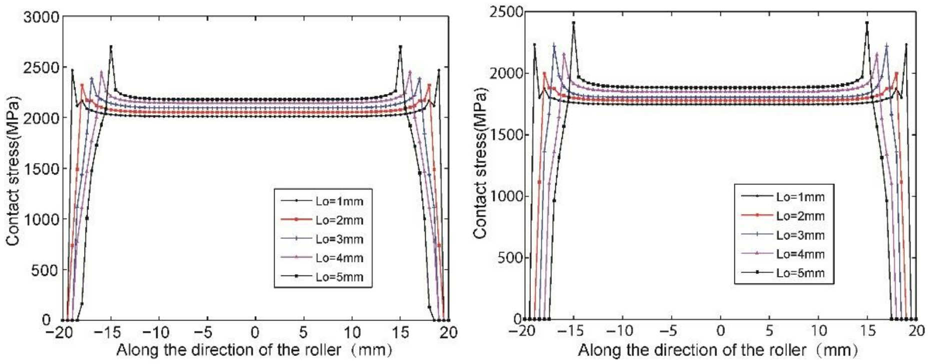
Fig. 7. Contact stress of roller and raceway

4 The determination of the optimal modified value for arc profile modification

It is suggested that the modified variables of roller should be considered in the design of arc profile modification, which include the radius of drum (R_o), the modified length(L_o) and the modified depth(h_o). In the chapter, the modified length and depth of arc profile modification are taken as the design variables. Considering the changes of contact stress under different radial loads and modified variables, the optimal modified values of roller are obtained under arc profile modified mode.

4.1 Determination of the optimal modified length

The load of needle roller bearing affects the internal loaded distribution and contact stress of roller. Fig.8 shows the contact stress distribution of maximum loaded roller along the roller length for the type of arc profile modification under the same modified depth and different loads. Fig.8 (a) shows that the central stress values of roller and outer raceway increase gradually and the edge stress values decrease initially and then increase with the increase of modified length. Fig.8 (b) shows that the central stress values of roller and outer raceway also increase gradually and the edge stress values are larger with modified length of 1mm and 3mm than others with the increase of modified length.



(a) Radial load 300kN (b) Radial load 200kN

Fig. 8. Distribution of contact stress of roller and outer raceway under different modified length

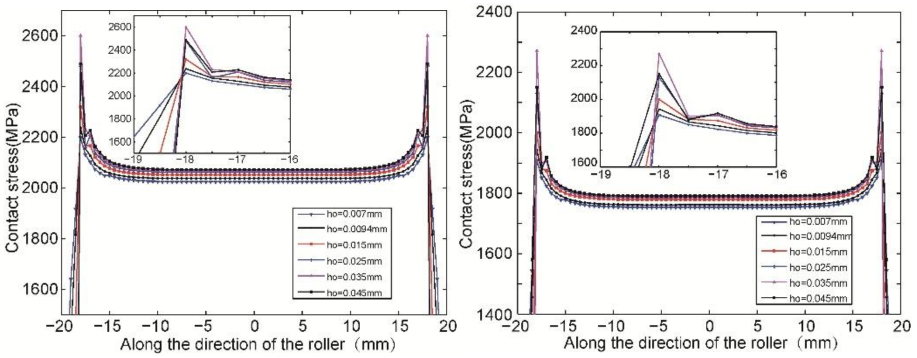
The specific values of contact stress of roller and outer raceway with different modified length are shown in Tab.3. It shows that the change rates are fluctuating under radical load. However, the modified length of crown roller which has the minimum change rate in the contact stress distribution analysis is 2mm. Then, the "edge effect" of the needle roller bearing can be reduced to the maximum.

Table 3. Comparative analysis of contact stress under different modified length

Radial load （kN）	Modified length(mm)	Maximum stress (MPa)	Central stress (MPa)	Change rate (%)
300	1	2467	2012	18.4
	2	2319	2052	11.5
	3	2384	2094	12.2
	4	2445	2143	12.3
	5	2700	2178	19.3
200	1	2233	1747	21.7
	2	2000	1778	11.1
	3	2220	1804	18.7
	4	2154	1849	14.1
	5	2410	1883	21.8

4.2 Determination of the optimal modified depth

The results of contact stress of roller and outer raceway are analyzed under the same modified length and different radial load in Fig.9. It can be seen that it is little different on the values of central contact stress under different modified depth. However, the edge stress values gradually increase with the increment of modified depth. In summary, the influence of modified depth is less than modified length on contact stress of roller and outer raceway.



(a) Radial load 300kN (b) Radial load 200kN

Fig. 9. Distribution of contact stress of roller and outer raceway under different modified depth

The values of maximum and central contact stress of roller and outer raceway are further extracted to compare the change rate. It can be seen from Tab.4 that the modified depth of crown roller which has the minimum change rate in the contact stress distribution analysis is 0.0094mm under radial load of 300kN, and the minimum change rate is 0.007mm under radial load of 200kN. In other words, the suitable depth of crown roller is elastic deformation of bearing under different radial loads.

Table 4. Comparative analysis of contact stress under different modified length

Radial load（kN）	Modified depth(mm)	Maximum stress (MPa)	Central stress (MPa)	Change rate (%)
300	0.007	2212	2010	9.1
	0.0094	2237	2038	8.9
	0.015	2319	2052	11.9
	0.025	2480	2062	16.8
	0.035	2600	2067	20.5
	0.045	2489	2072	16.7
200	0.007	1907	1753	8.1
	0.0094	1943	1761	9.4
	0.015	2000	1778	11.1
	0.025	2130	1785	16.2
	0.035	2270	1788	21.2
	0.045	2150	1790	16.7

In summary, the modified values have a significant influence on contact stress of roller and outer raceway. Especially, the modified depth has a bigger impact. An appropriate selection of modification length and depth, where the modified length is 10% of the roller and the depth is the elastic deformation of the bearing, can obtain desirable modification properties. Then, the modified values reduce the "edge effect" of needle roller bearing to the greatest extent. The present results can provide an efficient tool in the design of needle roller bearings.

5 Conclusions

In the paper, a three-dimensional finite element model of needle roller bearing with arc profile modification is established. The distribution of contact stress roller and raceway is analyzed and compared with different modified values. Then, the best values of arc profile modification are obtained. The following conclusions can be drawn from the present results：

(1) There is “edge effect” on both contacted ends of roller and raceway with straight generatrix shape. The arc profile modified model can change crown value of roller and also allow a straight portion on the roller, which can avoid “edge effect” and ensure uniform pressure distribution.

(2) The modified values directly affect the contact stress of needle roller bearing, the optimum modified values of arc slope generatrix shape through the analysis are obtained. When the modified length is 10% of the roller and the depth is the elastic deformation of the bearing, it can obtain desirable modified properties, which reduce the "edge effect" of needle roller bearing to the greatest extent.

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