Friction Properties of Axlebox Bearing Based on Quasi-static and Elastohydrodynamic Lubrication Theory

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Abstract: Axlebox bearing is a key component of the high-speed rail train, its friction performance affects the running state and safety of the train, the most common faults such as hot shafts, combustion shafts, etc. are closely related to friction. We established a roller-raceway contact stress and contact load calculation model based on Hertz contact theory and roller slice method in this paper; and then the lubrication state was analyzed based on the elastic flow theory, The roller-raceway friction calculation model and roller end face friction calculation model were established based on the modified Stribeck curve, Finally, a theoretical analysis model of friction performance of axle box bearing based on quasi-static theory and elastic flow theory is established. We have obtained the parameters influence of the friction by the theoretical analysis model of the friction performance of the axlebox bearing.

Keywords: Quasi-static Theory,Elastohydrodynamic Lubrication Theory, friction performance

1 Introduction

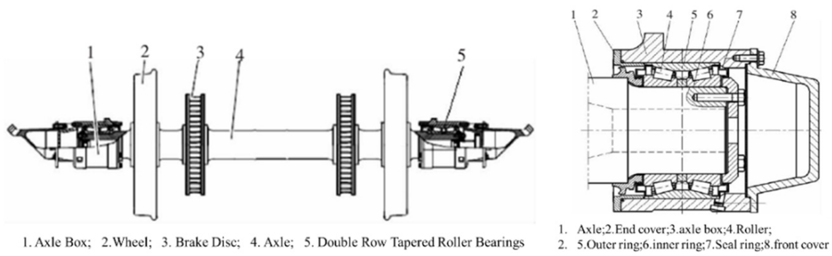

Fig. 1. Train wheel shaft system

Axlebox bearing is an important part of axle system of high speed train. In the running of the train, the bearing must bear the weight of the carriage, and also bear the axial force caused by the super high phenomenon during the bending, as well as the dynamic impact load caused by the track irregularity and the rail junction. The

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wheel shaft system of the high speed train is shown in Figure 1. The system consists of axles, wheels, brake discs and axle boxes on the both sides of the axle, and the bearings are installed in the axle box.

High-speed axle box bearings are mainly divided into cylindrical roller bearings, spherical roller bearings, needle roller bearings and tapered roller bearings. Because the tapered roller bearings have good radial, axial and tilting carrying capacity, so it is widely used, its structure mainly consists of outer ring, inner ring, rollers, cage and seals [1].

The study of the friction torque characteristics of rolling bearings began in the 1950s, Palmgren A [2] used the test method to find the bearing friction torque caused by all other mechanical friction except the lubricant, and obtained a formula for calculating the friction torque of the bearing.Snare B[3-5], Kakuta[6]analyzed the parameters of the rolling bearings caused friction torque on the basis of quasi - static analysis, they use the energy conservation principle to establish the formula and verify it by experiment. Townsend D P [7] studied the bearing friction torque characteristics of the ball bearing under several lubrication conditions, they carried out theoretical studies and tests to verify the relationship between bearing lubrication and friction torque.Todd M J [8] carried out theoretical studies and tests to verify the relationship between curvature radius coefficient of diagonal contact ball bearing and bearing friction Moment. Kitahara Tokio [9] established the angular contact ball bearing friction torque empirical formula in the radial and axial load on the basis of the test of the friction torque of precision ball bearings. Yokoyama Kazuhiro et al. [10] established a theoretical calculation formula for the friction torque of angular contact ball bearings containing bearing speed, preload and lubricating oil. Rodionov E M [11] analyzes the influence of the ball bearing ring and the ball rolling surface error on the friction torque, they found that the shape error the of rolling surface will cause the bearing friction torque fluctuations.

In our country, the research on the friction mechanics of rolling bearing has aroused the attention of the majority of scholars. Deng Sier et al [12] got the internal energy loss of the rolling bearing in operation based on the pseudo-dynamic analysis of the bearing, and calculated the friction torque of the bearing. Yang Peiran et al. [13] proposed a new numerical method, it can achieve high accuracy in consideration of the unsteady conditions of the load conditions and the non-Newtonian factors of the lubrication medium. Tang Peng [14] proposed a new bearing friction torque calculating method combined with high-side contact of the movement, film thickness ratio, lubrication state, which can accurately estimate the effect of the load condition on the friction torque. Zhu Aihua [15] contrasted the application scope and accuracy of SKF torque calculation model and traditional empirical calculation model, and discussed the effect of load condition on friction torque.

In summary, in our country and abroad, researchers had done a series of studies on the axle box bearing friction, but there are still some deficiencies: On the one hand, the classical theory of rolling bearing friction is based on deep groove ball bearings, which can not be used in double row tapered roller bearings; on the other hand, the friction and friction torque models are based on experimental data fitting or empirical formulas, which lack systematic theoretical analysis. Therefore, the friction performance of the high-speed train axle box bearing has important academic and engineering significance.

In this paper, we analyze the load distribution of high-speed axle box bearing, and then calculate the friction force of roller with inner raceway, the friction force of roller with outer raceway and the friction force of roller ball is based on quasi-static method, finally we analyzed the factors that affect the friction.

2 Load distribution analysis

Bearing deformation occurs under the action of the load, but the deflection angle generated by the overturning moment are ignored, and the bearing inner and outer ring as a whole body. The deformation is divided into axial deformation δa and radial deformation δr.

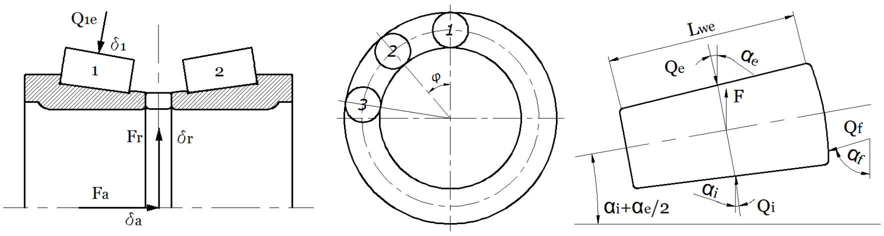


Fig. 2. Mechanical equilibrium diagram of bearing

The roller deformation in the normal direction of the outer raceway is：

$$\begin{cases} \delta_{1i} = \delta_r T\cos\alpha_e + \delta_a \sin\alpha_e + c\sin\alpha_e \\ \delta_{2i} = \delta_r T\cos\alpha_e - \delta_a \sin\alpha_e + c\sin\alpha_e \end{cases} \tag{1}$$

$$T = \begin{cases} \cos(i-1)\varphi; (\cos(\mathrm{i}-1)\varphi > 0) \\ 0; (\cos(\mathrm{i}-1)\varphi \le 0) \end{cases} \tag{2}$$

where, δ_{1i} is the raceway normal direction deformation of the corresponding roller in first row. Z is the number of rollers of single row. α_e is the contact angle of bearing outer ring. φ is the roller azimuth. C is the initial clearance.

$$\begin{cases} Q_{1i} = \begin{cases} K_l \delta_{1i}^{\ 1.11}; (\delta_{1i} > 0) \\ 0; (\delta_{1i} \le 0) \end{cases} \\ Q_{12} = \begin{cases} K_l \delta_{2i}^{\ 1.11}; (\delta_{2i} > 0) \\ 0; (\delta_{2i} \le 0) \end{cases} \end{cases} \tag{3}$$

$$K_l = 8.059 \times 10^4 L_{\mathrm{we}}^{8/9} [1 + C^{9/10} \cos(\alpha_e - \alpha_i)]^{-10/9} \tag{4}$$

$$C = \lambda \sin(\alpha_e + \alpha_f) / \sin(\alpha_i + \alpha_f) \tag{5}$$

ISSN 2572-4975 (Print), 2572-4991 (Online)

where, K_l is the contact stiffness coefficient of the roller and the outer ring. λ is the correction factor.

The force balance equation for the roller is:

$$\begin{cases} Q_{fj}\sin\alpha_f + Q_{ij}\sin\alpha_i - Q_{ej}\sin\alpha_e = 0 \\ Q_{ej}\cos\alpha_e + Q_{fj}\cos\alpha_f - F_j - Q_{ij}\cos\alpha_i = 0 \\ F_j = 3.39\times10^{-11} D_b{}^2 L_{we} n^2 D_w \end{cases} \tag{6}$$

where, L_{we} is the effective contact length of the roller and the outer raceway. D_b is the average diameter of the tapered roller. D_w is the rotation diameter of the contact area between the roller and the outer raceway. n is the bearing rotation speed.

The bearing force balance equation [16] is:

$$\begin{cases} F_a = \sum_{i=1}^{Z} Q_{1i}\sin\alpha_e - \sum_{i=1}^{Z} Q_{2i}\sin\alpha_e \\ F_r = \sum_{i=1}^{Z} Q_{1i}\cos\alpha_e \cos(i-1)\varphi + \sum_{i=1}^{Z} Q_{2i}\cos\alpha_e \cos(i-1)\varphi \end{cases} \tag{7}$$

By using the Newton iteration method, the contact deformation between the roller and the outer raceway is obtained, and the load distribution inside the high-speed train axle box bearing is obtained.

2.1 Friction analysis of roller and raceway

The characteristic roughness is considered into the dynamic pressure parameter SN to obtain an improved Sribeck curve.

$$S_N = \tau' \frac{\eta U_e}{q} \tag{8}$$

$$R_a' = C^{-1/(4-2D)} \tag{9}$$

$$U_1 = \frac{d_m}{2}(1-\gamma)\omega[1+\frac{1}{2}(1+\gamma)] \tag{10}$$

$$U_2 = \frac{D}{2}\cos\beta\cdot\frac{1}{2}\frac{(1-\gamma^2)}{\gamma}\omega \tag{11}$$

where, η is the viscosity of the lubricating oil [17]. U_e is the roll speed of the roller and the raceway. q is the unit area load. $\tau'=1(1/R_{a1}'+1/R_{a2}')$, R_{a1}', R_{a2}' are the characteristic roughness of the two friction surfaces. C is the measure coefficient. D is the rough surface fractal dimension. ω is the angular velocity of the bearing inner ring. B is the roller half cone angle. dm is the bearing pitch circle diameter. D is the curvature radius of the contact point. $\gamma=D/d_m$.

The friction force of the roller slices in the mixed lubrication state is:

$$F_f = \phi Q \tag{12}$$

where, Q is the contact load between the roller and the raceway.

2.2 Friction analysis of roller end face and inner ring flange

The bearing roller end and the inner ring are fully fluidized and lubricated so that a continuous oil film can be formed. The friction between the roller end face and inner ring flange [18] is:

$$F = \eta \frac{u_1 - u_2}{h_{\min}} \pi ab \tag{13}$$

Where: η is the dynamic viscosity of the lubricating medium. u is the flow velocity of the fluid in the y direction. τ is the shear stress of the two contact surfaces. a, b are the long and short half axes of the contact area, respectively.

2.3 Axle box bearing friction torque of high-speed train

We get the friction torque formula of the bearing according to the friction model of the roller and the outer raceway, the roller and the inner raceway, the roller end face and the inner wall [19].

$$M = \sum M_h \tag{14}$$

$$M_h = \sum_{n=1}^{21} F_{hn} l_{hn} \tag{15}$$

Where: M is the total friction torque of the axle box bearing. M_h is the friction moments of the corresponding part($h=i,e,f$ represents the inner raceway, the outer raceway and the ribs, respectively). F_{hn} is Roller friction. l_{hn} is the average turning radius of the contact area.

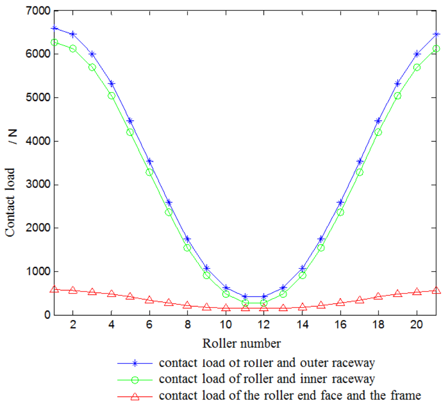
3 Engineering Applications

According to the friction torque model of the high-speed train axle box bearing, we calculate the friction torque of CRH3 high-speed train axle box bearing [20]. Bearing parameters as shown in Table 1.

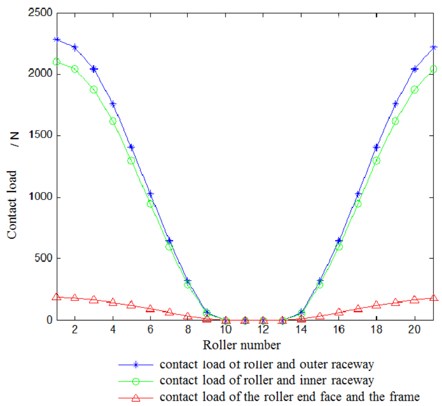
Tab. 1. Parameters of CRH3 high-speed train axle box bearing

Parameters	value	Parameters	value
inside diameter	130mm	small diameter of roller	22.1mm
outside diameter	230mm	large diameter of roller	23.9mm
roller half cone angle	1°	outer ring contact angle	10°
single row rollers number	21	inner ring contact angle	8°
roller effective length	45.8mm	flange contact angle	81°
bearing width	173.2mm	initial radial clearance	0.065mm

CRH3 high-speed train speed is 260km/h, radial force is 56KN, axial force is 4.97KN, the total friction force of the axle box bearing is 350.98 N·m, the friction torque of the roller and outer raceway is 197.64 N·m, the friction torque of the roller and inner raceway is 123.89 N·m, the friction torque of the roller end face and the inner ring flange is 29.45 N·m. The contact load distribution of the roller and the inner ring, the contact load distribution between the roller and the outer ring, and the contact load distribution of the roller end face and the bearing inner ring are shown in Fig3.



(a) first row



(b)second row

Fig. 3. Load distribution of axle box bearing

We can see from the fig3: all roller contact load of high-speed train axle box bearing are symmetrical distribution. The upper half of the axle box bearing is the bearing area, the lower half is the non-carrying area, the roller with the largest load is 0° in the bearing area. The roller load distribution of left and right row are similar, the first row of roller load accounted for 82% of the total load due to the role of axial force. In the same column, the roller and the inner ring, outer ring contact load distribution is similar, in the first row of rollers, the maximum contact load of roller and the outer raceway is 6590.5N, the maximum contact load of roller and the inner raceway is 6206.3N, the difference between the two is 5.83%.

3.1 Load affect on axlebox bearing friction moment

We investigated the effects of axial load and radial load on the total friction torque, respectively. On the one hand, CRH3 high-speed train speed is 260km/h, the axial load is 5KN, with the radial load from 46KN increased to 86KN, we study the total friction torque of axle box bearings; On the other hand, the radial load is 56KN, with the axial load from 0KN increased to 20KN, we study the total friction torque of axle box bearings; The curve of total friction moment of bearing is obtained as shown in fig 4.

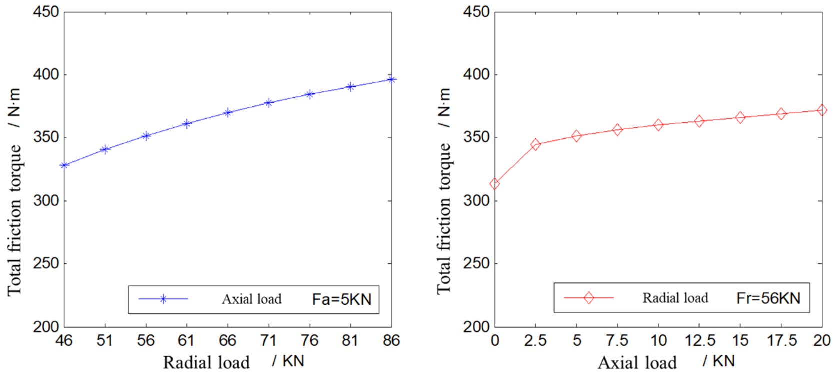


Fig. 4. Influence of load to total frictional torque of bearings

It can be seen from Fig. 4 that when the radial load increases from 46KN to 86KN, the total friction torque of the bearing increases from 327.83N · m to 395.82N·m, the increase rate is about 20.7%. Since the axial load changes the lubrication state between the contacts pairs, the total friction torque changes when the axial load is 2.5KN. As the load continues to increase, the friction torque changes little with the axial load.

3.2 Rotation speed affect on axlebox bearing friction moment

We set the bearing radial load is 56KN, the axial load is 5KN, and then we study the total bearing friction torque changes when the train speed increased from 200km/h to 360km/h, the results shown in Fig 5.

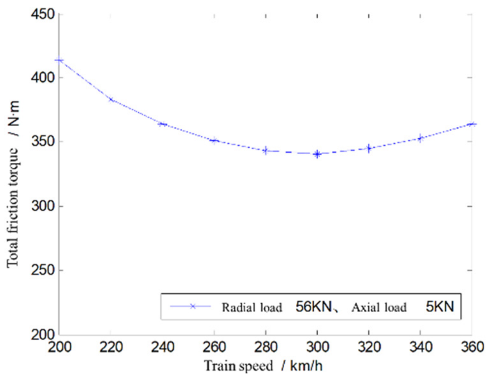


Fig. 5. Influence of speed to total frictional torque of bearings

It can be seen from Fig. 5 that when the train speed increases from 200km to 360km, the total frictional torque decreases firstly and then increases under the given load conditions, and the minimum value of the friction torque is about 340.81N·M when the train speed is about 300km/h. Mainly because when the bearing rotation speed is low, the bearing roller and raceway lubrication state is part elastic flow lubrication, with the bearing rotation speed increases, the friction coefficient decreases, and then the friction torque decreases, when the bearing rotation speed continues to increase , bearing roller and raceway lubrication state change to complete elastic lubrication, the friction torque increases with the bearing speed increases.

3.3 Structural parameters affect on axlebox bearing friction moment

We study the influence of the contact angle of the inner ring of the bearing, the contact angle of the outer ring and the taper of the inner ring on the friction torque respectively. We set the train speed is 260km/h, bearing radial load is 56KN, axial load is 5KN, the bearing inner ring contact angle is 8°, and then we study the axle box bearing friction torque changes when the bearing outer ring contact angle increase from 9° to 14°; the bearing outer ring contact angle is 10°, and then we study the axle box bearing friction torque changes when the bearing inner ring contact angle increase from 5° to 9°; Bearing outer ring contact angle and the inner ring contact angle are fixed, and then we study the axle box bearing friction torque changes when the bearing inner ring edge taper angle increase from 78° to 82°. The results is shown in Fig 6.

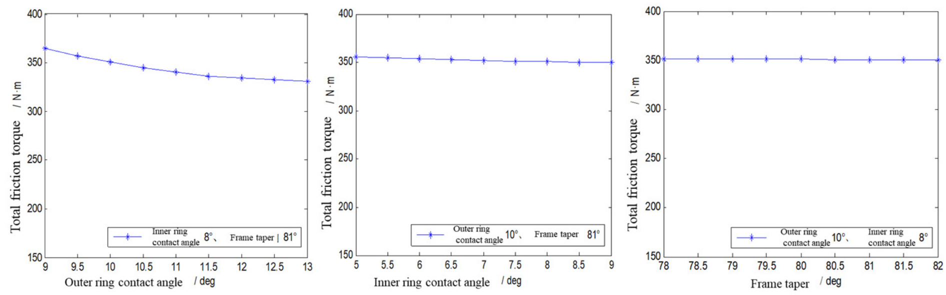


Fig. 6. Influence of outer contact angle to total frictional torque of bearings

It can be seen from the figure6: the outer ring contact angle and the inner ring contact angle increase can reduce the axle box bearing friction torque, when the contact angle of the outer ring is increased from 9° to 13°, the friction torque is reduced by 9.17%, when the inner contact angle increases from 5° to 9°, the friction torque is reduced by 1.66%. The contact angle of the outer ring has a great influence on the total friction torque of the bearing; the contact angle of the inner ring and the taper angle of the inner ring have a little effect on the total friction torque.

3.4 lubrication parameters affect on axlebox bearing friction moment

The other parameters are fixed and the dynamic viscosity of the grease increases from 0.3 Pa·s to 0.9 Pa·s,then we get the friction curve of the axle box bearing the curve shown in Fig 7.

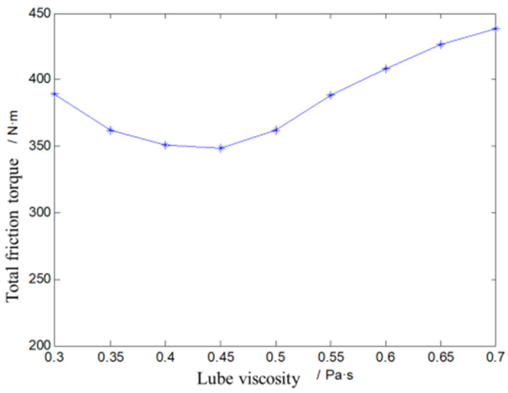


Fig.7. Influence of lube viscosity to total frictional torque of bearings

It can be seen from Fig. 7 that the viscosity of the lubricant has a dramatic effect on the friction torque of the axlebox bearing. When the viscosity of the lubricant changes in the range from 0.3 Pa·s to 0.9Pa·s, the friction torque of the bearing decrease firstly,and then increase. When the viscosity of the lubricant is 0.45 Pa·s, the friction torque is the smallest, which is 348.63N·m. Because the change in the viscosity of the lubricant results in a change in the comprehensive friction coefficient, which makes the friction torque appear as shown in the figure7.

4 Conclusions

A friction performance analysis model of high - speed train axle box bearing is established based on the quasi-static theory and the elastic flow lubrication theory in this paper. A numerical method is proposed to solve the internal load distribution, contact stress distribution, friction force and friction torque distribution of axle box bearing. We analyze the factors that affect the friction torque of the axlebox bearing based on the theory of tribology, and provide a theoretical basis for the structural design and selection of the axle box bearing. We can see from the analysis: the radial load has a greater impact on the bearing friction torque, axial load has a smaller impact on the bearing friction torque. In the normal operation of high-speed trains, as the speed increases, the bearing friction torque decreases first and then increases, and the friction torque is least at 300km/h. The contact angle of the outer ring has a great influence on the friction torque of the bearing, the contact angle of the inner ring and the angle of the taper have little effect on the friction torque; a reasonable lubricating oil can significantly reduce the bearing friction torque at the same time.

Acknowledgement

This project is supported by the Nation Natural Science Foundation of China (Grant No.51375001) the National Key Technologies R&D Program of Liaoning

Provience (Grant No.2015106016) and the Program for Liaoning Excellent Talents in Unversity (Grant No.LJQ2014005 and No.LJQ2014075).

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