Effects of Raceway Convexity on Friction Moment of Tapered Roller Bearings

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Abstract: The Effects of raceway convexities on friction moment of tapered roller bearings were studied in this paper. A new theoretical model of friction torque of tapered roller bearings was established considering the roughness and convexities of inner and outer ring raceways. Friction between roller and raceways were analyzed under the condition of elastohydrodynamic lubrication. Based on the theoretical model and experiments, the changes of friction torque with respect to raceway convexities of tapered roller bearings were obtained. The results show that the friction moment decreases with the increase of raceway convexities, while the increase of raceway convexity leads to the increase of contact stress. Therefore, the convexity values should be balanced between friction torque and contact stress. The optimum values of the bearing being measured raceway convexities should be 5 ~ 6μm under the test conditions.

1. Introduction

Friction torque is one of the most important performances of bearings, because it directly affects the running resistance and energy loss of bearings. Excessive friction torque will not only make the temperature of bearings rise up quickly, but also aggravate the wear of bearing surface and decrease the service life of bearings[1, 2].

The generation of roller bearings’ friction torque involves many influence factors, such as contact load, surface roughness, geometries of roller and raceway, lubrication conditions, etc. The interaction or interference between the factors make the analyses of friction torque complex [3, 4]. Matsuyama[5] showed that the friction torque of tapered roller bearings is mainly composed of the rolling friction between roller and raceway and the sliding friction between roller and inner ring rim by experiment. Justino[6] indicated that the friction torque of bearings increases with viscosity and level of lubricating oil. Todd[7] found that the curvature radius coefficient of rings can also influence the friction torque of bearings. Cousseau[8] indicated that the friction torque generated by spin sliding is about 70% of the total friction torque. Jing[9] proposed a theoretical model of friction torque based on Hertz contact theory, considering the roundness of inner ring and outer ring, and obtaining that the greater the roundness errors are, the greater the friction torque is. Recently, an empirical formula of friction torque was proposed by SKF, which is effective in low speed and heavy load conditions[10].
Because the distribution and magnitude of contact stress between roller and raceways are directly affected by raceway convexities, it is of great practical significance to study the influence of raceway convexities on friction torque of bearings.

To reduce the friction torque of tapered roller bearings, the effects of raceway convexities on friction moment were studied here. A new calculation model of friction moment of tapered roller bearings was established considering the roughness and convexities of inner and outer ring raceways. The friction between roller and raceways were analyzed under the condition of elastohydrodynamic lubrication. And the influences of raceway convexities on friction moment were verified by experiments, obtaining the optimum values of raceway convexities of the test bearings.

2. Theoretical model of friction moment of tapered roller bearing

The total friction in roller bearings is composed of rolling friction and sliding friction. According to the mechanism of friction, the friction torque of tapered roller bearings is mainly caused by

i) rolling friction resistance between roller and raceways;
ii) sliding friction resistance between inner ring rim and roller end face;
iii) sliding friction resistance between roller and cage;
iv) viscous resistance of lubricating oil.

When a tapered roller bearing is subjected to an axial load, the contact relationships between a roller and the inner and outer rings are as shown in Figure 1. When the inner ring rotates and the outer ring is fixed, the friction between inner ring and roller drives the roller to rotate and revolute, and a rolling resistance moment occurs due to the eccentricity of pressure distribution in the contact area [5].

According to the balance of force and torque [5], the friction moment of a tapered roller bearing can be formulated by

\[
\begin{align*}
M_i &= r_i F_i + F_r (r_i + \frac{l}{2} \sin \alpha_i + t \cos \alpha_o) + m_i \\
M &= z_1 \cdot M_i
\end{align*}
\]

when the sliding friction between cage and rollers and the resistance of lubricating oil are ignored. In Eq. (1), \(M_i\) represents the friction torque generated by a roller on the inner ring with respect to the center of inner ring, \(F_i\) the friction force acting on the inner ring raceway by the roller, \(F_r\) the friction force acting on the inner ring rim by the large end of roller, \(r_i\) the average diameter of inner ring raceway; \(m_i\) the rolling resistance moment of the roller and inner ring raceway; \(z_1\) the number of rollers; \(M\) the total friction moment of the tapered roller bearing.

2.1. Friction between roller and raceway of bearing inner ring

When the cross profile of inner ring raceway is arc-shaped, the force of friction between inner ring raceway and a roller can be calculated by integrating the surface shear stress over the contact area [11],
i.e.,

$$F_i = l_i \int_{-b_i}^{b_i} \tau_i dt$$  \hspace{1cm} (2)$$

where $\tau_i$ represents the surface shear stress, $b_i$ the half of contact width, $l_i$ the actual contact length. The actual contact length decreases with the increase of raceway convexity.

Based on GW contact model of real surface $^{[11]}$, the shear stress in contact area can be obtained by

$$\tau_i = c_v A_{C_i} \mu_a \sigma_i + (1 - \frac{A_{C_i}}{A_{b_i}}) \tau_{fi}$$  \hspace{1cm} (3)$$

where $A_{C_i}$ denotes the area of micro-contact, $A_{b_i}$ the normal contact area, $c_v$ the direction of sliding velocity, $\sigma_i$ the normal contact stress, $\mu_a$ the friction coefficient, and $\tau_{fi}$ the shear stress of fluid.

The normal contact stress at any point in contact area is given by

$$\sigma_i = \frac{2Q}{\pi b_i} \left[ 1 - \frac{y}{b_i} \right]$$  \hspace{1cm} (4)$$

where $Q_i$ represents the normal contact load, and $y$ the location of the point in width. The fluid shear stress is formulated by

$$\tau_{fi} = \left( \tau_{Ni} + \tau_{max} \right)^{-1}$$  \hspace{1cm} (5)$$

where $\tau_{Ni}$ is the part of Newtonian fluid shear stress, $\tau_{max}$ is the ultimate shear stress which is about 0.1 times of the contact pressure under oil lubricating condition.

Substituting Eq. (3) into Eq. (2) yields

$$F_i = 2l_i \int_{-b_i}^{b_i} \frac{Q}{\pi b_i} \left[ 1 - \frac{y}{b_i} \right] \frac{A_{C_i}}{A_{b_i}} \mu_a + (1 - \frac{A_{C_i}}{A_{b_i}}) \tau_{fi} dy$$  \hspace{1cm} (6)$$

2.2. Friction between the rim of inner ring and the base surface of roller ball

The friction between the inner ring rim and the large end face of roller is caused by sliding $^{[12]}$, and therefore the friction force can be calculated by

$$F_r = \mu_b Q_f$$  \hspace{1cm} (7)$$

where $\mu_b$ represents the sliding friction coefficient, $Q_f$ the contact load between the rim and roller. The friction coefficient $\mu_b$ is mainly dependent on an oil film parameter $A_{fi}$ that is,

$$\mu_b = c_1 \exp(-c_2 A_{fi}^{-c_4}) + c_4$$  \hspace{1cm} (8)$$

in which $A_{fi}$ is equal to the square mean root of the minimum film thickness and roughness, the values of $c_1, c_2, c_3$ and $c_4$ are constants obtained by actual measurements $^{[5]}$.

2.3. Viscous rolling resistance moment between roller and inner raceway

Under the condition of elastohydrodynamic lubrication, the actual pressure center is offset to the lubricating oil inlet comparing with the geometrical center in contact area. Therefore, the roller is subjected to a rolling resistance in the opposite direction of its movement, generating a viscous rolling resistance moment $^{[13]}$ which can by formulated by

$$m_i = \frac{2Q_l h_i}{3\pi}$$  \hspace{1cm} (9)$$

2.4. The maximum contact stress

When the cross profile of inner and outer raceways are modified into arc shapes, the contact stresses between raceways and rollers are also changed $^{[14]}$. In fact, the values of raceway convexities should be guaranteed within a certain range to prevent stress concentration. Taking the inner ring as an example, the maximum contact stress on raceway is given by

$$Q_{max} = \frac{4.08Q_{or}}{\epsilon_i \cos \alpha_i}$$  \hspace{1cm} (10)$$
where $Q_{or}$ is the rated static load of bearing, $\alpha_i$ is the contact angle. The maximum contact stress on the inner raceway can be obtained by

$$\sigma_{\text{max}} = \frac{2Q_{\text{max}}}{\pi d_i l_i}$$

(11)

The actual contact length decreases with the increase of raceway convexity, and therefore the maximum contact stress increases with the raceway convexity.

3. Experiments

3.1. Raceway convexities of inner ring and outer ring

To verify the influences of raceway convexities on friction torque of tapered roller bearings, groups of test were carried out here. The test specimens are a kind of tapered roller bearing used in gearbox of heavy vehicles. The main parameters of bearing structural are as listed in table 1. The bearing inner and outer rings with different raceway convexities and roughness were obtained by adjusting the process of grinding.

Table. 1 Structural parameters of tapered roller bearings.

| Parameters                  | Numerical value | Parameter             | Numerical value |
|-----------------------------|-----------------|-----------------------|-----------------|
| Inside diameter/(mm)        | 70              | Contact angle         | 28°48′39″       |
| Outside diameter/(mm)       | 150             | Contact angle of inner ring | 20°8′39″      |
| Bearing width/(mm)          | 38              | Number of roller      | 16              |
| Roller contact length/(mm)  | 24              | Material              | GCr15           |

The values of convexity and roughness of the test bearings are as listed in table 2. The inner and outer raceway convexities of the first three groups are 2 μm, 4 μm and 6 μm, respectively. The roughness of the three groups are the same, that is, Ra of the outer raceway is 0.20 μm, Ra of the inner raceway is 0.25 μm, and Ra of inner ring rim is 0.30 μm. The convexities of the second three groups are the same as the first three groups, but the values of roughness are changed. Similarly, the last three groups are also changed in roughness.

Table. 2 Convexities and roughness of tapered roller bearings used in experiments.

| Test groups | Convexity of inner and outer raceway (μm) | Roughness of outer raceway (Ra) | Roughness of inner raceway (Ra) | Roughness of inner ring rim (Ra) |
|-------------|------------------------------------------|-------------------------------|-------------------------------|--------------------------------|
| 1           | 2                                        | 0.20                          | 0.25                          | 0.30                           |
| 2           | 4                                        |                               |                               |                                |
| 3           | 6                                        |                               |                               |                                |
| 4           | 2                                        | 0.15                          | 0.20                          | 0.25                           |
| 5           | 4                                        |                               |                               |                                |
| 6           | 6                                        |                               |                               |                                |
| 7           | 2                                        | 0.10                          | 0.15                          | 0.20                           |
| 8           | 4                                        |                               |                               |                                |
| 9           | 6                                        |                               |                               |                                |

3.2. Measurement of bearing friction torque

The method of measuring friction torque of taper roller bearings is as shown in figure 2, and the structure of device is as shown in figure 3.

The test bearing was cleaned with kerosene firstly. Oil lubrication was applied and the height of oil was about 1/3 of the bearing height. Then the test bearing run for 3 minutes at a uniform speed of 100 r/min before test. In the beginning of measurement, the speed of spindle was 200 r/min and the axial load was 500 N. Then the load increased to 10000 N slowly within 5 minutes. The values of bearing friction torque was obtained by the pressure sensor. The test conditions are as listed in table 3. Each bearing was measured three times to obtain the average value of friction torque.
4. Results

4.1. Theoretical calculation

The effects of raceway convexities on bearing friction torque under different conditions of axial load was calculated and results are as shown in figure 4, in which the roughness Ra of outer raceway, inner raceway and rim are set to be 0.15 μm, 0.20 μm and 0.25 μm respectively. The range of raceway convexity is 1~10μm. It is shown that both of raceway convexity and axial load have great influences on bearing. With the increase of raceway convexity, the friction torque of test bearings decrease, especially when the values of convexity is less than 4μm. In the range of 6~10μm, the influences of convexity on friction torque are not obvious. Besides, with the increase of axial load, the friction torque increases significantly and the effects of raceway convexities on bearing friction torque become more obvious.

The influences of raceway convexity on the maximum contact stress are as displayed in in figure 5, in which the rated static load of tapered roller bearing are 210 KN as the same as test load. The range of raceway convexity is also 1~10μm. The maximum contact stress on raceway increases with raceway convexity notably, because the contact length decreases with the increase of convexity. The contact stress of inner raceway is greater than that of outer raceway, so the fatigue of inner raceway is more easily occurred than the outer raceway, which is consistent with the actual failure modes of bearings. In practice, the maximum contact stress of GCr15 bearings should not exceed 4000 MPa, and therefore the values of raceway convexities of the test bearings should be less than 6μm.

Table 3 Test conditions of bearing friction torque.

| Parameter                          | Numerical value | Parameter                          | Numerical value |
|------------------------------------|-----------------|------------------------------------|-----------------|
| Bearing rotating speed/(r/min)     | 200             | Height of lubricating oil/(mm)     | 12              |
| Axial load/(N)                     | 500~10000       | Measuring time/(min)               | 5               |
| Lubricating oil viscosity/(Paꞏs)   | 15              | Test temperature/(℃)               | 24              |
4.2. Tests of friction torque

The friction torque of tapered roller bearings with different convexities and roughness were measured by the tester in Figure 3. The changes of friction torque with respect to axial load, convexities, and roughness are sketched in Figure 6. Under each condition of roughness, the bearing friction torque is the highest when the value of convexity equals to 2 μm and lowest when the value of convexity equals to 6 μm, verifying the effects of raceway convexities on friction torque. In addition, the bearing friction torque decrease with the roughness of raceways and rim, as shown in Figure 6 (d).

It is noted that in the load range of 2500 ~ 3000 N, the bearing friction torque decreases with the increase of load under all test conditions, because an effective lubricating oil film is formed in the load condition. When the axial load is greater than 3000 N, the bearing friction torque continues to increase with load, although the effective lubricating oil film is formed.

Figure 6 Measured friction torque of tapered roller bearings under different conditions of convexities
Figure 7 shows the comparison between theoretical results and measured values of friction torque under the same condition, i.e., the raceway convexity is 6 μm, the roughness Ra of outer raceway, inner raceway and rim is 0.15 μm, 0.20 μm and 0.25 μm respectively. The measured friction torque conforms to the theoretical model basically under the test conditions. When the load is greater than 6000 N, the deviation between them become larger, because the area of micro-contact increases with load that is ignored in the theoretical model.

5. Conclusion

i) A new theoretical model of friction torque of tapered roller bearings was established considering roughness and convexities of inner and outer ring raceways.

ii) The friction torque of tapered roller bearings with different convexities and roughness were measured in experiments, verifying the effects of raceway convexities on friction torque.

iii) With the increase of raceway convexity, the friction torque of test bearings decrease, especially when the values of convexity is less than 4μm. But in the convexity range of 6 ~ 10μm, the influences of convexity on friction torque are not so obvious.

iv) The increase of raceway convexity is restricted to the maximum contact stress on raceway. For instance, the raceway convexity of the test bearings should not be greater than 6μm, and the optimum value of convexity is between 5 ~ 6 μm.

v) With the increase of axial load, the friction torque increases significantly and the effects of raceway convexities on bearing friction torque become more obvious. Besides, the bearing friction torque decreases with the roughness of raceways and rim.

Acknowledgments

This research was supported by National Natural Science Foundation of China (Nos. 51605190), Shandong Bearing Intelligent Manufacturing Innovation and Entrepreneurship Community Project (2021-2023), Shandong Innovation Capability Improvement Project of Scientific and Technological Small and Medium-sized Enterprises (2021-2023), and Key R & D projects in Liaocheng City (2021-2023).

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