Influence of Torque Load on Internal Load Distribution of Tapered Roller Bearings

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Abstract. The effect of simultaneous application of external torque load on the internal load distribution of tapered rolling bearings under axial and radial loads is studied. The radial deformation and axial deformation of the bearing under the torque load are studied, and the mathematical model of force balance of tapered roller bearing is established. Then the Newton-Raphson method is used to solve the established mathematical model, and the load distribution of the bearing under the two-direction and three-direction loads is obtained respectively. The influence of the torque load on the internal load distribution of the bearing is obtained by the global and local comparison analysis.

1. Introduction

Tapered roller bearings are widely used in many machinery industries because they can withstand radial loads, axial loads and large bearing capacity [1]. At the same time, the tapered roller bearings are separate bearings. It is assembled by the outer ring and the inner ring assembly assembled with the roller and cage. Due to the different axes of the inner and outer rings and other manufacturing and assembly errors, it is inevitable that there will be a torque load on the bearing [2]. The existence of the torque load not only causes the inner and outer rings of the bearing to deflect in the radial plane, but also affects the relative displacement of the inner and outer rings of the bearing, as well as the safe use and fatigue life of the bearing.

In the actual use of the bearing, the load determines the friction and wear performance and fatigue life of the rolling bearing. Therefore, it is of great significance to study the load state of rolling bearings subjected to additional torque load. In this paper, the bearing of a bearing company numbered 31308 is taken as an example to analyze the force of the tapered roller bearing under the action of two-direction and three-direction loads, and the formula for the influence of the torque load on the radial deformation and axial deformation of the bearing is derived, the force balance equation is established. The internal load distribution of the bearing under different load conditions is obtained by programming, and the influence of different torque loads on the load distribution is studied.

2. The internal load calculation of tapered roller bearing

2.1. The internal load calculation under radial and axial loads

Considering the tapered roller bearing with zero clearance, if the radial load and axial load are simultaneously applied to the center plane of the roller of the bearing, the inner and outer rings of the bearing will produce the displacement of $\delta_r$ and $\delta_a$ due to contact deformation in the radial and axial directions respectively, and keep parallel in the radial plane, as shown in Figure 1 [2-3].

If starting with the roller loaded largest, the contact deformation at any angular position $\phi$ is:
\[ \delta_{\phi} = \delta_a \sin \alpha_o + \delta_r \cos \alpha_o \cos \varphi \]  
(1)

Where, \( \alpha_o \) is the contact angle between the roller and the outer ring. The contact deformation reaches the maximum at \( \varphi = 0^\circ \):

\[ \delta_{\text{max}} = \delta_a \sin \alpha_o + \delta_r \cos \alpha_o \]  
(2)

Figure 1. Bearing displacement under radial and axial loads

According to the empirical formula of \( \delta \) in the case of line contact proposed by Palmgren \[4\]:

\[ \delta = 3.84 \times 10^5 \frac{Q^{0.9}}{L^{0.8}} \]  
(3)

The load-displacement relationships be derived:

\[ Q = K \delta^9 \]  
(4)

For the 31308 bearing, the roller and raceways are made of steel, \( K = 8.06 \times 10^4 L^{0.9} \).

The normal load between the roller and raceways at different angular positions \( \varphi \) is:

\[ Q_{\varphi} = K \delta_{\varphi}^9 = K (\delta_a \sin \alpha_o + \delta_r \cos \alpha_o \cos \varphi)^9 \]  
(5)

According to the static balance, the resultant force in all directions of the roller must be equal to the load acting in this direction \[5\], which is:

\[
\begin{align*}
F_a &= \sum_{\varphi = \varphi_1} Q_{\varphi} \sin \alpha_o \\
F_r &= \sum_{\varphi = \varphi_1} Q_{\varphi} \cos \alpha_o \cos \varphi
\end{align*}
\]  
(6)

Where, \( \varphi_1 = \cos^{-1} \left( \frac{\delta_a \tan \alpha_o}{\delta_r} \right) \) is the load angle of the loaded roller.

By using the Newton-Raphson iterative method to solve the nonlinear equation (6), \( \delta_r, \delta_a \), and the load \( Q \) of different rollers can be calculated for subsequent analysis of the contact stress.

2.2. The internal load calculation under radial, axial and torque loads

For bearings under radial, axial and torque loads, in addition to considering radial and axial loads, the effect of torque loads on roller-to-raceway contact deformation is also considered. For the contact deformation at any position angle \( \varphi \), in addition to the above-mentioned radial deformation and axial deformation, it is also necessary to consider the influence of the deflection angle \( \theta \) on the radial deformation and axial deformation due to the torque load \[3, 6-7\].

For tapered roller bearings with three-direction loads at the same time, the radial deformation and axial deformation will be changed due to the presence of the deflection angle \( \theta \), as shown in Figures 2 and 3.
As shown in Figure 2, the axial displacement at the position $\varphi$ due to the deflection angle $\theta$ is \[ \delta_a = 0.5d_m \cos \varphi \] (7)

Then the total axial deformation under the three-direction loads is:

\[ \delta_{a0} = \delta_a + \delta_{a\theta} \] (8)

For the radial displacement $\delta_{r0}$ at the position $\varphi$ caused by the deflection angle $\theta$, since the influence of $\theta$ on $\delta_{r0}$ is small, the calculation is performed approximately in the dimension of the pitch circle when considering the contact deformation between the roller and the outer race, as shown in Figure 3.

\[ \delta_{r0} = 0.5d_m \theta \tan \alpha + 0.5d_m (1 - \cos \theta) \] (9)

Where, $d_m$ is the diameter of the pitch circle. At the same time, since the deflection angle $\theta$ is small, the above formula can be reduced to:

\[ \delta_{r0} = 0.5d_m \theta \tan \alpha \] (10)

Then the total radial deformation at any angular position under the three-direction loads is:

\[ \delta_r = (\delta_a + \delta_{a\theta}) \cos \varphi \] (11)

Totally, the normal deformation of the roller-raceway at any position angle $\varphi$ is:

\[ \delta_\varphi = \delta_{a\varphi} \sin \alpha + \delta_{r\varphi} \cos \alpha \] (12)

Where, $\alpha = \alpha_o + \theta \cos \varphi$ \[^{[6-7]}\] the effect of angular deviation on the contact angle between the roller and the outer race is considered.

According to the load-displacement relationship, the normal load between the roller and the raceways at different position angles $\varphi$ is:

\[ Q_\varphi = K \delta_\varphi^{10} \] (13)

According to the static balance, the resultant force in all directions of the roller must be equal to the load acting in this direction \[^{[7]}\], which is:
Similarly, using the Newton-Raphson iterative method for the nonlinear equation (14), \( \delta_r, \delta_a, \theta, \) and the load \( Q \) of the different rollers can be calculated for subsequent analysis of the contact stress.

3. Comparison and analysis

In this paper, the 31308 bearing is taken as an example for calculation and analysis. The specific parameters of the bearing are shown in Table 1.

| Inter/outer diameter (mm) | Bearing width (mm) | Inner ring width (mm) | Outer ring width (mm) | Pitch diameter (mm) | Contact angle (°) | Number of roller |
|--------------------------|-------------------|----------------------|----------------------|--------------------|-----------------|-----------------|
| 40/90                    | 25.25             | 23                   | 17                   | 65                 | 28.8103         | 15              |

In this paper, the radial load is consistent, and the axial load is gradually changed. When the axial load increases from 6000N to 22000N, the number of loaded rollers gradually increases, and all are loaded finally; the contact load of each roller is gradually increased, and the larger the position angle, the smaller the contact load. Taking \( F_a = 20000N \) as an example, the number of loaded rollers is 13, and the load of the roller loaded minimum is 365N. Finally, the load distribution of the 31308 bearing under radial and axial loads is obtained by calculation, as shown in Figure 4.

3.1. Influence of torque load on radial and axial displacement

For bearings that simultaneously apply a torque load, the inner and outer rings have not only radial displacement \( \delta_r \), axial displacement \( \delta_a \), but also deflection angle \( \theta \) and influence on radial and axial displacement. As shown in Figure 5, \( \delta_\phi = 0.0004145 \) mm and \( \delta_\phi = 0.010479 \) mm when \( M = 0 \).

As the negative torque load increases, \( \delta_r \) gradually decreases or even becomes negative. This is because the presence of the negative torque load causes the inner ring to deflect counterclockwise relative to the outer ring, and the top roller gradually contacts the outer ring, the number of loaded rollers increases gradually. The contact deformation between the roller and the outer ring at the bottom gradually decreases until separating from the outer ring; the increasing of \( \delta_a \) is due to the negative torque load acting on the outer ring.
load, which causes the inner ring to press the outer ring more and more seriously. As the positive torque load is increased, δ gradually increases. This is due to the positive torque causing the inner ring to deflect clockwise relative to the outer ring. The top roller has a tendency to break away from the outer ring, while the bottom roller is pressed against the outer ring seriously. Therefore, the radial deformation increases; at the same time, the inner ring has a tendency to detach from the outer ring as a whole, so the axial deformation is reduced.

3.2. Influence of torque load on internal load distribution

As shown in Figure 6, when $M=0N$, the bearing is not subjected to the torque load, the number of loaded rollers is 13, and the minimum load of the roller is $365N$.

When subjected to a negative torque load, the inner ring deflects counterclockwise relative to the outer ring, and as the negative torque increases, the upper rollers gradually press the outer ring, the number of loaded rollers increases, and the load distribution is more stable. The upper roller contact load is gradually increased; the lower roller is gradually separated from the outer ring, the contact deformation is reduced, and the roller-raceway contact load is gradually reduced. When subjected to a positive torque load, the inner ring is deflected clockwise relative to the outer ring, the upper roller contact deformation is reduced, and the upper roller gradually separate from the outer ring, the roller-raceway contact load becomes small, the number of loaded rollers is reduced; The lower rollers press the outer ring seriously, the contact deformation increases, and the roller-raceway contact load increases.

3.3. Influence of torque load on the load of a single roller

The largest loaded roller is taken as the starting point under the two-direction load. When the torque load is not applied, the load of the roller at $\phi=0$ is $Q_{0}=6047$, the other rollers $Q_{1}=5752N$, $Q_{2}=4927N$, $Q_{3}=3738N$, $Q_{4}=2418N$, $Q_{5}=1222N$, $Q_{6}=365N$, $Q_{7}=0N$. When the negative torque load is applied initially, the load of each roller is reduced. This is because the roller at $\phi=0$ starts to be loaded, and the load of each roller decreases when the number of loaded rollers increases, as shown in Figure 7.

As the negative torque load continues to increase, the roller load at $\phi=0$ to $\phi=3$ continues to decrease, the roller load at $\phi=0$ becomes 0, and the roller load at $\phi=4$ to $\phi=7$ gradually increases. This is because the inner ring deflects counterclockwise relative to the outer ring. The contact deformation of upper roller-raceway is increased, and lower roller-raceway contact deformation is reduced, even separate from each other.

With the increase of the positive torque load, the inner ring deflects clockwise relative to the outer ring, the upper roller-raceway contact deformation is reduced, the contact load is reduced. While the lower roller greatly presses the raceway, and the contact deformation is increased, the contact load increases.
4. Conclusion

(1) According to the previous research results, the paper derives the force balance equation of tapered roller bearings under two-direction load and three-direction load, and the effect of deflection angle $\theta$ on radial deformation and axial deformation is studied. Then, the Newton-Raphson iterative method is used to solve the contact deformation and load distribution under different load conditions.

(2) Comparing and analysing the overall contact deformation, load distribution and the load of the single roller when applying the torque load, we can conclude that the appropriate moment load can increase the number of loaded rollers and improve the load distribution of the bearing.

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