Thin-wall bearing contact characteristics analysis and deformation research

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Abstract. Considering the elastic deformation of the bearing ring structure, a nonlinear contact model of the thin-walled deep groove ball bearing (61856) was established. The contact stress and deformation of the thin-walled deep groove ball bearing were simulated and analyzed under the two conditions of rigid and flexible constraints. The distribution law of contact stress and deformation of bearing rolling elements and different parts of inner and outer rings was studied. The comparison between the simulation results and the test data shows that the error between the simulation results and the test data is relatively large under rigid constraints, which is not suitable for the analysis of the finite element mechanical characteristics of thin-walled bearings. However, it has obvious consistency under flexible constraints, and the simulation results are basically consistent with the experimental data. The experimental results show that it is feasible to use ANSYS to analyze the contact problem of thin-walled bearings under flexible constraints, which is in line with the actual situation of thin-walled bearings under stress and deformation. The results of finite element simulation calculations provide a reference for the lubrication, comprehensive mechanical properties and failure analysis of thin-walled rolling bearings and lay a foundation for further research on the thin-walled bearings’ elastic contact problems.

1. Introduction
As the first choice of supporting bearings for modern industrial robots to become lighter, thin-walled bearings have excellent properties such as light weight, small size, high accuracy, and high stiffness ratio [1, 2]. However, as a high-precision product, thin-walled bearings have little domestic research. Most of thin-walled bearings have never considered the deflection of thin-walled bearings in actual use. The design theory of thin-walled bearings is not perfect, and the preparation process parameters also lack theoretical and experimental basis [3, 4, 5]. Research on the contact stress and deformation of thin-walled bearings is a key factor restricting the development of industrial robots. Therefore, in order to gain a deeper understanding of the contact characteristics of thin-walled bearings, this article takes thin-walled deep groove ball bearings (61856) as the research object. Based on the combination of theoretical analysis and experiment, a nonlinear finite element contact model of thin-walled bearings is established. The distribution of contact stress and deformation of the inner and outer rings of thin-walled bearings was studied. The influence mechanism of contact force and parameter change on the deformation of the ferrule under rigid and flexible constraints is analyzed.
2. Analysis of contact state of thin-walled bearings

2.1. Hertz contact theory and calculation results

As shown in Figure 1, during the actual operation of the rolling bearing, the rolling element contacts the inner and outer rings under the load, and the contact point is elastically deformed. The contact area increases from point to surface and increases with increasing load. The contact area is similar to an ellipse, and the surface pressure is distributed as a semi-ellipsoid. The contact stress calculation formula of deep groove ball bearings can be derived from Hertz contact theory:

The main curvature and function are:

\[
\sum p = p_{I_1} + p_{I_2} + p_{II_1} + p_{II_2}
\]

The main curvature difference function is:

\[
F(\varphi) = \frac{p_{I_1} p_{II_2} - p_{I_2} p_{II_1}}{2p} \quad (2)
\]

Maximum contact pressure:

\[
P_{\text{max}} = \frac{3F_r}{2\pi a b} \quad (3)
\]

Contact elastic deformation:

\[
\delta = \delta_1 \left( \frac{3F_r}{2E}\right) \frac{n_0^2 p}{r_0^2} \quad (4)
\]

In the formula: \( p \) is the principal curvature of the contact body, which is the inverse of the radius \( r_{I_1}, r_{I_2}, r_{II_1}, r_{II_2} \); \( Q \) is the normal contact load between the rolling body and the raceway; \( \sum p \) is the main curvature and function; \( F_r \) is the radial load of the bearing; \( k \) is the ellipticity of the contact ellipse; \( e \) is the ellipse parameter of the contact ellipse.

\[\text{Figure 1. Distribution of compressive stress in contact ellipse}\]

The Hertz contact theory is used to solve the maximum contact stress, deformation and semi-axis size of the rolling bearing at the contact ellipse. The calculated value of the thin-walled deep groove ball bearing 61856 under radial load of 10KN is shown in Table 1 and Table 2.

\[\text{Table 1. Calculated values of the inner ring Hertz theory}\]

| \( f_i \) | a/mm | b/mm | \( P_{\text{max}}/\text{Mpa} \) | \( \delta/\text{mm} \) |
|---|---|---|---|---|
| 0.515 | 5.239 | 0.513 | 3283.87 | 0.0551 |
| 0.520 | 4.651 | 0.544 | 3491.06 | 0.0593 |
| 0.525 | 4.244 | 0.569 | 3660.70 | 0.0625 |
| 0.530 | 3.940 | 0.589 | 3805.32 | 0.0653 |
2.2. Finite element analysis under rigid constraints

Using SolidWorks modeling software to establish a three-dimensional analysis model of thin-walled deep groove ball bearings 61856, as shown in Figure 2. The materials of the steel ball and the inner and outer rings are GCr15, the elastic modulus is 206GPa, the Poisson's ratio is 0.3, and the main structural dimensions are shown in Table 3.

Table 3. Main parameters of deep groove ball bearings

| Structural parameters                     | Parameter value | Structural parameters                     | Parameter value |
|-------------------------------------------|-----------------|-------------------------------------------|-----------------|
| Bearing width /mm                         | 33              | Rolling body diameter/mm                  | 19.05           |
| Bearing inner diameter/mm                 | 280             | Bearing outer diameter/mm                 | 350             |
| Radius of inner ring groove curvature/mm  | 9.81            | Radius of outer ring groove curvature/mm  | 10              |
| Contact angle /(°)                        | 0               | Number of balls                           | 27              |

Figure 2. Bearing model

Automatic sweeping method is used to divide the grid of contact area. The mesh between the steel ball and the inner and outer raceway contact surface is refined, the size is 1.5mm. As shown in Table 4, under the condition of rigid constraints, the bearing is completely fixed in the seat hole. The outer ring is used to fix the inner ring to apply the speed of rotation. To displacement. Apply axial and tangential constraints to the balls. The inner ring speed is 150 rad / s, and a radial load of 10KN is applied to the bearing inner ring. Bearing dynamic friction coefficient is 0.1, normal contact stiffness factor is 2.5, the contact formula uses a penalty function, and the number of working cores of the processor is 15.

Table 4. Constraints on main bearing parts

| main part     | Translation constraint | Rotation constraint |
|---------------|------------------------|--------------------|
| Inner circle  | Z                      | X, Y               |
| Outer ring    | X, Y, Z                | X, Y               |
| Rolling body  | X, Z                   | Unconstrained      |

Figure 3 shows the distribution cloud diagram of the contact stress and deformation of the bearing ball and the inner and outer raceway contact surfaces under different channel curvature radii. The size and shape of the semi-axis of the contact area are shown in Figure 3. When the radius of curvature of the inner channel is 9.81mm, the maximum contact stress between the rolling element and the inner ring is
2926.7 MPa, and the maximum contact deformation is 0.052 mm. When the radius of curvature of the inner channel is 9.90 mm, the maximum contact stress between the rolling element and the inner ring is 3250.8 MPa, and the maximum contact deformation is 0.054 mm. When the radius of curvature of the inner channel is 10.02 mm, the maximum contact stress between the rolling element and the inner ring is 3574.8 MPa and the maximum contact deformation is 0.058 mm.

![Figure 3. Finite element calculation results](image)

The above finite element calculation results show that the maximum contact stress of the bearing under rigid constraints should be located at the contact point of the radially bottommost rolling element with the inner and outer raceways. The force range of the contact surface between each rolling element and the inner and outer rings is distributed elliptically. As the curvature coefficients of the inner and outer grooves increase, the deformation and contact stress of the ball contact surface also increase. As shown in Table 5, as the curvature coefficient increases, the error between the finite element calculation result and the theoretical calculation value also increases.

### Table 5. Simulation calculation results

| Radius of inner channel curvature /mm | stress/Mpa | Error | Maximum deformation /mm | Error |
|--------------------------------------|------------|-------|-------------------------|-------|
| 9.81                                 | 3283.87    | 5.6%  | 0.0526                  | 4.5%  |
| 9.90                                 | 3491.06    | 8.9%  | 0.0547                  | 7.7%  |
| 10.00                                | 3660.70    | 11.1% | 0.0559                  | 10.2% |

### 3. Finite element analysis and calculation under flexible constraints

Generally, the finite element analysis of rolling bearings is considered to be carried out under a rigid support, that is, the outer ring is fixed and the inner ring rotates. But thin-walled bearings will have a certain degree of tangential elastic deformation in actual operation. At this time, the ring should consider flexible support, that is, there is a fit gap between the bearing seat hole and the bearing outer ring. Through the ANSYS finite element analysis software, the finite element contact model of the thin-walled deep groove ball bearing and the housing bore is established. Analyze the influence of load parameters and fit clearance on the contact force and deformation of bearing rolling elements.
Figure 4. Bearing model under flexible support

Figure 5. Meshing and constraint application

Figure 6. Bearing deformation under 15um clearance

Figure 7. Distribution curve of contact pressure of rolling elements under different clearances

When the radial load is set to 10KN, the gap between the bearing outer ring and the bearing seat hole is 5um, 10um, 15um, 30um. It can be seen from the analysis results in Figure 7: With the gradual increase of the clearance between the bearing and the bearing housing, the position of the maximum
contact load of the rolling element will shift. When the rigidity constraint or the fit clearance is small, the rolling element load distribution still follows the Hertz contact theory, and its peak value is still on the external load line of action. However, when the fit clearance increases to a certain degree, the peak value of the load gradually shifts to both sides and the value gradually increases due to the bending deformation of the outer ring, and the number of loaded balls is gradually increasing. The increase in the fit clearance causes a significant increase in the maximum load on the rolling elements. As the fit clearance increases, the contact pressure difference between the loaded rolling elements also increases.

Figure 8. Distribution curve of tangential deformation of outer ring under different mating clearances

It can be seen from Figure 8: when the fit clearance is 5um, the maximum deformation peak of the bearing outer ring is still on the force action line, but as the fit clearance continues to increase, the maximum deformation peak of the outer ring is not on the action line of the external load. Is offset to both sides. Compared with the previous analysis, it can be seen that in the practical application of thin-walled ball bearings, the traditional rigid support theory is irrational in analyzing the load distribution of rolling elements.

4. Experimental study on the pressure of thin-walled deep groove ball bearings

Through the contact characteristics analysis of the thin-walled deep groove ball bearings using the finite element software in the previous two sections, it was found that a certain amount of deformation occurred in the tangential direction of the bearing under flexible constraints, and the contact pressure of the balls and the inner and outer rings also changed accordingly. In order to verify whether the simulation results are correct, it is necessary to study the radial and tangential deformation of the bearing outer ring and the ball under load. As shown in Figure 9, the bearing is fixed on the hydraulic press working table by a special fixture, and the loading pressure is applied to the upper clamp through the hydraulic pressure punch and is sequentially transmitted to the bearing outer ring, steel ball and inner ring. Gradually increase the pressure load on the tested bearing, the bearing will undergo a certain degree of elliptical deformation and produce a downward displacement. The amount of deformation is measured by the displacement sensor on the press and displayed on the data acquisition instrument. The tangential deformation of the outer ring of the bearing is measured by a vernier caliper.
Figure 9. Comparison of tangential deformation results of outer ring under different radial loads

Take the bearing fixture when the fit clearance is 30um to match with the bearing and measure the tangential deformation of the bearing outer ring when the radial load is 10KN and 20KN respectively. Comparing the measured value with the simulated solution, as shown in Figure 9, the thin-walled bearing did not undergo significant tangential deformation under rigid constraints, while under the flexible constraint, there was significant tangential elastic deformation. The simulation solution basically agrees, and is slightly higher than the simulation solution.

5. Conclusion
Thin-walled bearings produce tangential elastic deformation during work, and the contact analysis must take into account the effect of bending deformation on the contact load after the inner and outer rings of the bearing. The traditional rigid support analysis of bearing contact load is obviously not enough, and the calculated numerical error is large.

The rigidity assumption of the existing ferrule structure was abandoned, and a thin-wall bearing flexible contact model was established. Using ANSYS software to analyze the force of thin-walled bearings under flexible support, it is found that when the fit clearance is small, the simulation values are not much different from the calculation results under rigid support. With the gradual increase of the fit clearance, the maximum stress point of the balls began to shift to both sides, and the number of the stressed balls also gradually increased.

By comparing the measured values with the simulation results, the analysis found that the thin-walled bearings did not undergo significant tangential deformation under rigid constraints. In the case of flexible constraints, there is obvious tangential elastic deformation. The measured values are basically consistent with the simulation results and are slightly higher than the simulation results. It shows that it is feasible to use ANSYS to analyze the contact problem of thin-walled bearings under flexible constraints. But with the application of larger loads, the error between the measured values and the simulation results also gradually increased.

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