Sub-surface stress analysis of slewing bearing ball based on plastic deformation

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Abstract. Slewing support is widely used in large slewing machinery such as rocket launching platform, wind turbine and port crane. The accurate analysis of the internal stress in the raceway of ball roller is an important basis for studying the cause of bearing fatigue failure and improving the reliability of slewing support. In this paper, the finite element model of thrust angular contact rotary support was established in ANSYS. The stress distribution of the raceway of the ball under different working conditions was calculated by using bilinear isotropic plastic materials and linear elastic materials, and the equivalent stress resulted of the contact surface were compared with the Hertz contact model. This paper studied the factors that affect the distribution of the equivalent stress on the contact surface, calculates the ratio of the equivalent shear stress and the normal stress on the contact surface by finite element method, namely the equivalent friction coefficient, and studied the effect of the equivalent friction coefficient on the distribution of the stress inside the rolling ball. Simulation results show that plasticity occurs in the rolling ball.

1. Introduction

Slewing bearings are widely used in large equipment such as hoisting machinery, engineering machinery, military equipment, etc. The mechanical structure of slewing bearings has raceways and rolling bodies similar to ordinary bearings. However, slewing bearings are large in size and heavy in load. The load situation is complicated.

Since the beginning of the 20th century, research has been conducted on the contact problems of rolling bearings. The study of the stress distribution during the contact between the bearing raceway and the ball has been an important issue for scholars at home and abroad. Harris [1] A complete set of bearing analysis methods based on the Hertz stress contact model It has been proved through experiments that this method has a high accuracy for the calculation of the contact stress of the bearing. However, when the support works under heavy load conditions, the contact deformation of the raceway ball will often experience elastic deformation. To plastic deformation. For slewing bearings used in industrial heavy loads, the contact pressure of the ball raceway often exceeds 3000Mpa. Xu Hongyi et al. [2] used ANSYS / LS-dyna to establish a deep groove ball based on the plastic constitutive equation. The bearing finite element model, simulation results show that under heavy load conditions, the bilinear plastic model and the follow-up model can better simulate the contact deformation of rolling bearings. Zheng Hongmei et al. [3] confirmed the plastic material model through experiments with an electronic universal material testing machine The accuracy of finite element analysis under heavy load conditions. Shang Zhenguo et al. [4] used ANSYS connection unit
combine to contact the inner and outer ring raceways with the rolling elements. To effect axial compression spring, in order to explore the plastically deformable material effect on the large heavy rotary bearing internal load distribution and size of the contact stress, ignoring the influence of the contact ball raceway moving the loading process.

In this paper, a slice model of single-row ball thrust angular contact slewing bearing is established, and the equivalent stress distribution of the ball contact surface under different loads is compared and analyzed. The equivalent friction coefficient is applied to the secondary surface of the ball, that is, the stress below the contact surface.

2 Finite element model

2.1 Three-dimensional model of slewing bearing

Taking a single-row two-point contact slewing bearing used in a large slewing machine as an example, a finite element analysis is performed. The slewing bearing has a ball diameter of 40mm, a bearing outer diameter of 4980mm, a bearing inner diameter of 4880mm, an outer ring curvature radius factor of 0.53, and an inner ring. Coefficient of curvature radius 0.53, number of rollers 298.

In order to facilitate the meshing and improve the quality of the mesh, the sliced solid is segmented. The upper and lower raceways use 20-node solid high-order hexahedral elements, and the rolling body adopts 8-node solid. Tetrahedral element, as shown in Figure 1.

![Figure 1 Finite element model](image)

2.2 Material model

The plastic material model can not only reflect the elastic deformation behavior of the material, but also reflect the plastic deformation that occurs after the material reaches the yield limit.

This material model simplifies the stress-strain curve into a polyline, where the yield strength \( \sigma \) is 1280Mpa, Poisson's ratio 0.3, elastic modulus E is 210Gpa, and shear modulus Et is 2Gpa.

2.3 Solution settings

The entire solving environment was performed in ANSYS Workbench. The contact algorithm selected extended Lagrangian multiplier algorithm. The study found that excessive contact stiffness would cause the overall stiffness matrix to be ill-conditioned, and the calculation results would be difficult to converge. After repeated debugging, the normal contact stiffness FKN The setting is 0.8 and the penetration value FTON is set to 0.02. The calculation result is ideal.

3. Analysis of calculation results

3.1 Contact analysis of slice models

Through finite element simulation calculations, the contact stresses of different finite element models of different material properties under different displacement loads are shown in Table 1.
Table 1 Maximum equivalent stress of the ball in plastic and elastic material models

| Displacement load $F_a$/mm | Contact stress between ball and upper raceway / Mpa | Contact stress between ball and upper raceway / Mpa | Ball maximum stress / Mpa |
|-----------------------------|--------------------------------------------------|--------------------------------------------------|--------------------------|
| 0.1                         | 441.33                                           | 475.25                                           | 475.25                   |
| 0.2                         | 899.16                                           | 968.83                                           | 968.83                   |
| 0.5                         | 989.45                                           | 1079.30                                          | 1278.50                  |
| 1                           | 1282.60                                          | 1351.5                                          | 1678.40                  |

The simulation history is 1s, and every 0.05ms, software sampling calculations. When the displacement load is 0.2mm, the maximum plastic strain is 0.00041; when the displacement load is 0.5mm, the maximum plastic strain is 0.1429, and when the loading time is 0.4s, The plastic strain is 0.0002708, and the displacement load is 0.2mm. At a load of 1mm, the maximum plastic strain is 0.02481. When the loading time is 0.2s, the plastic strain is 0.0002535, and the corresponding displacement load is 0.2mm. Displacement load is the critical load for the plastic behavior of the model.

3.2 Stress analysis under the Hertzian contact model

Hertz made two important assumptions in contact analysis [3]:

1. All deformed bodies are in the range of elastic deformation, and do not exceed the limit of material proportion.

2. The load is perpendicular to the surface, and the effect of surface shear stress is ignored.

During the point contact process, Hertz established the following normal stress mathematical model, and the normal stress distribution of the contact ellipse is shown in the Figure 2:

![Figure 2 Ellipse model of Hertzian contact](image)

Where $Q$ is the normal load, $a$ is the long radius of the stress ellipse, and $b$ is the short radius of the stress ellipse, $\sigma$ is the normal stress of the contact ellipse. It is easy to get the maximum pressure center to appear at the geometric center.

3.3 Stress analysis of rolling ball by finite element model

During the loading process, due to the friction between the gap and the contact surface, the contact angle between the ball and the raceway changes during the actual loading process.

As shown in Figure 3, the maximum stress surface of the ball raceway is misaligned, indicating that the raceway and the ball have shifted from each other during the loading process. The contact deformation and clearance of the ball and the upper and lower raceways will affect the ball. Actual contact angle.
3.4 Effect of equivalent friction coefficient on subsurface stress distribution

From Figure 4, it can be seen that when the displacement load is 0.1mm, the maximum equivalent stress is less than the yield strength of the ball raceway material. Under this condition, the raceway and rolling elements are in the stage of elastic deformation, and the maximum stress concentration point is still at the center of the contact surface, as the load increases, the stress ellipse continues to increase, but it is always close to the surface. When the displacement load is increased to 0.2mm, the maximum equivalent stress appears at the position of the subsurface.

In Figure 4, it can be observed that when the equivalent friction coefficient is less than or equal to 0.1925, the von-miss is the largest, etc. The position of the effect force is on the sub-table, and the stress depth is constantly increasing; when the ratio is greater than 0.1925, the von-miss maximum equivalent stress position is on the sub-table.

4.Conclusions

When the support undergoes plastic deformation, the equivalent friction coefficient of the contact surface decreases with increasing displacement load. When the equivalent friction coefficient decreases to 0.1925, the maximum stress of the support occurs on the secondary surface, that is, the contact surface. The following positions.

(1)In the plastic stage, it can be observed from Figure 3 that the maximum equivalent stress appears...
on the subsurface of the ball and raceway, so it can be determined that the fatigue failure of the ball occurs from the inside of the material, and the cracks propagate from the inside to the outside. Causes pitting on the ball surface.

(2) Analysis of the effect of the equivalent friction coefficient on the distribution of subsurface stress is conducive to the reasonable setting of processes such as carburizing, locally strengthening the raceway, and prolonging the service life of the slewing bearing.

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