Analysis of the influence of raceway structure on contact characteristics of turntable bearing

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Abstract. Turntable bearing is widely used in heavy industry. Raceway structure is the most direct factor that affects the contact stress change between raceway and rolling element. Therefore, it is the key to improve the safety of turntable bearing and determine the processing technology reasonably to study the stress distribution of contact area in different raceways. In this paper, the finite element slicing model of rolling element and raceway in turntable bearing is established in ABAQUS. Under the action of axial load and radial load, the distribution characteristics of contact stress of raceway are compared under different raceway structures. The finite element results show that, compared with the two-point contact disc bearing, the rolling element and raceway in four point contact turntable bearing are in four point contact under axial load. The contact stress distribution of raceway is uniform, which is a pair bearing distribution. However, when only considering the radial load, the rolling element and raceway are in two-point contact, and the contact stress distribution characteristics are consistent with those of two-point contact rotary table bearing.

1. Introduction

Turntable bearing is widely used in aerospace, military, petroleum, coal mine and ocean and many other heavy industry fields. It is the core part of heavy rotary equipment. The failure of turntable bearing is mostly the fatigue failure of raceway. Therefore, the research on raceway strength, stress distribution and failure mechanism has always been a hot spot.

In 1947, Lundberg and palmgren found that the maximum alternating shear stress under the subsurface contact between the bearing and raceway was the main reason for the initiation of cracks in the bearing raceway. The maximum alternating shear stress was parallel to the rolling direction of the ball. The relationship between the static load capacity and the service life of the rolling bearing was studied by experiments, and the bearing rated dynamic load theory, i.e. L-P theory[1], was established. So the foundation of bearing life calculation is established. In the industry, the design of rotary bearing is mostly based on the static load limit condition. In 1991, Li Chenggang counted the fatigue life of rolling bearing and analyzed the data. The experiment proved that the fatigue life of large rolling bearing obeys Weibull distribution[2]. In 2003, Su liyue et al. Drew the curve diagram of load and life according to the load spectrum corresponding to the actual working conditions, and improved the selection of turntable bearing[3] under complex working conditions. In 2016, Zheng Hongmei et al. Carried out indentation test on 50Mn by material electronic universal testing machine, and established the relationship equation between contact permanent deformation of rotary table bearing material and ratio of rolling element diameter to contact stress, and compared with the
experimental results in ABAQUS indentation simulation process, it has better consistency, on the other hand, it also shows the reliability of finite element software calculation[4].

2. Hertz contact theory model

The stress distribution of the contact point, the maximum contact stress and the stress distribution of the roller bearing are analyzed.

The calculation method of Hertz contact theory is based on two basic assumptions[6]

(1) All deformable bodies are in the range of elastic deformation, and do not exceed the proportional limit of materials.

(2) The load is perpendicular to the surface, ignoring the influence of surface shear stress.

In the process of point contact, Hertz establishes the following mathematical model of normal stress, and the normal stress distribution of contact ellipse is shown in Fig. 1

\[ \sigma = \frac{3Q}{2\pi ab} \left[ 1 - \left( \frac{x}{a} \right)^2 - \left( \frac{y}{b} \right)^2 \right]^{\frac{1}{2}} \] (1)

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 contact ellipse normal stress. It is easy to get that the maximum pressure center appears in the geometric center for the elliptical contact area of curved surface.

3. Establishment of finite element model

3.1. Mesh generation

In order to facilitate mesh generation and improve mesh quality, sliced solid model of bearing is sliced, as shown in Fig. 2. The raceway is divided properly, the model size is set as 3mm by global seed, and the mesh size is set as 0.4mm by edge seed for local densification; the global seed setting is used for rolling ball, and the unit size is set as 0.8mm.

![Finite Element Model](image1)

(a) Two point contact turntable bearing  
(b) Four point contact turntable bearing

**Figure 2.** finite element model
3.2. Material model

The raceway of turntable bearing is made of alloy steel 5CrMnMo, the elastic modulus is 207000mpa, Poisson's ratio is 0.3, and the allowable contact stress is 3850mpa. GCr15 is usually used as rolling element, the elastic modulus is 20600mpa and Poisson's ratio is 0.29.

Turntable bearings are usually operated under heavy load. In order to consider the plastic deformation between the rolling element and the raceway, the material ideal elastic-plastic model is proposed, and the yield strength is 1280mpa.

3.3. Contact settings

Only the normal and tangential attributes are considered in the contact properties between the rolling body and the raceway. When there is pressure between the rolling element and the raceway, the raceway contacts with the rolling body, otherwise, separation occurs, so the normal behavior chooses "hard" contact; the friction coefficient is set as 0.03, and the contact behavior is constrained by penalty function method in the contact attribute. There may be a large range of rotation and sliding of the ball in the raceway, so there is limited sliding between the ball and the raceway.

4. The influence of raceway structure on contact stress distribution of turntable bearing

4.1. Contact stress analysis of two point contact turntable bearing

The stress distribution characteristics derived from Hertz theory refer to the distribution of contact compressive stress on the contact surface of objects. Therefore, the extraction of contact surface contact stress in ABAQUS post-processing module can verify the accuracy of the finite element results by comparing with the analytical calculation results of Hertz contact theory. The distribution nephogram of contact stress on the surface of upper and lower raceways is shown in Fig. 3 and Fig. 4.

![Figure 3](image1)

**Figure. 3** Distribution of contact stress between two point contact raceway and rolling element under axia force

![Figure 4](image2)

**Figure. 4** Distribution of contact stress between two point contact raceway and rolling element under radial force

Cpress stress in ABAQUS is the actual pressure on the contact surface of raceway. In Fig. 3, the results of finite element calculation show that the distribution shape of compressive stress on the contact surface between raceway and rolling element is elliptical.

The contact pressure calculation method of rolling ball bearing based on Hertz contact theory, finite element calculation results and analytical calculation results are compared and analyzed, as shown in Table 1 below. Under the axial load, the maximum contact pressure stress of the upper raceway and
the maximum contact pressure stress of the lower raceway are reduced by 15.62% and 16.8% compared with the theoretical solution, and 27.87% and 28.25% respectively under the radial load. The larger the contact force between the rolling element and the raceway, the greater the difference. This is because Hertz contact theory only considers the elastic deformation of the contact body. However, on this basis, the plastic deformation of the raceway and the rolling element is considered in this paper. Therefore, although there is a big difference between the results of finite element calculation and that of Hertz contact theory, the results of finite element calculation are more reliable.

**Table 1. Comparative Analysis of finite element solution and Hertz theory solution**

|                     | Axial load action | Radial load action |
|---------------------|-------------------|--------------------|
|                     | Maximum contact pressure of upper raceway | Maximum contact pressure of lower raceway | Maximum contact pressure of upper raceway | Maximum contact pressure of lower raceway |
| Finite element numerical solution | 2063 | 2064 | 2144 | 2156 |
| Hertz theory solution | 2445 | 2481 | 2931 | 3005 |
| Error results       | 15.62% | 16.8% | 27.87% | 28.25% |

4.2. **Influence of four point contact raceway on contact state of turntable bearing**

By applying axial force and radial force, the contact stress of raceway and the movement change of rolling element are analyzed to explore the change of contact state between rolling element and raceway during loading process.

Under the axial load, the stress distribution characteristics of the contact area between the raceway and the rolling element are shown in Fig. 5. There are two contact areas between the upper raceway and the lower raceway and the rolling element, and the contact stress distribution shape is elliptical. The maximum contact pressure stress of the upper raceway is 2009mpa, and that of the upper raceway is 1972mpa. The stress on both sides of the raceway is uniform, which reduces the contact compressive stress of the raceway and improves the strength of the turntable bearing.

Under the action of radial load, the upper raceway has radial displacement relative to the lower raceway, and there is only an elliptical contact area between the rolling element and the raceway. As shown in Fig. 6, the contact relationship between the rolling element and the raceway is the same as that of the two-point contact turntable bearing. When the four point contact turntable bearing only bears the radial force, the contact relationship between the rolling element and the upper and lower raceways is two-point contact. The contact stress of the upper raceway is 2107mpa and that of the lower raceway is 2111mpa.

![Figure 5](image) Contact stress distribution between four point contact raceway and rolling element under axial force
5. Conclusion

In this paper, the contact models between single rolling element and raceway of single row four point contact turntable bearing and single row two-point contact turntable bearing are established by ABAQUS, and the plastic deformation of contact area of raceway and rolling element in the loading process is considered, and compared with the analysis results of Hertz contact stress, the accuracy of finite element calculation results is verified, At the same time, the axial force and radial force are applied to the two groups of models respectively. When the upper and lower raceway are in contact with each other, the relative displacement of the upper and lower raceway is assumed to be relatively stable when the upper and lower raceway contact modes are used, The rolling element has better radial stability in the new designed raceway.

References

[1] T.A Harris, M.H Mindel. Rolling element bearing dynamics[J].1973, 23(3):311-337.
[2] Pu Lianggui, Chen Yu Mei. Mechanical Design Course [M]. 1998.99-102.
[3] Li Chenggang, Yu Jun. reliability optimization design of special tapered roller bearings [J]. Journal of Huazhong University of technology, 1991 (06): 71-76.
[4] Su Liyue, he Xiangyang. Selection calculation of turntable bearing [J]. Bearing, 2003 (04): 6-9.
[5] Zheng Hongmei, Tian GUI, Liang Changwen, Zheng Yan, Li gengyuan. Study on allowable contact stress of turntable bearing materials based on elastoplastic finite element method [J]. Bearing, 2016 (01): 36-39.
[6] Feng Le, Zhang Jing, Shen Guangfei, Peng Jing, Fei. Dynamic study of roller bearing with local defects based on ABAQUS [J]. Mechanical design and manufacturing engineering, 2018,47 (10): 21-24.
[7] Deng Sier, Hua Xianwei, Zhang Wenhu. Analysis of friction torque fluctuation of gyro angular contact ball bearing [J]. Journal of Aeronautical dynamics, 2018,33 (07): 1713-1724.
[8] Wang Zhihui. Analysis and Research on mechanical Properties of aeronautical rolling Bearing [D]. Nanjing University of Aeronautics and Astronautics, Nanjing,2010,45-47.