Original Research Article

Contact Stress Analysis of the Slewing Bearing of Truck Cranes

Abstract: With the development of industrial technology, lifting equipment has been widely used in various industries. Among them, truck cranes are often chosen in daily use due to their good mobility and rapid transfer. The slewing bearing is an important component in a truck crane that connects the upper and lower parts of the vehicle. Taking a certain model of truck crane as an example, this paper analyzes the contact stress of its slewing bearing by using two methods: Hertz contact theory and finite element method. It also explores the influence of the main structural parameters of the slewing bearing on the contact stress, so as to provide a reference for product selection and design.
Keywords: Truck Crane; Slewing Bearing; Contact Stress; Finite Element Analysis

1Introduction

Industrial equipment is developing towards integration, large-scale and heavy-load, and cranes, as key equipment for material transportation and handling, engineering equipment construction and energy exploitation, are widely used in metallurgy, wind power generation and ship transportation. Compared with other cranes, truck cranes have the advantages of good flexibility, easy installation and disassembly, rapid transfer operation, flexible operation and versatility, and are widely used in various fields[Error! Reference source not found.][1]. The truck crane is mainly divided into the upper part and the lower part, and the dismounted part includes the chassis, driving system, power system and outriggers. The upper part includes a turntable bearing and a slewing mechanism, a crane arm and a hoisting mechanism, among which the turntable bearing is an important part of connecting the upper part and the lower part, which mainly bears the self-weight of the upper part and the overturning moment generated when lifting the heavy object, and at the same time ensures that the upper part can carry out 360° rotary movement relative to itself[Error! Reference source not found.]Therefore, analyzing the contact stress of the slewing bearing of a truck crane is of great significance.

The study of mechanical properties of slewing table bearings has always been a concern of many scholars at home and abroad. Shuguang Qiao et al[Error! Reference source not found.]established the equilibrium equation of force and torque of ball-column combined slewing plate bearing under the combined action of multiple loads from the perspective of the relationship between force and deformation of rolling elements, and analyzed the load distribution of slewing plate bearing. Peter Gondza et al.[6]used a vector method to represent the shape of the slewing ring bearing and the movement of the ring, which was used to determine the contact load of the three-row roller slewing ring bearing, thereby plotting the static load capacity and load relationship curve.Yunfeng Li[Error! Reference source not found.-Error! Reference source not found.]established a mechanical model of all design parameters including clearance of six-row roller turntable bearings for 1,000-ton all-terrain cranes, and systematically analyzed the internal load distribution of turntable bearings under combined loads. Spiewak et al.[8]used the finite element method, ADINA program, analytical Eschmann formula and classical mechanics to calculate the contact load of the double-row slewing support, and comprehensively described the static load capacity profile curve of the slewing bearing under joint load. Congying Gao[Error! Reference source not found.]analyzed the different support modes of single-row four-point contact ball slewing table bearings for tower cranes, and studied the influence of different support modes on the static bearing capacity of slewing bearings. Yanshuang Wang et al.[Error! Reference source not found.]used the Newton-Raphson method to study the influence of different negative clearances on the contact load distribution of double-row four-point contact ball turntable bearings, and analyzed the relationship between negative clearance and bearing contact loads. Marek Krynke[11]proposed a one-dimensional finite element based calculation model, which used a set of nonlinear element trusses to simulate the contact between rollers and raceways, which improved the calculation efficiency of the load distribution of roller turntable bearings. The structure of the slewing ring bearing is shown in**Figure 1**[Error! Reference source not found.]First, a theoretical model of the slewing bearing is established through Hertz contact theory. Then, a solid model is created and analyzed using the finite - element analysis software ANSYS. By comparing the calculation results of the two methods, the accuracy of the results is verified. Analyze the influence of different structural parameters such as contact angle, raceway curvature radius coefficient, rolling element diameter, and number of rolling elements on the contact stress, providing a reference for the selection and design of crane turntable bearings.

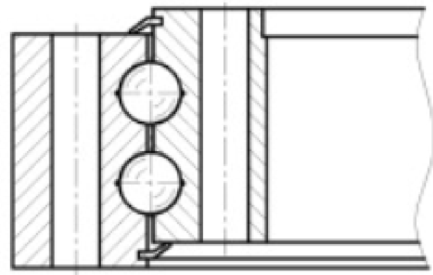

Figure 1 Structural diagram of double-row four-point contact ball bearings

2. Force analysis of slewing bearing

When a crane is in the process of operation, the slewing bearing primarily sustains the combined weight of the superstructure of the crane and the lifted heavy objects, along with the overturning moment engendered during the lifting of heavy objects., as shown in**Figure 2**.

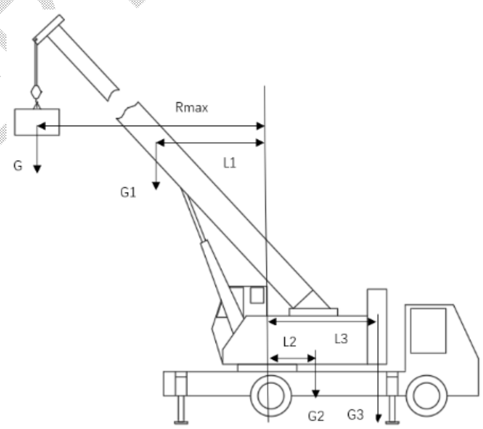

Figure 2 The crane works under force

When selecting and calculating the truck crane slewingbearing, there are usually three different working conditions: (1) the maximum working load when considering the wind, (2) the maximum working load when considering the 125% test load without the wind, and (3) the maximum working load without the wind. Case (1) is mainly used as a dynamic capacity to calculate the load, and case (2) can be used as a static capacity to calculate the load[Error! Reference source not found.Error! Reference source not found.]. In this paper, only the static analysis is carried out, so the working condition (2) is the most consistent. The influence of the radial force is neglected during the calculation process.

As can be seen from Figure 2, the axial load and overturning moment of the slewing bearing are:

$$Fa = 1.25G + G1 + G2 + G3 \quad (1)$$

$$M = 1.25G \times Rmax + G1 \times l1 - G2 \times l2 - G3 \times l3 \quad (2)$$

where: G is the lifting weight: G1 is the weight of the crane boom; G2 is the weight of the crane slewing table; G3 is the Weight of the Counterweight; Rmax is the working radius of the crane; L1, L2 and L3 are the center distances from the center of the boom, the center of the slewing Table and the center of the balance weight to the center of the slewing bearing, respectively.

Take a certain working condition of a 70t truck crane of a certain type as an example. Its main parameters are shown in**Table 1**.

Table 1

70t crane working condition main parameters

property	Value
Lifting weight	23.7t
Boom weight	13t
Slewing TableWeight	9t
Weight of the Counterweight	17t
Rmax	7m
L1	3.5m
L2	1.75m
L3	3.5m

* take the gravitational acceleration g=10N/kg

Bringing the parameters into equation (1) and equation (2), it can be obtained that the axial force of the slewing bearing is 686.25KN, and the overturning moment is 1776.25KN/m.

3 Theoretical calculations

In the case of four-point contact ball bearings, the ball is in contact with the raceway only at the point of contact under no load, and the contact area gradually becomes elliptical after being loaded**Error! Reference source not found.**.

According to the Hertzian contactformula, the maximum stress of the rolling elements in contact with the raceway is:

$$\sigma_{max} = \frac{3Q}{2\Pi ab} \quad (3)$$

$$a = a^*[\frac{3Q}{2\Sigma\rho}\left(\frac{1-\xi_I^2}{E_I}\right) + \left(\frac{1-\xi_{II}^2}{E_{II}}\right)]^{\frac{1}{3}} \quad (4)$$

$$b = b^*[\frac{3Q}{2\Sigma\rho}\left(\frac{1-\xi_I^2}{E_I}\right) + \left(\frac{1-\xi_{II}^2}{E_{II}}\right)]^{\frac{1}{3}} \quad (5)$$

$$\delta = \delta^*[\frac{3Q}{2\Sigma\rho}\left(\frac{1-\xi_I^2}{E_I}\right) + \left(\frac{1-\xi_{II}^2}{E_{II}}\right)]^{\frac{2}{3}}\frac{\Sigma\rho}{2} \quad (6)$$

where: Q is the normal force between the rolling element and the raceway; *a* is the radius of the major axis of the contact ellipse; b is the radius of the minor axis of the contact ellipse; δ is the relative approach of the contact remote control point; ∑ρ is the curvature and function; F(ρ) is the curvature difference function; ξ is the Poisson's ratio; E is the modulus of elasticity; a^* is the semi-major axis of the contact ellipse dimension 1; b^* is the minor semi-axis of the contact ellipse dimension 1; δ^* is the contact displacement with dimension 1.

For steel bearings, it can be simplified to:

$$a = 0.0236a^* \left(\frac{Q}{\Sigma\rho}\right)^{\frac{1}{3}} \quad (7)$$

$$b = 0.0236b^* \left(\frac{Q}{\Sigma\rho}\right)^{\frac{1}{3}} \quad (8)$$

$$\delta = 2.79 \times 10^{-4}\delta^* Q^{\frac{2}{3}}\Sigma\rho^{\frac{1}{3}} \quad (9)$$

When theslewing bearing is only subjected to an axial force, the normal load on the rolling elements is:

$$Q_{max} = \frac{F_a}{Zsin\alpha} \quad (10)$$

When only subjected to the action of the overturning moment, the normal load borne by the rolling elements is:

$$Q_{max} = \frac{4.37M}{d_m Zsin\alpha} \quad (11)$$

Considering the influence of factors such as manufacturing and installation errors and material inhomogeneity, the maximum load distribution of the double-row balls is not the same, the load distribution is 0.45 and 0.55.According to the empirical formula, the normal load on the most - loaded rolling element is：$Q_{max} = 0.55\left(\frac{F_a}{Zsin\alpha} + \frac{4.37M}{d_m Zsin\alpha}\right)$ (12)

In this paper, a double-row four-point contact ball slewing bearing is selected, and the inner and outer rings and rolling elements are made of 42CrMo, and the main parameters of the bearing are shown in **Table 2**.

Table 2

The main parameters of double-row four-point contact ball slewing table bearings

Parameter	Value
Pitch diameter dm/mm	1612
Ball nominal diameter D_w/mm	30
Initial contact anglea_0/°	45
Distance of the center of the two balls between the upper and lower row d_c/mm	50
Coefficient of curvature of inner and outer raceways f_i、f_e	0.525
Number of ball Z	118×2
Elastic modulus of ball and rings E/GPa	210
Poisson's ratiov	0.28

According to the above parameters, the maximum contact stress is calculated to be 2646MPa, and the contact deformation is 0.0528mm.

4 Finite element analysis

4.1 Meshing and Contact

An entity finite - element model is adopted for analysis. To reduce the computational time, a part of the model is selected for analysis. During the meshing process, the size of the mesh affects the solution speed and the accuracy of the results. The contact between the rolling elements and the raceways only occurs in the vicinity of the contact points. Based on "Saint Venant's Principle", only the meshes in the contact area need to be refined.According to the radius of the major axis of the contact ellipse, the influence radius of the virtual sphere is set to 5mm, and the size of the virtual ball element is 0.33mm according to the radius of the minor axis of the contact ellipse, and the size of the remaining elements is automatically generated. The overall model uses a tetrahedral mesh, and the meshing results**Error! Reference source not found.Figure 3**. When setting the contact surface and the target surface, the following principles should be followed:

(1)、If a convex surface contacts a flat surface or a concave surface, the flat surface or the concave surface should be the target surface.

(2)、The surface with a denser mesh is the contact surface, while the surface with a coarser mesh is the target surface.

(3)、If one surface has a larger elastic modulus than the other, the surface with a smaller elastic modulus should be designated as the contact surface, and the surface with a larger elastic modulus should be the target surface.

(4)、The high - order contact element is the contact surface, and the low - order contact element is the target surface.

(5)、If one surface is significantly larger than the other, the larger surface should be the target surface[15].

Based on the above principles, the rolling elements are selected as the contact surface, and the raceway is selected as the target surface. After bearing a load, the contact area between the rolling elements and the raceway is elliptical. A surface - to - surface contact is adopted, the contact type is frictional contact with a friction coefficient of 0.1, and an asymmetric contact behavior is used.

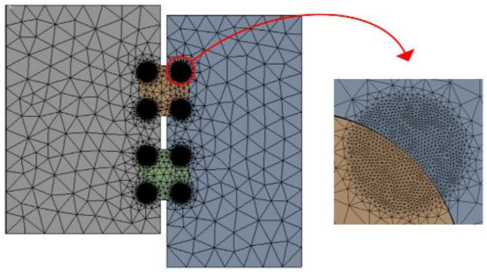

Figure 3 Mesh of the monolithic model

4.2 Loads and Constraints

Apply fixed constraint supports to the upper and lower surfaces as well as the side surface of the outer ring of the bearing, and apply frictionless supports to the cross - section of the bearing. When the bearing is only subjected to the axial force, the normal load on the bearing is Fa/Z = 2.908kN. When the bearing is only subjected to the overturning moment, the normal load on the bearing is 4.37M/(dm×Z) = 20.403 kN. The initial contact angle is 45°. Therefore, the axial force generated by the overturning moment is 14.425 kN, and the radial force is 14.425 kN. According to the superposition principle, an axial load of 8.667kN is applied to the upper surface of the inner ring of the bearing, and a radial load of 7.213 kN is applied to the side surface.

4.3 Finite Element Analysis Results

As shown in**Figure 4**, the maximum contact stress between the rolling element and the raceway is 2566.8MPa, which is 3% different from the theoretical calculation result, and the total deformation is 0.051598mm, which is 2.3% different from the theoretical calculation result.Due to the assumption of Hertzian theory, there are some errors between theoretical calculation results and finite element analysis.As can be seen from**Figure 4**, after the bearing is loaded, it exhibits a four - point contact state. The maximum contact stress occurs at the contact part between the upper - row rolling elements and the inner ring.

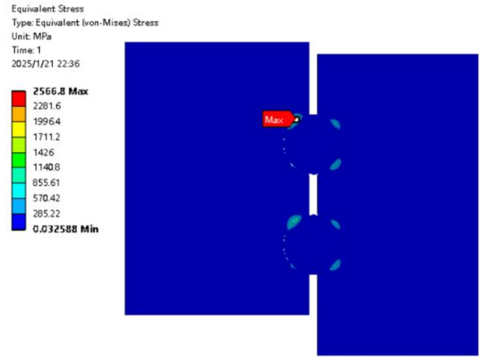

Figure 4 Bearing equivalent stress

5The Influence of Main Structural Parameters on the Contact Stress of Slewing Bearings

For slewing bearings, most of their structural parameters have been standardized, such as the inner and outer diameters, height, and the diameter of mounting holes. However, some parameters, like the raceway curvature radius coefficients of the inner and outer rings, the initial contact angle, the number of balls, and the diameter of balls, etc., also have a significant impact on the contact stress of the bearings[Error! Reference source not found.]. Analyzing these structural parameters can provide references for bearing selection and manufacturing.

The raceway curvature radius coefficients of the inner and outer rings are taken as 0.505, 0.515, 0.525, 0.535, 0.545 and 0.555 respectively; the initial contact angles is taken as 45°, 50°, 55°, 60°, 65° and 70°respectively; the diameters of the rolling elements is taken as 30mm, 31.5mm, 33mm, 34.5mm, 36mm, 37.5mmrespectively; and the numbers of rolling elements is taken as 176, 188, 200, 212, 224, 236respectively. The influence of these parameters on the bearing contact stress is shown in**Figure 5**.

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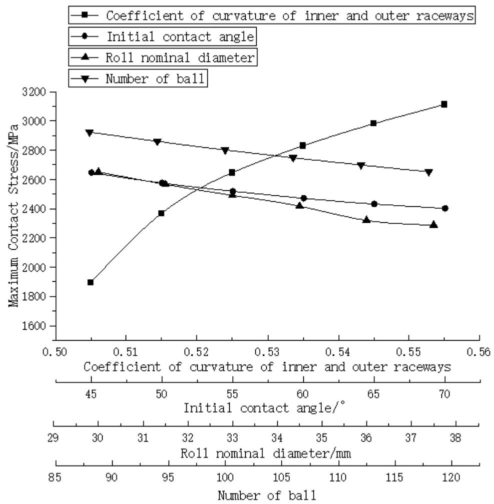

Figure 5 The influence of the main structural parameters on the contact stress of the slewing bearing

As can be seen from **Error! Reference source not found.**, the contact stress between the ball and the raceway increases with the increase of the raceway curvature radius coefficient, and decreases with the increase of the initial contact angle, ball diameter, and the number of balls. The main reason is that as the raceway curvature radius coefficient increases, the area of the contact ellipse between the ball and the raceway after bearing the load decreases, resulting in an increase in contact stress. As the initial contact angle increases, the normal load generated by the axial load and the overturning moment load decreases, leading to a decrease in contact stress. When the ball diameter remains unchanged and the number of balls increases, since the number of balls participating in load - bearing increases, the contact stress generated by each ball will be smaller. And, when the number of balls remains unchanged and the diameter increases, the area of the contact ellipse increases, thus resulting in a decrease in the contact stress between the ball and the raceway.

6. Conclusion

(1)、According to the actual working conditions of the truck crane, 125% of the working load is selected to analyze the load on the slewing bearing, and the mechanical model is established, and the finite element results and the theoretical model results are compared, and the correctness of the results is verified.

(2)、From the results of the finite - element analysis, the maximum contact stress occurs in the contact area between the upper row of rolling elements and the raceway. Therefore, during the process of model selection and design, some measures can be taken to reduce the contact stress of the bearing. For example, the diameter of the upper - row rolling elements can be made larger than that of the lower - row rolling elements, the curvature radius coefficient of the upper - row raceway can be decreased, and the initial contact angle between the upper - row rolling elements and the raceway can be increased.

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[1] Yang Lei, Dong Quanzhong, Liu Haifeng. Safety management of lifting with 800 t truck crane in complex environment[J]. Construction Safety, 2020, 35(12): 27-31.https://www.doc88.com/p18461747747338.html

[2] Xu Bin. Design of QY25 truck crane[D]. Dalian: Department of Mechanical Engineering, Dalian University of Technology, 2003.https://www.doc88.com/p59559826803557.html

[3] Dai Jun. Introduction to Truck Crane[J]. Mechanical and Electrical Equipment, 2006, 23(4): 63-65,72.https://www.doc88.com/p78261782260580.html

[4] Zhu Xuemin. Hoisting Machinery[M]. Machinery Industry Press, 2003.

[5] Qiao Shuguang, Xu Mingqi, Wen Jingbo, et al. Calculation of static load capacity of ball-column combined turntable bearing [J]. Mechanical Transmission, 2011, 35(9): 27-29.https://www.doc88.com/p-9982671047055.html

[6] Göncz P,Potočnik R, Glodež S. Computational model for determination of static load capacity of three-row roller slewing bearings with arbitrary clearances and predefined raceway deformations[J]. International Journal of Mechanical Sciences, 2013, 73: 82-92.
https://www.doc88.com/p-9179558892348.html

[7] Li Yunfeng, Jiang Di. Analysis on the load-bearing capacity of slewing bearings for 1,000-ton all-terrain cranes[J]. Journal of Mechanical Engineering, 2016, 52(1): 109-115.
https://www.doc88.com/p09839411254737.html

[8] Śpiewak S. Methodology for calculating the complete static carrying capacity of twin slewing bearing[J]. Mechanism and Machine Theory, 2016, 101: 181-194.
https://www.sciencedirect.com/science/article/pii/S0094114X16300106

[9] Gao Congying. Finite element analysis and design parameter optimization of turntable bearing for tower crane[D]. Henan: Henan University of Science and Technology, 2016.
https://www.doc88.com/p21899555417958.html

[10] Wang Yanshuang, Yuan Qianqian. Effect of negative clearance on load distribution of extra-large double-row four-point contact ball bearings[J]. Journal of Mechanical Engineering, 2012, 48(21): 110-115.
https://www.doc88.com/p-9552389593130.html

[11] Krynke M, Borkowski S, Selejdak J. Analysis of influence of bearing clearance on the static carrying capacity of multi-row slewing bearings[J]. PeriodicaPolytechnica Transportation Engineering, 2014, 42(1):4.

[12] Zhang Ping, Xu Xuelin, Li Xinhua, et al. Finite element calculation and analysis of slewing bearing of truck crane [J]. Engineering Machinery, 2008, 39(5): 42-45.
https://www.doc88.com/p-10987111022703.html

[13] Zhang Zhiwen, Yu Heqian, Wang Jinnuo, et al. Crane Design Manual[M]. China Railway Publishing House, 1998.

https://www.doc88.com/p6488636344558.html?s=rel&id=2
[14] Deng Sier. Principles of rolling bearing design[M]. China Standards Press, 2014.
[15] Harris T A, Kotzalas M N. Advanced concepts of bearing technology: rolling bearing analysis[M]. CRC press, 2006.
https://www.doc88.com/p53247430517003.html?s=rel&id=2
[16] Tian Jianhui. ANSYS 18.0 Finite Element Analysis and Engineering Application[M]. Machinery Industry Press, 2019.
[17] Zhu Lin, Zhang Gang, Ni Xiaoting, et al. Mechanical performance analysis of wind turbine pitch bearings[J]. Industrial Control Computer, 2015(8):106-109.
https://jz.docin.com/p-1594935748.html