Study on Dynamics Properties of High Speed Ball Bearing Considering Eccentric Load of Shaft

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***Abstract*—Taking 7309 ball bearing as the research object to study the dynamic characteristics of high speed ball bearing which considering eccentric load of shaft. The finite element model is established by ANSYS/LS-DYNA. The load is applied on the bearing and the dynamics simulation analysis is carried out. Bending moment is used as a variable to analyze displacement, stress, velocity and acceleration of the bearing. Stress distribution and vibration of the bearing with different bending moment are analyzed. Simulation results show that when the eccentric load increases, the maximum stress of each part of the bearing rises, the frequency of the impact stress increases. The analytical method and model in the present paper could aid in the design of ball bearings.**

Keywords-component; ball bearing; finite element analysis; dynamic analysis; eccentric load

I. INTRODUCTION

Ball bearings are widely used in engineering, and are indispensable mechanical components for transmitting rotating motion and supporting loads. It is very important to analyze the related dynamics of the complicated mechanical relations among the parts in the process of bearing rotation.

Seong-Wook Hong [1] devoted significant efforts to modeling and analysis of rolling-element bearings to aid in their design and application and reviewed the modeling and analysis of rolling-element bearings with emphasis on single-row ball and roller bearings. The application of bearing models was reviewed as well along with illustrative results. Cui [2] constructed the finite element analysis model of rolling bearing. Rotating speeds and constraints are exerted on the balls and cage, after that the explicit dynamic simulation of rolling bearing is simulated and the stress distribution and dynamic response of rolling bearing are obtained. Effects of the radial clearance, pocket-hole clearance on the dynamic characteristics of the cage are analyzed. Liu [3] introduces oil-film damping and hysteresis damping, and deals with the collision contact as imperfect elastic contact. In addition, the effects of inner ring rotational speed, the ratio of pocket clearance to guiding clearance and applied load on the cage stability are investigated by simulating the cage motion with the model. The results can provide a theoretical basis for the design of ball bearing parameters. Jiangshan Liu [4-6] taken the deep groove ball bearing as a research object, which is imported into simulation software LS-DYNA for finite element analysis. The change regularity of contact stress, rotating speed and vibration of ball bearing are researched during operation process, and the feasibility of simulation is verified by theoretical calculation. Chunrong Lu [7-10] proposed a dynamic model of gear-rotor-bearing system, and proposed a dynamic model of gear-rotor-bearing system considering dynamic backlash, tooth face friction, gear eccentricity and time-varying mesh stiffness. And the dynamical characteristics with the effects of rotational speed, gear eccentricity, bearing clearance are analyzed. Chao-Feng[11] deduced a mathematical model of angular contact ball bearing with Hertz contact theory considering the axial and radial loads. With the coupling effects of lateral, torsional and axial vibrations taken into account, a lumped-parameter nonlinear dynamic model[12] of helical gear rotor-bearing system (HGRBS) is established to obtain the transmission system dynamic response to the changes of different parameters. The vibration differential equations of the drive system are derived through the Lagrange equation, which considers the kinetic and potential energies, the dissipative function and the internal/external excitation. Based on the Runge-Kutta numerical method[13], the dynamics of the HGRBS is investigated, which describes vibration properties of HGRBS more comprehensively. At present, the dynamic finite element analysis of the bearing seldom considered the effect of the axial deflection of the inner ring on the bearing dynamic characteristics when the bearing bears eccentric load in the actual working condition.

In this paper, taking 7309 ball bearing as the research object, the author establishes finite element model in the software ANSYS/LS-DYNA, using the explicit dynamic method to analyze dynamic characteristics of ball bearings, and process simulation results in the LS-PREPOST post-processing software, thus to get displacement, stress, velocity and acceleration time-history curve of bearings, and analyze the stress distribution and vibration of the inner ring in different deflection angles, which provides theoretical basis for the design and development and fault diagnosis of bearings .

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II. MODELLING OF BALL BEARING

As shown in Fig. 1, the mass of the rotating shaft itself and the pressure load F on the rotating shaft will generate unbalanced pressure on both sides of the bearing under the actual working conditions, resulting in deflection of the rotating axis of the rotating shaft. The pressure load F is changed to a pressure load F′ having a deflection angle, and F′ is decomposed into a radial load F_r and an eccentric load F_a. The radial load F_r causes a radial displacement d of the inner ring of the bearing, and the eccentric load F_a causes the inner ring of the bearing to generate a deflection angle α.

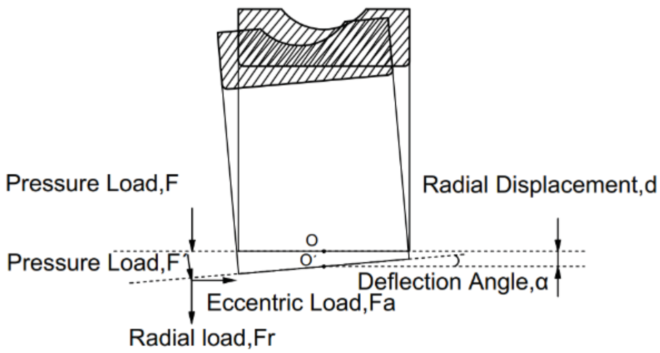


Figure 1. Schematic diagram of inner ring deflection

The dynamic model of ball bearing can be written as:

$$m_i \ddot{y}_i = \sum_{j=1}^{Z} (-Q_i^j \cos\varphi_j + F_i^j \sin\varphi_j) - F_r - G_i \tag{1}$$

$$m_i \ddot{z}_i = \sum_{j=1}^{Z} (Q_i^j \sin\varphi_j + F_i^j \cos\varphi_j) \tag{2}$$

$$J_i \dot{\omega}_{ix} = \sum_{j=1}^{Z} (F_i^j \frac{D_w}{2}) \tag{3}$$

$$J_i \dot{\omega}_{iy} = \sum_{j=1}^{Z} (-M_{i,mis}^j \sin\varphi_j + M_{i,d}^j \sin\varphi_j) \tag{4}$$

$$J_i \dot{\omega}_{iz} = \sum_{j=1}^{Z} (M_{i,mis}^j \cos\varphi_j + M_{i,d}^j \cos\varphi_j) \tag{5}$$

Where, F_r is radial load of ball bearing. m_i is mass of inner ring. G_i is gravity of inner ring. $\ddot{x}_i$, $\ddot{y}_i$, $\ddot{z}_i$ are acceleration of inner ring. $\dot{\omega}_{ix}$, $\dot{\omega}_{iy}$, $\dot{\omega}_{iz}$ are angular acceleration of inner ring.

III. FINITE ELEMENT MODEL OF BALL BEARING

A. Finite element model

The dimensions of the bearing are shown in Table. Ignoring some chamfering and edges, SOLID164 unit is used to establish the finite element model in a bottom-up way. The inner ring of the bearing is simplified as a rigid body, and the other parts are set as linear elastic materials. Tetrahedral elements were used to divide the grid, and local mesh refinement was carried out in the contact area. A total of 35806 nodes and 157430 elements were formed.

TABLE . GEOMETRIC PARAMETERS OF BEARINGS

Name	**Size**
Outer Diameter of Bearing D_o/mm	100
Internal Diameter of Bearing D_i/mm	45
Bearing width B/mm	25
Rolling Element Number	11
Diameter of Rolling Body D/mm	16.669

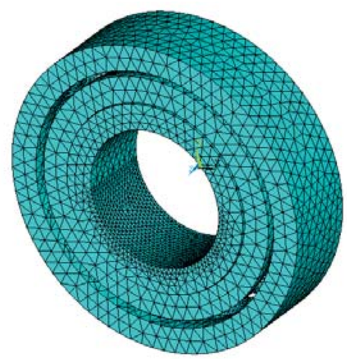


Figure 2. Finite element model of ball bearing

B. Contact, load and boundary conditions

The friction factor μ_C of ANSYS/LS-DYNA is determined by equation (6):

$$\mu_C = f_D + (f_S - f_D)e^{-DC \cdot V_{rel}} \tag{6}$$

Where: f_S-static friction factor; f_D-dynamic friction factor; ;DC -exponential attenuation coefficient; V_{rel} -relative velocity between contact surfaces, which is automatically calculated by the program according to the contact condition.

The rational speed load and pressure load are applied to the bearing. A uniform speed motion is simulated by applying 5000 r/min. The lower half of the inner ring of the bearing is subjected to F_r of 2000N.

By simulating the working condition of the bearing fixed in the high-stiffness bearing seat, all degrees of freedom of the outer surface of the bearing outer ring are restrained, while the degree of freedom of rotation in the X and Y directions of the bearing inner ring is restrained, and the rest parts are not restrained.

IV. ANSLYSIS AND RESULTS

The deflection angles are set as 0 mrad, 5 mrad, 10 mrad, and 20 mrad, respectively. The simulation was performed using the LS-DYNA solver as described above. The calculation time was 60ms. The simulation results were viewed in ANSYS and LS PREPOST.

A. Analysis of axial displacement of cage

Select a node on the cage to compare the axial displacement of the cage under different deflections and analyze the stability during the rotation of the cage.

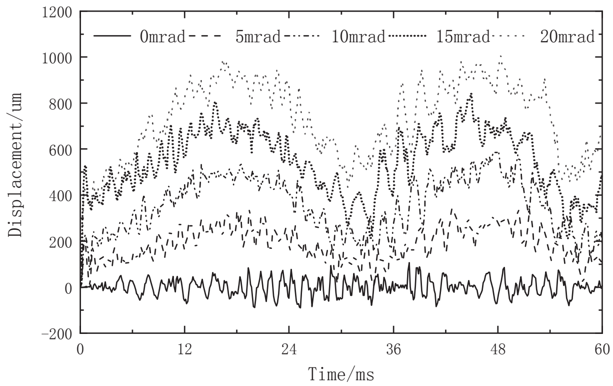
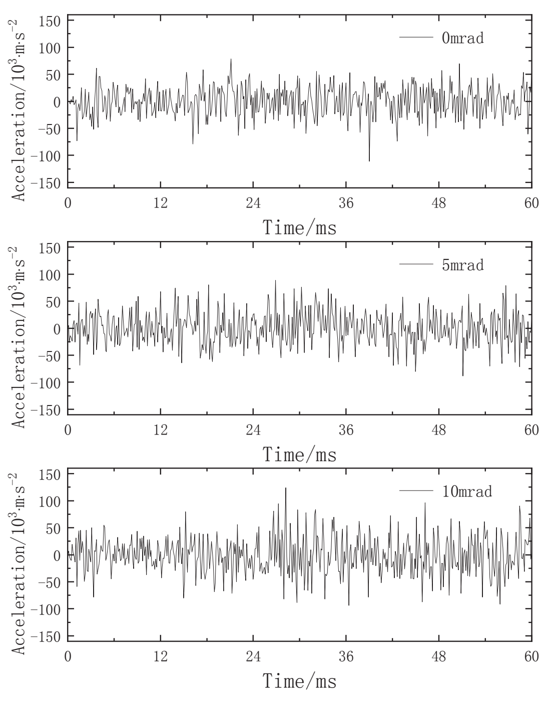


Figure 3. Axial displacement of the cage

As shown in Fig. 3, when the inner ring of the bearing is not deflected, the axial displacement of the cage changes little and the operation is stable. The greater the deflection angle of the inner ring is, the greater the axial displacement of the cage will be. At this time, the cage cannot maintain a stable rotating motion, but periodic shaking at the same time of the rotating motion, which intensifies the collision between the cage and the roller, greatly reducing the stability of the bearing.

B. Analysis of dynamic responses of bearing

As shown in Fig. 4, the nodes of inner ring, roller and retaining frame are selected respectively to make acceleration time-history curve and analyze the vibration of each bearing part in the rotation process.



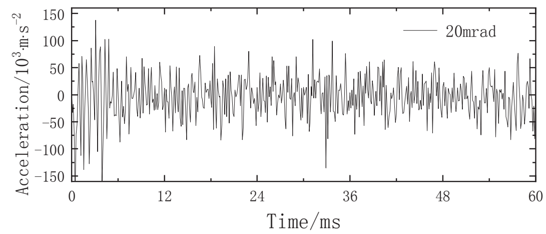


Figure (a) The acceleration of bearing cage

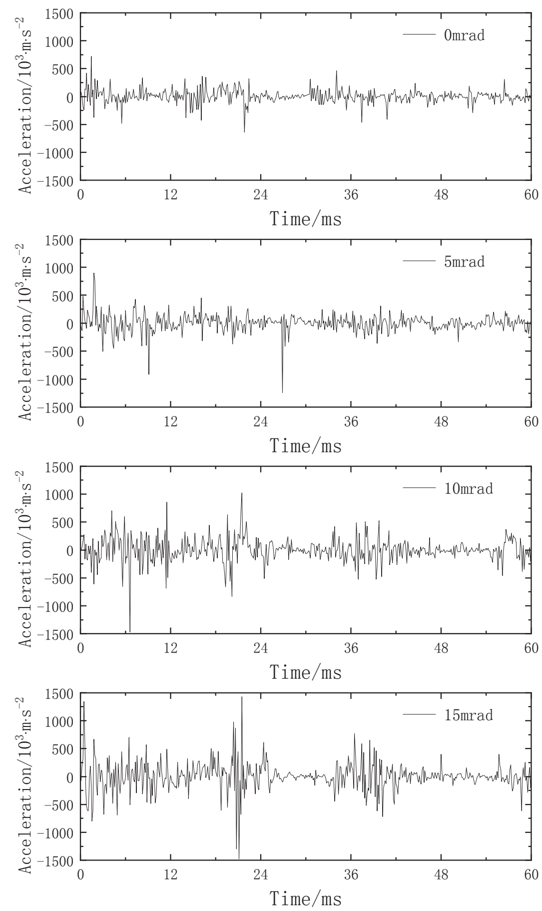


Figure (b) The time history curve of bearing roller acceleration

Figure 4. Acceleration time history curve of each part of the bearing

According to Fig. 4, the larger the deflection angle of the inner ring is, the wider the vibration range of the roller and the cage is, the higher the frequency of impact stress, and the worse the stability will be. The periodic impact vibration in the roller time-history curve is caused by the contact collision between the node on the roller and the bearing part.

C. Time-maximum equivalent stress analysis

The position and size of maximum stress are different in bearing areas. The time-history curves of maximum stress of bearing elements at different times are shown in Fig. 5.

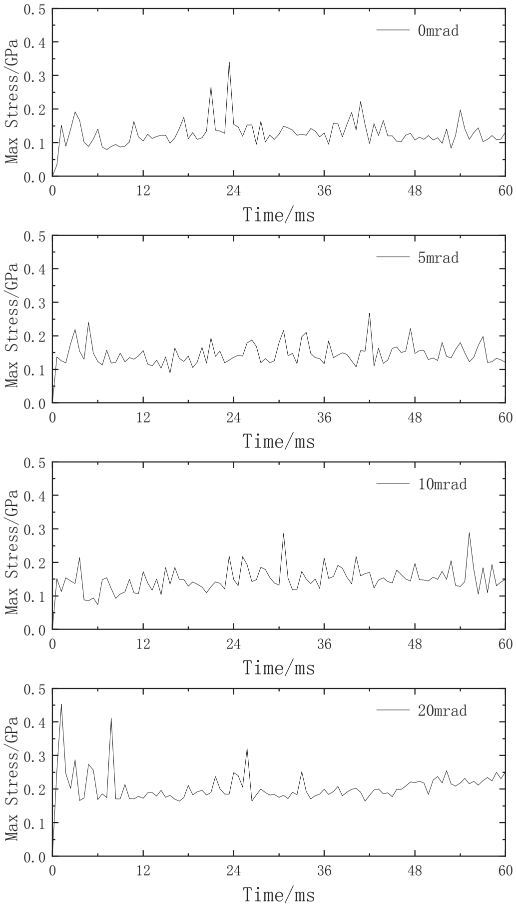
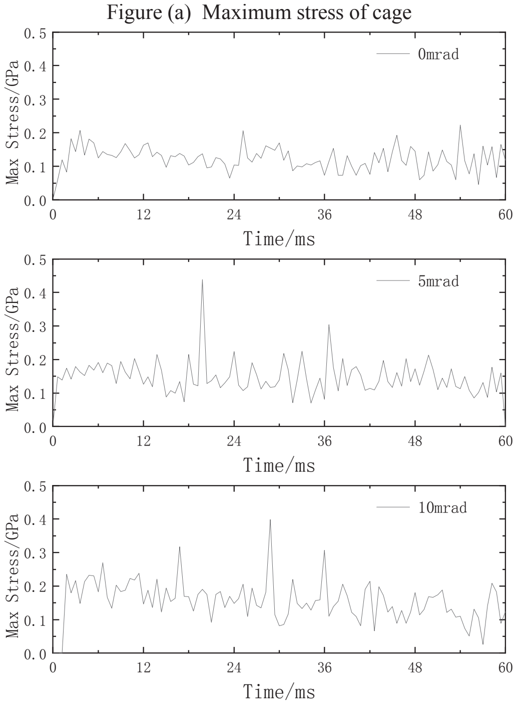


Figure (a) Maximum stress of cage



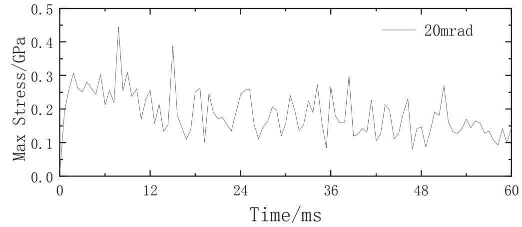


Figure (b) Outer ring maximum stress

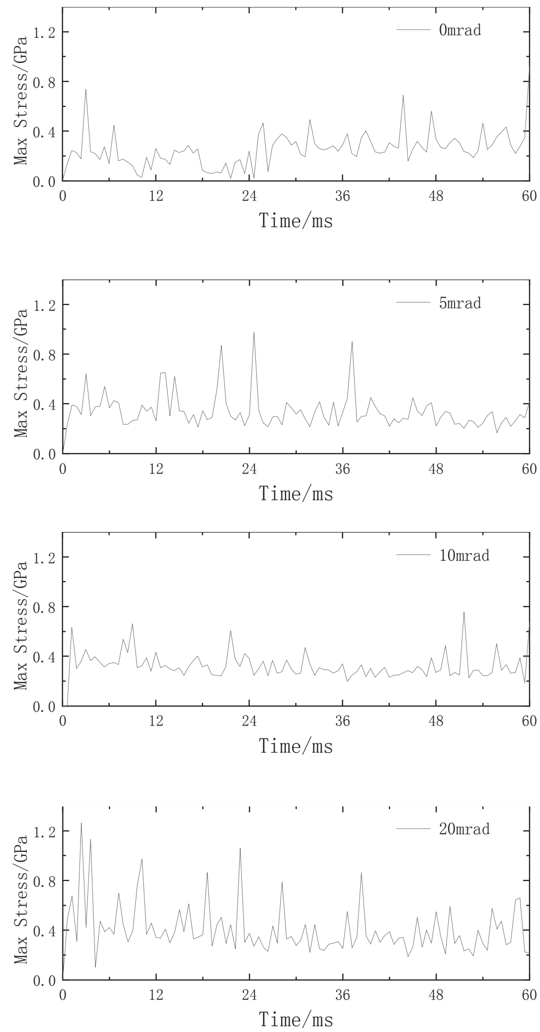


Figure (c) Ball maximum stress

Figure 5.   Maximum stress of bearing

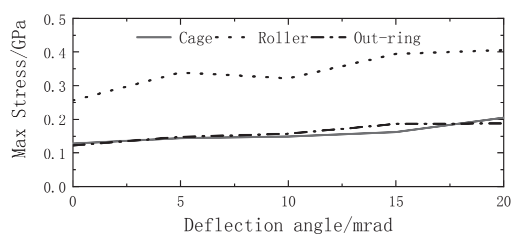


Figure 6.   Compare of maximum stress of each part of the bearing

As can be seen from Fig. 5 and Fig. 6, with the increasing deflection angle, the frequency of impact stress of each bearing part increases, stability decreases, and the average value of maximum stress of each part increases. The maximum stress of roller is generally larger than the maximum stress of cage and outer ring.

V. CONCLUSIONS

This paper takes 7309 ball bearing as the research object, establishes a finite element model in software ANSYS/ LS-DYNA to analyze the dynamic characteristics of ball bearing, obtains the displacement, stress, velocity and acceleration time-history curve of the bearing, and analyzes the stress distribution and vibration of the inner ring under different deflection angles. The conclusions are as follows:

(1) When the eccentric load is zero, the axial displacement of the cage is almost zero, the operation is stable. As the eccentric load increases, the axial displacement of the cage increases, the cage cannot maintain a stable rotary motion, the collision between the cage and balls intensified, and the bearing stability is greatly reduced, which will result in the decrease of bearing life.

(2) As the eccentric load increases, vibration amplitude of balls and cage increase. The frequency of the impact stress also increase. The stability of bearing will become worse.

(3) As the eccentric load increases, the frequency of impact stress of each part of the bearing increases, the stability decreases, and the maximum stress of each part rises.

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