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The Analysis of The Bearing Fit Effect on Spindle Rotor System Vibration

Shiyu Xing[1, 2], Jian Ma[1], Aoxiang Liu[3], Siqi Niu[1, *], Binbin Hu[1]

[1]School of Mechanical Engineering, Shenyang Jianzhu University, Shenyang, China
[2]Shenyang Piotec Technology Co. Ltd, Shenyang, China
[3]North Huajin Chemical Industries Group Corporation, Panjin, China
[*]Corresponding author: 1404589743@qq.com

Abstract: Bearing fit is closely related to the dynamic characteristics of the spindle system. When the spindle rotates at high speed, affected by centrifugal force, load and other factors, the bearing fit will change. In this paper, the finite element method is adopted. A dynamic model of motorized spindle is established to analyze the relationship between bearing clearance and bearing stiffness. Bearing stiffness directly affects the dynamic characteristics of the spindle rotor system. The machining accuracy of spindle system decreases with the increase of clearance amount. The establishment of dynamic model of motorized spindle provides theoretical guidance for structure design of motorized spindle.

Keywords: Motorized spindle, Bearing, Fit mode, Vibration.

1. Introduction

Motorized spindle has been widely used in high precision machining because of its high precision machining[1-2].The matching mode of bearing determines the machining quality of spindle. The clearance fit between the bearing outer ring and the bearing seat is one of the important components of the fit mode. Clearance fit is related to bearing stiffness[3-4]. Therefore, it is very important to develop the connection between clearance fit and dynamic characteristics of spindle.

Some scholars carry out some research on the dynamic characteristics of spindle around the bearing mating mode. Chen et al. analyzed the relationship between dynamic characteristics of bearing and noise[5]. Ma et al. introduced the measurement of bearing vibration through BVT-6 instrument[6]. Yang et al. developed a spindle-bearings model to study the spindle stiffness with different bearing configurations[7]. Li et al. investigated the influence of a new non-uniform preload on the spindle static and rotating performance[8]. Although some authors have carried out relevant studies, the effect of intermittent fit on the vibration of motorized spindle rotor system has been ignored.

In the paper, a dynamic model of motorized spindle is established through finite element method, which aims to analyze the relationship between bearing clearance and spindle rotor system vibration. The establishment of dynamic model of motorized spindle provides theoretical guidance for structure design of motorized spindle. Experiments of different bearing clearance fits and rotating speeds are conducted on the motorized spindle test platform. The work lays a theoretical basis for further optimizing the assembly relationship and structural design of motorized spindle.

2. Model of Spindle Rotor-bearing System

The operation of the inner and outer rings of the bearing in the motorized spindle is relatively simple, and the outer rings are fixed. The inner ring rotates circumferentially with the axis of rotation, but the motion of the ball is complicated, and there are two motion modes: revolution and autobiography. And because of the existence of gyroscopic moment, the ball will also produce skid phenomenon. According to the bearing movement law, the bearing in a certain instantaneous movement coordinate system is established, as shown in Figure 1.

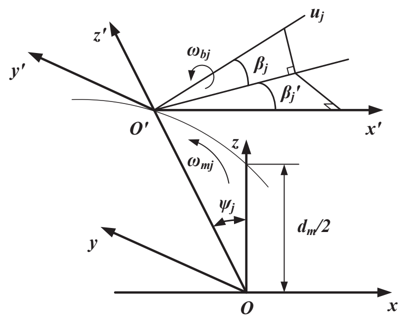


Figure 1. Bearing motion coordinate system

And the oxyz coordinate system is a fixed coordinate system, the x axis coincides with the bearing axis, $o'x'y'z$ coordinate system is a rotating coordinate system, $x'$is parallel to the x axis, $o'$is the spherical center of the ball. The components of ω_{bj} in the rotating coordinate system are respectively ω_{bjx}, ω_{bjy} and ω_{bjz}, which can be expressed as Equation 1.

$$\begin{aligned} \omega_{bjx} &= \omega_{bj}\cos\beta_j\cos\beta_j' \\ \omega_{bjy} &= \omega_{bj}\cos\beta_j\sin\beta_j' \\ \omega_{bjz} &= \omega_{bj}\sin\beta_j \\ \omega_{bj} &= \sqrt{\omega_{bjx}^2+\omega_{bjy}^2+\omega_{bjz}^2} \end{aligned} \tag{1}$$

According to the pseudo-static theory of bearings, through the geometric relationship of bearing, the inner and outer ring raceway groove curvature centers can be expressed as Equation 2.

$$\begin{cases} A_{1j} = \overline{AC}\sin\alpha + \overline{CC_z'} = (f_e+f_i-1)d_w\sin\alpha+\delta_z+R_i\theta_x\cos\psi_j-R_i\theta_y\sin\psi_j+u_a \\ A_{2j} = \overline{AC}\cos\alpha + \overline{CC_y'} = (f_e+f_i-1)d_w\cos\alpha+\delta_x\sin\psi_j+\delta_y\cos\psi_j+\Delta u \end{cases} \tag{2}$$

Where f_i and f_e are the groove curvature radius coefficients of inner and outer ring raceways. δ_x, δ_y, δ_z, θ_x and θ_y are the translational and angular displacements of bearing inner ring. ψ_j is the angular position of the jth bearing ball, R_i is the radius of bearing inner ring groove curvature center. The relative displacement Δu between the inner and outer ring raceway groove curvature centers is caused by interference fit can be expressed as Equation 3.

$$\Delta u = u_r + u_f \tag{3}$$

The outer ring fit of spindle angular contact ball bearing is usually a small clearance fit. Under the influence of thermal deformation, the clearance fit value C can be expressed as Equation 4.

$$C = C_0 - a_p \times (1+\mu_p) \times d_e + a_e \times \Delta T_e \times D \tag{4}$$

Where a_p and a_e are the thermal expansion coefficients of bearing pedestal and outer ring. C_0 is the initial clearance fit value, μ_p is Poisson's ratio of bearing pedestal and D is the diameter of bearing outer ring.

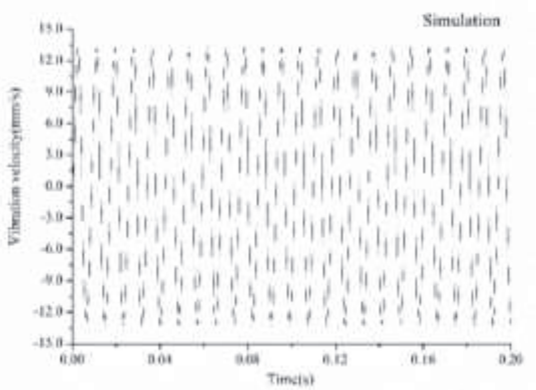
According to the Newton's second law, the dynamic equation of the rotor-bearing system can be expressed as Equation 5.

$$\begin{cases} m_1\ddot{x}_1 + c_{12}\left(\dot{x}_1 - \dot{x}_2\right) + k_{12}\left(x_1 - x_2\right) = m_1 e\omega^2 \cos\omega t \\ m_1\ddot{y}_1 + c_{12}\left(\dot{y}_1 - \dot{y}_2\right) + k_{12}\left(y_1 - y_2\right) = -m_1 g + m_1 e\omega^2 \sin\omega t \\ m_2\ddot{x}_2 + \left(c_{12} + c_{23}\right)\dot{x}_2 - c_{12}\dot{x}_1 - c_{23}\dot{x}_3 + \left(k_{12} + k_{23}\right)x_2 - k_{12}x_1 - k_{23}x_3 \\ = -c_{n1}\dot{x}_2 - \sum_{i=1}^{N_1} k_{n1}\left[\left(x_2\cos\theta_i + y\sin\theta_i + C\right)\cos\alpha_1\right]^{1.5}\cos\theta_i \\ m_2\ddot{y}_2 + \left(c_{12} + c_{23}\right)\dot{y}_2 - c_{12}\dot{y}_1 - c_{23}\dot{y}_3 + \left(k_{12} + k_{23}\right)y_2 - k_{12}y_1 - k_{23}y_3 \\ = -m_2 g - c_{n1}\dot{y}_2 - \sum_{i=1}^{N_1} k_{n1}\left[\left(x_2\cos\theta_i + y_2\sin\theta_i + C\right)\cos\alpha_1\right]^{1.5}\sin\theta_i \\ m_3\ddot{x}_3 + \left(c_{23} + c_{34}\right)\dot{x}_3 - c_{23}\dot{x}_2 - c_{34}\dot{x}_4 + \left(k_{23} + k_{34}\right)x_3 - k_{23}x_2 - k_{34}x_4 \\ = m_3 e\omega^2 \cos\omega t \\ m_3\ddot{y}_3 + \left(c_{23} + c_{34}\right)\dot{y}_3 - c_{23}\dot{y}_2 - c_{34}\dot{y}_4 + \left(k_{23} + k_{34}\right)y_3 - k_{23}y_2 - k_{34}y_4 \\ = -m_3 g + m_3 e\omega^2 \sin\omega t \\ m_4\ddot{x}_4 + \left(c_{34} + c_{45}\right)\dot{x}_4 - c_{34}\dot{x}_3 - c_{45}\dot{x}_5 + \left(k_{34} + k_{45}\right)x_4 - k_{34}x_3 - k_{45}x_5 \\ = -c_{n2}\dot{x}_4 - \sum_{i=1}^{N_2} k_{n2}\left[\left(x_4\cos\theta_i + y_4\sin\theta_i + C\right)\cos\alpha_2\right]^{1.5}\cos\theta_i \\ m_4\ddot{y}_4 + \left(c_{34} + c_{45}\right)\dot{y}_4 - c_{34}\dot{y}_3 - c_{45}\dot{y}_5 + \left(k_{34} + k_{45}\right)y_4 - k_{34}y_3 - k_{45}y_5 \\ = -m_4 g - c_{n2}\dot{y}_4 - \sum_{i=1}^{N_2} k_{n2}\left[\left(x_4\cos\theta_i + y_4\sin\theta_i + C\right)\cos\alpha_2\right]^{1.5}\sin\theta_i \\ m_5\ddot{x}_5 + c_{45}\left(\dot{x}_5 - \dot{x}_4\right) + k_{45}\left(x_5 - x_4\right) = m_5 e\omega^2 \cos\omega t \\ m_5\ddot{y}_5 + c_{45}\left(\dot{y}_5 - \dot{y}_4\right) + k_{45}\left(y_5 - y_4\right) = -m_5 g + m_5 e\omega^2 \sin\omega t \end{cases} \tag{5}$$

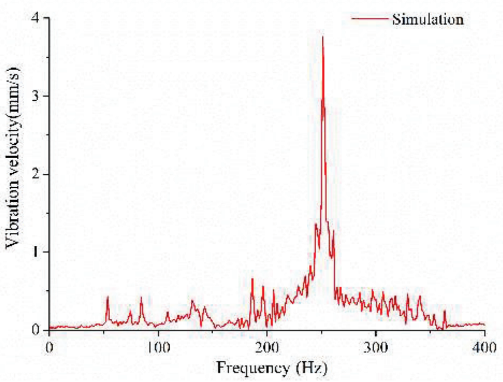
Where m_1, m_2, m_3, m_4 and m_5 are the centralized mass of the spindle system, k_{12}, k_{23}, k_{34} and k_{45} are the stiffness coefficients of the rotating shaft, c_{12}, c_{23}, c_{34} and c_{45} are the material damping coefficients of the rotating shaft.

3. Simulation Analysis of Vibration

The frequency domain variation of rotor vibration of ceramic motorized spindle system is directly related to intermittent fit. Through the finite element method, the vibration time-domain and frequency-domain diagrams of the rotor system are obtained, as shown in Figure 2. At 15000r/min, considering the influence of clearance fit, the vibration can reach 13mm/s. It can be seen from the frequency domain that gap fit is one of the main factors affecting vibration, affecting the frequency doubling of vibration.



(a) Time-domain



(b) Frequency-domain

Figure 2. The vibration and diagrams of the rotor system

4. Experimental Verification

Motorized spindle vibration test system includes motorized spindle performance test platform, Polytec OFV-505/5000 Xtra high performance single point laser vibrometer, MR8875-30 data storage recorder, tripod, etc. The laser vibrometer is based on the Doppler vibrometer method, which extracts the amplitude and frequency from the signal by generating a frequency shift from the laser reflection. The vibration test diagram of spindle is shown in Figure 3.

Figure 3. The vibration test diagram of spindle

According to Fig. 4, it can be seen that the time-domain difference of vibration error between experiment and simulation is 1.7mm/s. The frequency domain is not very different, it has good consistency with experiment. The results show that the rotor system dynamics model has good accuracy.

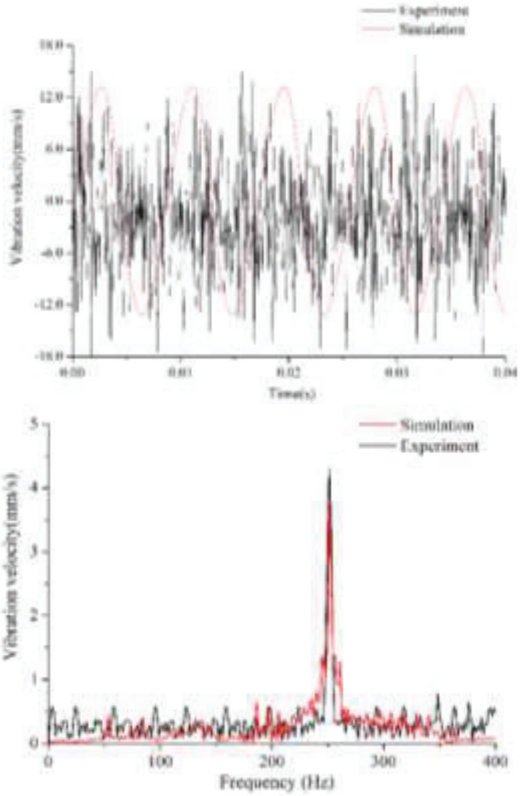


Figure 4. Comparison of simulation and experiments

5. Summary

In this paper, taking motorized spindle as the research object, a dynamic model of spindle rotor-bearing system considering clearance fit is established. The influences of the clearance fit of bearing outer ring on the vibration characteristics of rotor system are important. The results provided are useful for the optimal design of motorized spindle system.

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