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Research and Design of Flexible Shaft of a Squeeze Film Damper

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Abstract
In high speed rotating machinery, damper parameters setting directly affects dynamic stability of the flexible shaft. This paper established Capone nonlinear oil film force model to get the nonlinear response of sliding bearing oil film on flexible shaft. Modal and harmonic analysis were made by using the finite element method (FEM). The critical parameters of squeeze film damper through theoretical calculation were verified by establishing dynamic model of rotor system. The results show that the damper was designed in this paper can be used to achieve good damping effect, and provided a theoretical basis for the subsequent mechanical strength high speed experiment testing machine. At the same time, using this method to solve the squeeze film damper parameters is also a kind of beneficial exploration.

1. Introduction

In high speed rotating machinery, mechanical vibration seriously affects the reliability of the system and the whole structure, especially the general mechanical strength high speed test machine. It has to work on the first order critical speed of shaft. This machine needs to overcome vibration caused by the resonance frequency speeds (Cui Lingjun, et al, 2010).

Squeeze film damper (SFD) can effectively reduce the excessive vibration when flexible shaft is over the critical speed. However, due to the diversity and complexity of the flexible shaft, reasonably determining the squeeze film damper design parameters is not easy. Appropriate parameters can make the mechanical vibration reduced more than 60% (Feng Xinhai, et al, 1986).In the recent ten years, the design and research of squeeze film damper had made some progress. A new squeeze film damper with metal rubber ring was proposed (Zhang Ruihua, et al, 2010), it did not take the complex nonlinear effect into consideration, there was certain limitation. Some scholars researched parameters for solving the damper structure value, although using the mechanical model of elastic rotor bearing system (Guo Zenglin, et al, 1997), but the maximum amplitude value was assumed, which could not meet the actual needs of the project. Therefore, in this paper, we considered the nonlinear effect in sliding bearing of shaft, and calculated the actual value of the maximum amplitude of flexible shafting through the finite element analysis to inverse solution of various structure parameters of squeeze film damper value.

2. Methods

2.1. Model Geometry

The highest work speed of the universal test machine is 55000r/min, the flexible shaft is fine and short, that can reduce the shaft line speed to reduce friction heat at high speed, short axis of the amplitude can be reduced.

According to the above requirements and practice experience, flexible shaft adopts 40CrNi material, the density is 8000kg/m3, the elastic modulus is 195000MPa, Poisson's ratio is 0.3, the diameter size is10mm, length is 200mm. The 3D entity model of flexible shaft is established by using Solid Works 3D modeling software, see figure 1. The red parts of the figure are the installation areas of sliding bearing.

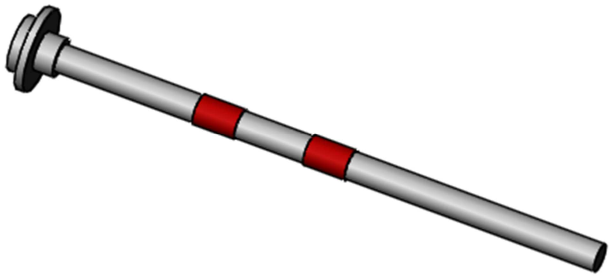

Figure 1. *The 3D solid model of flexible shaft.*

2.2. The Spindle Modal Analysis

Modal analysis determines the inherent frequency and vibration type of flexible shaft, the calculation of vibration frequency ω_i and modal ϑ_i is:

$$\left([K]-\omega_i^2[M]\right)\{\vartheta_i\}=0 \quad (1)$$

Where:[K] is the stiffness matrix; [M] is the mass matrix.This equation uses the modal module of ANSYS Workbench and the subspace iteration method. To simplify the analysis model, the oil film force amplitude is equivalent to spring stiffness for loading constraints, and boundary conditions are given in terms of the fixed constraints on two end faces offlexible shaft. At last,we can get four order natural frequency of the flexible shaft.

2.3. The Spindle Harmonic Response Analysis

Analysis can be used to determine the structure of the steady state in the sinusoidal excitation response of harmonic response, thus verifying if the structure can successfully overcome harmful effects caused by resonance and fatigue. The motion equation for the harmonic response analysis is:

$$(-\omega^2[M]+i\omega[C]+[K])(\{\vartheta_1\}+i\{\vartheta_2\})=(\{F_1\}+i\{F_2\}) \quad (2)$$

Where: [C] is the damping matrix; ϑ_1 is the real part of the displacement; ϑ_2 is the imaginary; F_1 is the real part of the excitation force; F_2 is the imaginary.

2.4. Calculation of Key Parameters of Squeeze Film Damper

According to the characteristics of flexible shaft and squeeze film damper, the radius value of the damper set in this paper is 36mm and the unilateral bearing width is 21mm, the gap can be determined by the following formula:

$$C=\sqrt[3]{\frac{\mu RL^3\varepsilon\omega}{K(1-\varepsilon^2)}} \quad (3)$$

Where: $\mu=0.196N\cdot S/m^2$ is the lubricating oil viscosity coefficient; R is damper radius; L is damper bearing width; ε is eccentricity; ω is the rotor synchronous frequency; K=160000N/m is damper equivalent support stiffness, it can be obtained from the upper section of the harmonic response.

To verify the reasonableness of damper parameters, the dimensionless bearing parameter B has important effect on the damping evaluation, it can be calculated by the formula:

$$B=\frac{\mu RL^3}{m\omega_c C^3} \quad (4)$$

Where: μ is lubricating oil dynamic viscosity coefficient; R is damper radius; L is damper bearing width; m is journal quality of vibration; ω_c is the first critical speed; C is damper gap.

3. Results

Based on the modal analysis, the first four order flexible shaft critical speed was obtained, natural frequency and the corresponding are shown in Table 1.

Table 1. *The natural frequency and critical speed of flexible shaft.*

Order number	Natural frequency/Hz	Critical speed/r/min
1	997.91	59874.6
2	998.04	59882.4
3	2625.3	157518
4	2625.6	157536

From the data in table 1, we knew that the first order critical speed of spindle was 59874.6r/min, which was much larger than the machine work speed. The first order vibration shape is shown in Figure 2. It shows that the red area is maximum amplitude, while the large amplitude will make the spindle produce plastic deformation or even destruction. Modal vibration significance lies not only in qualitative to see each order vibration shape of the spindle, but also in the response to provide theoretical basis for quantitative harmonic analysis.

The modal superposition method is employed in this paper of the harmonic response analysis of the spindle. The value of excitation force amplitude is 200N, damping coefficient is 0.03, the initial phase is 0, frequency response is 0 ~ 3800Hz and divided into 100 sub steps. The force in vibration mode is at the maximum point. The displacement frequency response curve is shown in Figure 3.

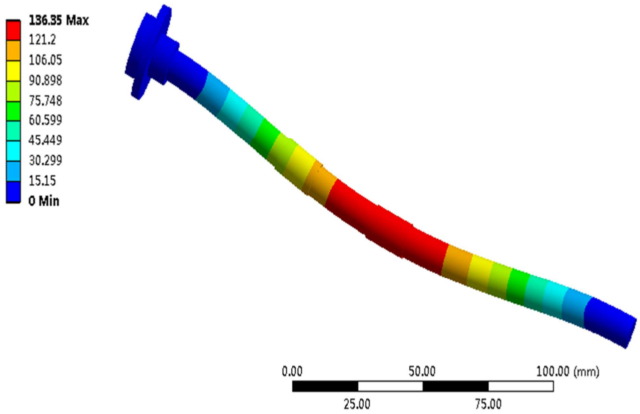

Figure 2. The first order vibration shape of spindle.

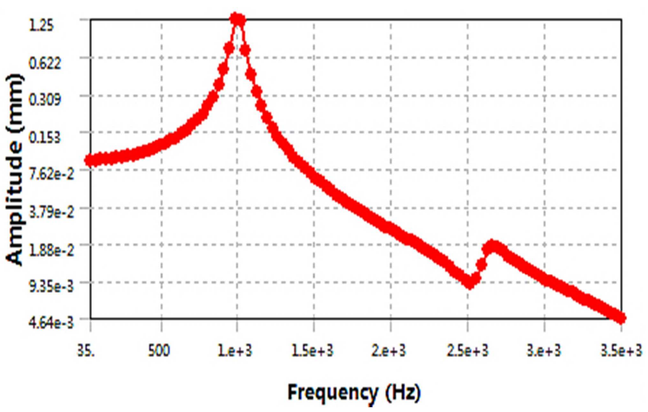

Figure 3. The position of maximum displacement amplitude frequency response curve.

From the figure, under the sinusoidal excitations, the maximum displacement of the spindle is 1.25mm, the value is less than the design requirement1.5mmin project, verifies the reliability of the flexible shaft. The result provides an important basis for the clearance of squeeze film damper and solution of dimensionless parameter of the bearing.

Substituting the obtained data, value B is 0.1689. When the value is beside 0.1, damper has a good damping effect (Feng Xinhai, et al, 1989).The damper parameters obtained above is reasonable and reliable, the key parameters of damper are shown in Table 2.

Table 2. The design results of parameters of squeeze film damper.

Project	Result
Damper radius R/mm	36
Damper width L/mm	21
Damper gap C/mm	0.54
The equivalent support stiffness K/N/m	160000
Eccentricity e/mm	0.05
Viscosity of lubricating oil $\mu / N\cdot S\cdot m^{-2}$	0.196
Lubricating oil density $\rho / Kg\cdot m^{-3}$	850

4. Discussion

In order to carry out the vibration analysis of squeeze film damper, the damper-bearing-shaft coordinate system was established. Making static equilibrium point as the reference coordinate system origin, the lumped mass method was adopted to solve the motion equation of the system. Coordinate system is shown in Figure 4.

According to the figure, three equations are:

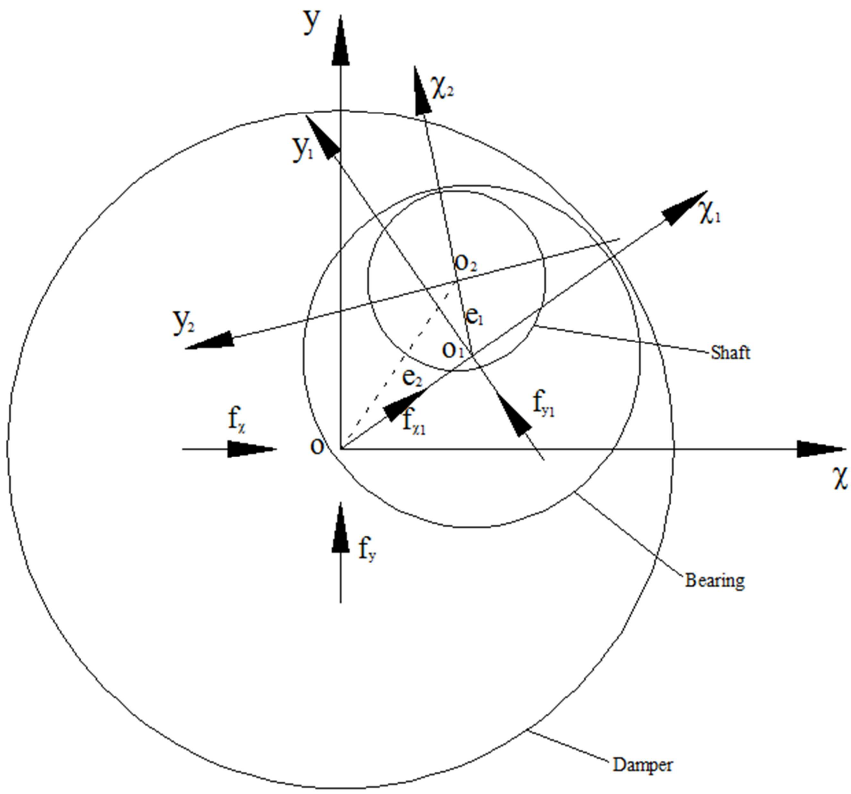

Figure 4. Damper- bearing –shaft coordinate system.

For damper:

$$\begin{cases} m_2\ddot{x}_2=-k_2\left(x_2-x_1\right)+m_2\rho\omega^2\cos(\omega t) \\ m_2\ddot{y}_2=-k_2\left(y_2-y_1\right)+m_2\rho\omega^2\sin(\omega t) \end{cases} \tag{5}$$

For bearing:

$$\begin{cases} m_1\ddot{x}_1=-k_1x_1+f_x\cos\alpha_1-f_y\sin\alpha_1+f_{x1}\sin\alpha_2-f_{y1}\cos\alpha_2 \\ m_1\ddot{y}_1=-k_1y_1+f_x\sin\alpha_1+f_y\cos\alpha_1-f_{x1}\cos\alpha_2-f_{y1}\sin\alpha_2 \end{cases} \tag{6}$$

For shaft:

$$\begin{cases} m\ddot{x}=-k_2\left(x_1-x_2\right)-f_{x1}\sin\alpha_2+f_{y1}\cos\alpha_2 \\ m\ddot{y}=-k_2\left(y_1-y_2\right)+f_{x1}\cos\alpha_2+f_{y1}\sin\alpha_2 \end{cases} \tag{7}$$

Where: "O" is the center of the damper body; "O1" is the center of bearing body; "O2" is shaft center; f_x, f_y is the oil film force component from damper to bearing; f_{x1}, f_{y1} is the oil film force component from bearing to shaft. m_2 is the damper quality; m_1 is the bearing body quality; m is the bearing quality; k_2 is the shaft neck stiffness; k_1 is the damper stiffness; α_1 is the bearing body coordinate system $o_1x_1y_1$ turning polar angle; α_2 is the shaft coordinate system $o_2x_2y_2$ turning polar angle; ρ is the mass eccentricity; ω is the rotor rotating angular velocity.

Setting e_1 as the eccentricity between journal center and bearing center and e_2 as the eccentricity between bearing center and the damper center, they have the following geometric relationship:

$$e_1=\sqrt{(x_1-x_2)^2+(y_1-y_2)^2}$$

$$e_2=\sqrt{x_1^2+y_1^2}$$

$$\alpha_1=\arcsin\frac{x_1}{e_2}$$

$$\alpha_2=\arcsin\frac{x_2-x_1}{e_1} \tag{8}$$

For bearing, oil film force component was obtained by using the Capone oil film force model of the damper (Huang Wenhu, et al, 2006; G. Adiletta, et al, 1996; Capone,1991).For the oil film force of damper, using end bearing theory, oil film force component expressions are:

$$f_x=\frac{\mu RL^3}{C^2}\left[\frac{2\omega\varepsilon}{(1-\varepsilon^2)^2}\varepsilon+\frac{\pi}{2}\frac{(1+2\varepsilon^2)}{(1-\varepsilon^2)^{\frac{5}{2}}}\varepsilon\right]$$

$$f_y=\frac{\mu RL^3}{C^2}\left[\frac{\pi\omega\varepsilon}{2(1-\varepsilon^2)^{\frac{3}{2}}}+\frac{2\varepsilon}{(1-\varepsilon^2)^2}\varepsilon\right] \tag{9}$$

This paper uses 4 to 5 order Runge-Kutta method for solving the equations of motion of the system. When the rotor speed is 24000r/min, comparing the system with damper and without damper under the condition of steady state response, the result is shown in Figure 5. As can be seen from the graph, after adding damper, vibration amplitude of the rotor system has obvious attenuation, further demonstrates the damper parameters designed in this paper is reasonable and reliable.

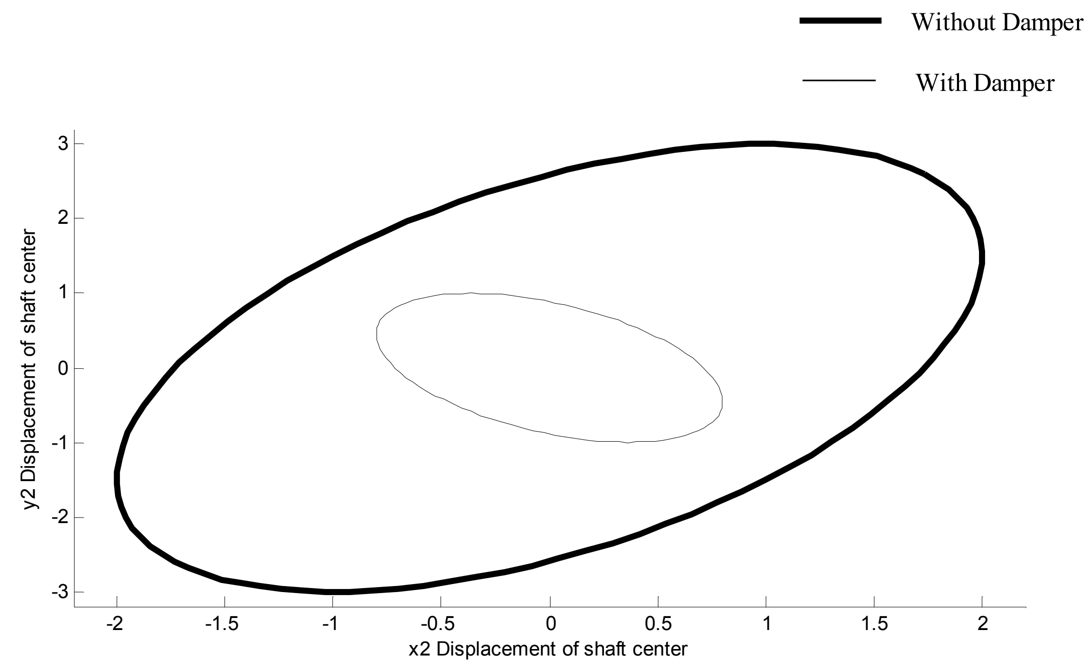

Figure 5. *24000r/min axial steady orbit response curve.*

5. Conclusion

By using the Capone nonlinear oil film force model, the modal and harmonic response analysis of flexible shaft was done and vibration response curve was obtained. On the basis, the key parameters of the damper was calculated, the results showed that, the parameters were reasonable and reliable.

The sliding bearing-flexible rotor-squeeze film damper dynamic model was established, results showed that the damper was designed by this method can play an ideal damping effect.

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