Research on Vibration Reduction of a Squeeze Film Damper

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Abstract. With the trending to high speed, high precision and heavy load of large-scale rotating machinery, the problem of dynamic instability of rotor-bearing systems has become increasingly serious. In this paper, a dynamic model of squeezed-film damper-journal bearing-rotor system was established and the influence of different stiffness and damping on rotor response was analyzed. Three kinds of squeeze film dampers with different stiffness and damping were designed to be tested under various conditions. The signals of acceleration amplitude of bearing seat and rotor displacement were acquired, processed and analyzed. The experimental results show that there is a significant decrease in vibration response and acceleration amplitude of the bearings embedded with SFD compared to journal bearings without SFD. In addition, the greater the damping value of SFD, the smaller response of rotor. Stiffness has less effect on vibration reduction compared to damping which is basically consistent with the theoretical analysis.

1. Introduction

Squeeze film damper (SFD) has a simple structure, small footprint, easy manufacture and obvious damping effect. It is first applied in aero-engines and has become a typical design to reduce engine vibration and extend life. Domestic and foreign scholars have done a lot of research on eliminating the non-coordinated response of the rotor and improving the stability of the bearing rotor system.

Cooper et al [1] published a report on the performance study of SFDs, pointing out that SFDs could reduce engine amplitude and created a new era of SFD applications. In order to solve synchronous vibration of the unit, Leader et al [2] used an O-ring SFD to reduce synchronous vibration amplitude by 30%, but the rubber was prone to creep under the condition of large load. KanKi et al [3] designed a pin centering SFD to solve sub-synchronous whirl. At the same time, Boyaci et al [4] systematically studied the effects of stiffness, clearance and inner ring mass of the centering spring of floating ring SFD on rotor vibration. In addition, Zhu et al [5] studied nonlinear behavior of uncentered spring damper, and pointed out that the motion mode of uncentered spring SFD was more complicated and variable than the centering spring SFD. Andrés et al [6] built an SFD test device and carried out theoretical calculations and experimental tests on dynamic characteristics of SFD.

Domestic research on SFDs began in the 1970s. Shen and Liu et al [7-8] conducted theoretical and experimental research on SFDs. Hong et al [9] studied the properties of oil film pressure field distribution and stiffness damping by using finite element theory with elastic ring SFD. SFDs exhibit richer nonlinear dynamics in systems with multiple degrees of freedom and complex structures. Zhu et
al [10] studied the bistable characteristics of a flexible rotor system with a centering spring SFD. It was found that when rotor unbalance was less than a critical value, bistable state would occur, and the rotor orbit would be abruptly changed. Qin et al [11] studied the SFD’s nonlinear response under elastic support and the differential equation of motion of the single disc rotor system. Recently, Liao et al [12] found that SFD could effectively reduce vibration of turbine disk in rubbing fault. Also, rotor vibration could be reduced for the misalignment fault, and the suppression of amplitude of the double frequency vibration was particularly obvious.

Most of the above studies are related to the rolling bearing SFD, and there are few researches on journal bearing SFD. Ertas et al [13] applied SFD to multi-span turbine tilting pad bearings, which not only solved the problem of large sub-synchronous eddy vibration, but also ensured the stability of the unit under full load and high speed. In this paper, through theoretical analysis and experimental study, dynamic characteristics of SFD under the support of the journal bearing are studied. The purpose is to find a reasonable design for the structural parameters of the journal bearing with SFD to effectively reduce amplitude of the rotor and improve its stability for reducing vibration transmitted by the rotor to the frame.

2. Dynamic Characteristics Analysis and Structure Design of Rotor System

As an elastic damping element, the stiffness and damping of SFD in the rotor system are nonlinear functions. If the design is not proper, damping cannot exert the effect of vibration reduction. Therefore, the parameter design of SFD is a key factor in determining its performance.

2.1. Damping calculation

The fluid lubrication equation for the SFD is:

\[ \frac{1}{r^2 \frac{\partial}{\partial \varphi}} \left( \frac{r^2}{12 \mu} \frac{\partial^2 P}{\partial \varphi^2} \right) + \frac{\partial}{\partial Z} \left( \frac{r^2}{12 \mu} \frac{\partial h}{\partial Z} \right) = - \frac{\Omega^2 (\rho c) h}{2} + \rho \left( V_x \cos \varphi - V_y \sin \varphi \right) - h \delta \]

(1)

The inner and outer rings of the squeeze film are not rotated at 0, so \( \Omega = 0 \) and \( V_x = 0 \). Assuming liquid is incompressible, the equation can be simplified:

\[ \frac{1}{r^2 \frac{\partial}{\partial \varphi}} \left( \frac{r^2}{\mu} \frac{\partial P}{\partial \varphi} \right) + \frac{\partial}{\partial Z} \left( \frac{r^2}{\mu} \frac{\partial h}{\partial Z} \right) = 12 \frac{dh}{dt} \]

(2)

The pressure at both ends of SFD is the same as the external pressure, and the pressure in the middle of the SFD is high during operation, therefore, the pressure gradient in the circumferential direction does not change much. In this case, ignoring the first item on the left side of the above-mentioned formula, one-dimensional short bearing Reynolds equation is as follows:

\[ \frac{\partial}{\partial Z} \left( \frac{h^3}{\mu \frac{\partial h}{\partial Z}} \right) = 12 \frac{dh}{dt} \]

(3)

Under full Sommerfeld conditions, the damper damping calculation formula is:

\[ C = \frac{4 \pi \mu l^3}{9 \varepsilon} \times \frac{\pi}{\left( 1 + \varepsilon^2 l \right)^2} \]

(4)

In the above-mentioned formula, \( \mu \) is the dynamic viscosity of the lubricating oil, \( r, c \) and \( l \) are the damper radius, gap and width respectively, and \( \varepsilon \) is the eccentricity.
2.2. Stiffness calculation
The stiffness of SFD centering spring is a key step in SFD design. Generally, the trial and error method are used to determine the centering spring stiffness.

2.3. Response analysis
Assuming that rotor performs steady-state coordinated motion and ignores the rotor acceleration and the disc gyro effect, the simplified rotor dynamics equations are:

\[ M_2 \ddot{X}_2 + C_1 \dot{X}_2 + C_{11} \dot{X}_2 - iQX_2 - (K_1 - ioC_1)X_2 = M_2 e^{i\alpha t} \]  
\[ C_2 \dot{X}_2 + C_{21} \dot{X}_2 + K_2 X_2 - (K_2 - ioC_2)X_2 = 0 \]  
\[ M_1 \ddot{X}_1 + C_1 \dot{X}_1 + C_{11} \dot{X}_1 - (K_1 - ioC_1)X_1 = 0 \]  

In the above-mentioned equations, \( X_2 \) is the relative displacement of the shaft; \( Q \) is the cross stiffness; \( M_2 \) is the inner ring mass; \( M_1 \) is the rotor mass; \( K_2 \) is the bearing stiffness; \( C_2 \) is the bearing damping; \( K_1 \) is the shaft stiffness; \( C_1 \) is the shaft damping; \( K_3 \) is the SFD stiffness; \( C_3 \) is SFD damping.

The rotor imbalance response is in the Y direction, which is the vertical direction value. The rotor amplitude caused by the eccentric mass is as follows, where \( m \) is the unbalanced mass, \( \varepsilon \) is the eccentricity, \( K \) and \( C \) are the stiffness damping coefficients respectively.

\[ A = \frac{\max^2}{\sqrt{(K-m\varepsilon^2)^2 + (C\varepsilon)^2}} \]  

In this test, three kinds of SFDs with different stiffness and damping are designed according to the journal bearing parameters as shown in Table 1. The structure is shown in Figure 1. The specific parameters are shown in Table 2.

### Table 1. The parameters of journal bearing

| Rotor diameter (D/mm) | Bearing length (B/mm) | Gap ratio | Rotor length (mm) |
|----------------------|-----------------------|-----------|-------------------|
| 19.05                | 12                    | 3‰       | 838               |

### Table 2. The parameters of testing SFD

| Bearing number | Radial clearance (mm) | Stiffness (N/m) | Damping (N(m/s)) |
|----------------|-----------------------|-----------------|------------------|
| Case 1(Journal bearing) | 0.057                | 7.9e5           | 4.06e3           |
| Case 2          | 0.018                | 2.4e3           | 7.3e4            |
| Case 3          | 0.045                | 2.4e3           | 7.3e3            |
| Case 4          | 0.045                | 2.4e5           | 7.3e3            |

![Figure 1. The structural design of SFD](image)
According to the parameters of test rig, the vibration response of the bearings embedded with different SFD and journal bearing without SFD is calculated. As it shown in Figure 2, the vibration response of the bearings embedded with SFD is smaller than that of journal bearing without SFD, and the greater the damping value of SFD, the smaller response of rotor. Stiffness has less effect on vibration response compared to damping, and the stiffness change does not cause a large change in the response.

![Different stiffness](image1)

![Different damping](image2)

**Figure 2.** The comparison of rotor amplitude on different SFD

3. Test Rig

Figure 3 shows the overall layout of a flexible rotor test rig for vibration and stability testing of sliding bearings.

![Test Rig](image3)

**Figure 3.** Overview of the test rig
The driving motor is coupled to the rotor by the flexible coupling with clamp style fixing, and a pair of test journal bearings for the rotor support are horizontally placed at both ends of the rotor. When the rotor starts to rotate, hydrodynamic oil film is formed between the test bearing and the rotational journal. The rotor rotational speed is measured by an optical tachometer probe producing a string of impulses. The reflecting target is attached to the motor output shaft. The main parameters of the test rig are shown in Table 3.

| Table 3. Experimental conditions |
|---------------------------------|
| Spindle speed range (rpm)       | 0~6000 |
| Type of lubricant               | ISO-VG32 |
| Lubricating oil dynamic viscosity| 0.039 |
| Oil inlet temperature (℃)       | 20 |
| Oil supply pressure (MPa)       | 0.1 |

In the test, non-contact eddy current sensor and acceleration sensor are used, and the installation method is shown in Figure 3. Two TR810503 non-contact eddy current sensors (±5% sensitivity) are mounted in the same plane along the X and Y directions (45° from the horizontal) for measuring relative shaft displacement on rotor test rig. The accelerometer which model is PCB352C03 (sensitivity is ±10%) is placed in the horizontal and vertical directions of the bearing housing to measure the relative value of acceleration amplitude of bearing housing. Finally, the sensor is connected to Bruel&Kjaer PULSE signal sampling and processing system to process the acquired signal.

4. Results and Analysis
The vertical vibration directly transmits to the bearing housing, which directly affects the vibration environment of the bearing base. Therefore, the analysis is mainly based on the vertical direction signal of the free end of the rotor. In addition, the rotor system instability mainly occurs when the system's forced vibration frequency is proportional to the natural frequency. Through Matlab simulation analysis, it can be seen from the rotor model and Campbell diagram shown in Figure 4 that the critical speed of the rotor is about 3300r/min. This parameter provides a basis for the analysis of the dynamic characteristics of the subsequent journal bearing. In order to effectively analyze the vibration of the bearing, this paper mainly analyzes the change of the axial orbit and the acceleration value of the bearing base.

Figure 4. The model and Campbell chart of rotor
4.1. Analysis of the influence of stiffness and damping on the axis orbit

Figure 5 plots the axial orbits of different stiffness and damper bearings at four speeds. The green, red, yellow, and blue lines represent the different bearings of Case 1, Case 3, Case 4, and Case 2.

![Figure 5. Axis orbits at different rotational speeds](image)

It can be seen from the above four axial orbits diagrams that as the rotational speed increases, the axial orbit of the rotor gradually increases. At the critical speed (3300 rpm), the relative shaft displacement on rotor test rig reaches the maximum, after exceeding the critical speed, the axis orbit is gradually getting smaller.

By analyzing the influence of stiffness on the rotor response, it can be seen that the axial orbit of Case 3 is smallest, but this kind of SFD has little effect on vibration reduction. Before the critical speed, the axis displacement of this kind of SFD can be effectively reduced, but after the critical speed, this kind of SFD has less effect on vibration reduction than the SFD with greater stiffness value.

By analyzing the influence of damping on the rotor response, it can be seen that the greater the damping value of SFD, the smaller respond of rotor, especially for the critical speed region. It shows that increasing the damping of the SFD can effectively reduce the response of rotor, and improve the stability of the bearing-rotor system. Therefore, the damping value of the SFD is a key parameter of the design.
4.2. Analysis of the influence of stiffness and damping on acceleration
As can be seen from Figure 6 is that the relationship between the vertical acceleration amplitude of bearing embedded with different SFD and the rotor speed. The rotor speed is in the range of 1200-6000 r/min. and magnitude of acceleration is expressed in decibels (dB), and its calculation formula is:

\[ \text{VAL} = 20 \log \frac{a}{a_0} (\text{dB}) \]  (9)

In the above-mentioned formula, \( a \) is value of vibration acceleration, the unit is \( \text{m/s}^2 \); \( a_0 \) refers to the reference acceleration, \( a_0 = 10^{-6} \text{m/s}^2 \).

![Figure 6. Vertical acceleration amplitude of bearing housing at the flexible end](image)

(a) Acceleration values at different stiffnesses (b) Acceleration values under different damping

It can be seen from Figure 6 that at the critical speed, the vertical acceleration amplitude reaches the maximum, and near critical speed, the acceleration amplitude rises and fall rapidly. There is a significant decrease in acceleration amplitude of bearings embedded with SFD Compared to journal bearings without SFD, especially for the critical speed region.

Comparing the SFDs with different stiffness and damping, it can be seen that the smaller the stiffness value of SFD, the smaller response of rotor, and the greater the damping value of SFD, the smaller response of rotor. Stiffness has less effect on vibration reduction compared to damping after the critical speed, the SFD with smaller stiffness has less effect on vibration reduction than the SFD with greater stiffness. Therefore, the reasonable stiffness and damping value design of the SFD can effectively reduce the vibration of rotor.

5. Conclusion
In this paper, the dynamic equations of the bearing-rotor system are established, the critical speed of the bearing-rotor system are calculated, the rotor unbalance response of bearings embedded with different SFDs and journal bearings without SFD are compared. The results show that increasing the damping of the SFD can effectively reduce the response of rotor for the calculated double-disc rotor model. In addition, a flexible rotor test rig was built and three kinds of SFDs with different stiffness and damping were designed to be tested under different conditions. The signals of rotor response were acquired, processed and analyzed. The experimental results were consistent with the theoretical analysis, which verified the accuracy of the theoretical calculation.

Different stiffness and damping parameters have different effects on vibration reduction. In this paper, the greater the damping value of SFD, the smaller response of rotor. Stiffness has less effect on vibration reduction compared to damping. Therefore, reasonable design of stiffness and damping values is significant for vibration reduction of rotor-bearing system.
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