Study on Vibration Damping Characteristics of Controllable Squeeze Film Damper

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Abstract: How to control vibration is very important in the field of high speed rotors. Squeeze film dampers are considered an effective method. The existing controllable squeeze film damper is less used for vibration control of the rotor system supported by the sliding bearing. In order to suppress the vibration of the rotor system supported by sliding bearing, a hydraulically-controlled squeeze film damper (HSFD) is studied and analyzed in this paper. A flexible rotor test rig is set up and shaft center orbits and Fourier spectrums of the rotor are measured in order to study vibration damping effect of the sliding bearing with HSFD under different oil supply pressure. A hydraulic servo system is established in this paper to explore the damping effect of controllable squeeze film damper under the control system. The experimental results show that the sliding bearing with HSFD has significant damping effect and can reduce system nonlinearity.

1. Introduction
The impact of vibration is very important for high speed rotor system, especially for aero-engine, industrial steam turbine, centrifugal compressor and other key equipment in modern industries. The equipment is more likely to have larger vibration due to its high speed, rotor eccentricity and other reasons. Malfunction or serious damage accident may occur caused by excessive vibration if the equipment is not controlled [1-3].

Using of squeeze film damper (SFD) becomes the first choice for most people in order to suppress the vibration of the rotating machine rotor supported by the sliding bearing. Many researchers have done a lot of research on it. J. Agnew and D. Childs [4] measured the dynamic characteristics of tilting pad (TP) bearings with integral squeeze film damper. B. Ertas and et al. added ISFD to a 46MW 6230Kg multi-stage and multi-purpose turbine to solve the problem of sub-synchronous vibration at full load and full speed operation of the machine[5]. Researchers also provided a predictable force coefficient for the test frequency range [6]. Cai-Wan et al. had systematically analyzed dynamic behavior of the PSFD gear bearing system and found that PSFD can improve dynamic stability of gear bearing system [7]. A. Bouzidane and M. Thomas studied nonlinear dynamic behavior of a rigid rotor supported by a static pressure squeeze film damper[8-10].

Most of traditional SFD and structure improved damper adopt passive control methods, which is now more applicable to vibration damping. In the area of active control, these researchers have done some work. Cai-Wan et al. added electro-hydraulic actuator to a static SFD compensation and found that it was possible to reduce unsynchronized vibration of ordinary squeeze film damper [11]. Based on ordinary squeeze film damper, Ajay et al. studied a system of actively controlled lubrication with magneto rheological fluids and found that it is possible to reduce amplitude by as much as 70% near
the critical speed [12]. R.P. Spada and R. Nicoletti applied the Udwadia-Kalaba method to trajectory control of nonlinear systems, reducing the lateral vibration of the rotor at the control point when passing the first critical speed successfully [13].

Passive control methods have been widely adopted in rotor system damping field. While active control is mostly applied to the vibration damping of rolling bearing rotor system, and there are few studies on the application of sliding bearing rotor system.

In this paper, a hydraulically-controlled squeeze film damper (HSFD) is considered to be used for the vibration reduction of sliding bearing rotor system. The stiffness and damping characteristics of the squeeze film damper are changed by adjusting the oil supply pressure so as to control the vibration of bearing rotor system [14]. Vibration damping effect of the sliding bearing with HSFD is discussed in this paper through theoretical and experimental studies.

2. Equipment introduction

2.1 Test bench introduction

A flexible rotor test rig is set up as shown in Figure 1. The test rig is mounted on a T-slot cast iron base with a single-disk symmetrical rotor and two identical test bearings placed symmetrically on both ends. An oil supply system is available to provide a steady flow of lubricating oil for the testing system while an oil sink mounted on the T-slot cast iron base can reserve the lubricants. Shaft rotating speed is adjustable range 0-10000 rpm. The parts marked in figure 1 are: 1 speed motor; 2 bearing base; 3 eddy current sensor; 4 shaft; 5 disk; 6 acceleration sensor; 7 PC; 8 measurement and control system.

The test bearing used in experiment is shown in figure 2. The test bearing is composed of an ordinary sliding bearing and the HSFD. The HSFD studied in this paper is a four oil pad type static pressure squeeze film damper. A restrictor valve is connected to each oil chamber of the damper, and oil is supplied to the damper through the restrictor valve.

A restrictor valve is connected to each oil chamber of the damper, and oil is supplied to the damper through the restrictor valve. Parameters of test rig and test bearing are shown in Table 1.

| Parameter                  | Value   |
|----------------------------|---------|
| Disk mass (Kg)             | 0.576   |
| Shaft mass (Kg)            | 0.336   |
| Shaft stiffness (N/m)      | 1.2e4   |
| Bearing radius (mm)        | 12.5    |
| Oil inlet diameter(mm)     | 1       |
| Bearing mass (Kg)          | 0.16    |
| Lubricant viscosity (N*s/m^2) | 0.039  |
| Bearing width (mm)         | 14      |
| Bearing clearance (mm)     | 0.12    |

Figure 1 Flexible rotor test bench schematic
The eddy current sensors (TR81 model: 810503; sensitivity: (±5%) 5 v/mm) are installed perpendicular to each other to measure displacements of the shaft center. The accelerometers (PCB Model: 352C03; Sensitivity: (±10%) 10 mV/g) were fixed vertically on the bearing base for testing vibration level. The vibration data of the bearings will be eventually obtained by signal sampling and processing system from Bruel & Kjaer PULSE.

2.2 Model analysis

Accurately calculating critical speed of the rotor is important for determining the oil supply pressure and analyzing vibration damping effect of the rotor system. Using transfer matrix method, the critical speed of the rotor shaft is studied to obtain better conceptions about the dynamic behavior of the system.[15] The particular parameters of the rotor are recorded in Table 2. Figure 4 shows the first mode of the rotor mode shape, the natural frequency of the rotor is about 50.5 Hz, the 1st critical speed is 3031 rpm.

3. Experimental Results and Discussion

The purpose of this experiment is to study the effect of different oil supply pressure on vibration damping effect of the rotor system. At the same time, nonlinear problem of damping bearing under controllable oil pressure is experimentally contrasted.

3.1 Rotor amplitude under different oil pressure

In order to further investigate the damping effect and stability of the sliding bearings with HSFD, shaft center orbits and Fourier spectrums under different oil pressures are drawn and compared in Figure 5-8. The instability of sliding bearings is proportional to the size of area enclosed by the shaft center orbit [16].

As can be seen from Figure 5, the axial trajectory appears as an irregular shape at low speeds (1800 rpm), and as the oil pressure improves, the axial locus area gradually increases. In Figure 7, the area of the orbital axis locus of the rotor at the critical speed decreases as the oil supply pressure increases, which is opposite to that of Figure 5 (low speed). In the vicinity of the critical speed, the test bearing has more significant vibration damping effect and stability under high oil supply pressure. The axis orbit of Figure 8 (c) differs significantly from Figure 8 (a-b) in that it has an outer ring and the swirl is subsynchronous.

At low speeds, high oil pressures show poor results. The main cause of this phenomenon is that the hydraulic pulsation of the high pressure oil has an influence on the measurement. When the speed is
less than the critical speed and reaches the critical speed, the high oil pressure is more effective for rotor amplitude suppression and rotor stability. This is because the critical speed of the rotor increases as the stiffness of the support composite increases, causing the resonance peak to deviate from the operating speed. At high speeds, high pressure oil is more prone to half speed rotation than low pressure oil. It is well known that excessive bearing clearance can exacerbate oil whirl. The effect of nonlinear oil film force is greater than the unbalanced force. When the eccentricity is small, the rotor is susceptible to oil whirl. Low oil pressure contributes to rotor stability because lower support stiffness introduces greater eccentricity.

Figure 5 shaft center orbits at 1800rpm (a) blue line-0.1MPa, (b) red line-0.5MPa, (c) purple line-1MPa

Figure 6 shaft center orbits at 2400rpm (a) blue line-0.1MPa, (b) red line-0.5MPa, (c) purple line-1MPa

Figure 7 shaft center orbits at 3000rpm (a) blue line-0.1MPa, (b) red line-0.5MPa, (c) purple line-1MPa

Figure 8 shaft center orbits at 3800rpm (a) blue line-0.1MPa, (b) red line-0.5MPa, (c) purple line-1MPa
3.2 Control system introduction

A hydraulic servo system is established in this paper to explore the damping effect of controllable squeeze film damper under the control system. The change in flow rate and pressure of oil output from hydraulic cylinder can be caused by the voltage change of electro-hydraulic servo valve. The rotor journal amplitude signal measured by sensor is processed and fed back to controller. Controller adjusts servo valve voltage according to the signal deviation of measured value and set value to change oil pressure.

3.3 Comparative analysis of response

During analysis of the rotor response, respectively using ordinary sliding bearings and sliding bearings with HSFD, by setting the motor speed control device to make the rotor speed up from 0-4200rpm, the acceleration time is about 42s. Figure 10-11 shows the Y-direction rotor response diagram and HHT spectrum of the remote motor.

Comparing Figure 9 (a) with Figure 9 (b), it can be seen that the rotor response of the sliding bearings with HSFD is 57% of that of ordinary sliding bearings. Compared with ordinary sliding bearing, the sliding bearings with HSFD can significantly suppress the rotor amplitude, making the rotor more stable when passing the critical speed. As shown in Figure 10 (a), in addition to the fundamental frequency there is a significant components of 2X frequency, while there is a large amount of energy spread. In Figure 10 (b), the components of energy diffusion are less, which shows that the sliding bearings with HSFD has the effect of improving the nonlinearity compared with the ordinary sliding bearing.

The maximum displacement of the rotor response is about 2.5 times the bearing clearance, but this value is not consistent with the actual displacement at the bearing axis. The measurement is done at a distance from the axis, which introduces extra displacement due to misalignment and high elastic deformation experienced by the bearing during rotation of the bearing as the rotor obtains relative speed [17].

![Figure 9](image1.png)

(a) Figure 9 the response of the journal during the run up of the system (a) ordinary sliding bearing (b) sliding bearings with HSFD

![Figure 10](image2.png)

(b) Figure 10 HHT for the time histories of the response of the journal during the run up of the system (a) ordinary sliding bearing (b) sliding bearings with HSFD
4. Conclusion

This paper mainly studies damping characteristics and stability of the HSFD under different oil supply pressures according to the shaft center orbits, Fourier spectrums and acceleration amplitude of bearing housing. In addition, a speed run up response analysis was performed on the rotor system under control. Some conclusions can be obtained as follows:

1) The effect of changes in oil pressure on amplitude of the rotor is significant. High oil pressure is more effective for rotor amplitude suppression and rotor stability when speed less than the critical speed. However, the low oil pressure performance is better after 1st critical speed. At low speed, hydraulic pulsations have a significant impact and an accumulator should be added to reduce the interference.

2) It is found that low oil pressure can inhibit the half-speed vortex oil film at high speed.

3) The sliding bearings with HSFD have better vibration damping effect compared with ordinary sliding bearings. In addition, the sliding bearings with HSFD have better results in suppressing nonlinearities from the HHT chart.

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