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Utilizing a radial magnetic bearing to stabilize self-excited vibrations in a rotor-oil film bearing system

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Abstract

Oil film bearings have higher load capacity and larger damping than ball bearings. Therefore, the rotors of large turbomachines are usually supported by oil film bearings. However, at high rotational speeds, the swirling flow of the lubricant destabilizes the rotor system and induces self-excited vibrations such as oil whip and oil whirl. Since these vibrations limit the operational speed and reliability, a method for eliminating them and increasing the operational range is necessary. We propose a method for stabilizing these self-excited vibrations by using a radial magnetic bearing (RMB) to cancel the force due to the cross-coupled stiffness described in the Bently-Muszynska model. The purpose of this study is to experimentally examine the relationship between the gain of the controller and the stability.

Keywords : Vibration of rotating body, Oil film bearing, Self-excited vibration, Oil whip, Magnetic bearing

1. Introduction

Oil film bearings have higher load capacity and larger damping than ball bearings. Therefore, the rotors of large turbomachines such as centrifugal pumps and gas compressors are usually supported by oil film bearings. Recently, a need has arisen to improve the safety and reliability of turbomachinery systems for oil and gas applications, which operate at higher rotational speeds. However, at high rotational speeds, the swirling flow of the lubricant destabilizes the rotor system and induces self-excited vibrations such as oil whip and oil whirl. Since these self-excited vibrations limit the operational speed and decrease reliability, a method for eliminating these vibrations is required.

Although active magnetic bearings (AMBs) have small load capacity, their controllability is superior to that of oil film bearings. Thus, a rotor system supported by both oil film bearings and AMBs has both large load capacity and high controllability. This combination has the potential to enhance the operating speed and the stability. There has been some previous research on controlling self-excitation by utilizing AMBs. However, most of this research has not considered the dynamic characteristics of the fluid force generated by the cross-coupled stiffness (Kasarda et al., 2005, El-Shafei et al., 2010, Sahinkaya et al., 1985).

In order to eliminate self-excited vibrations in a rotor-oil film bearing system by utilizing an AMB, we proposed a control method that cancels the cross-coupled stiffness force described in the Bently-Muszynska model (Matsushita et al., 1988, Tsunoda et al., 2015). However, the relationship between the gain of the controller and the system instability was not examined. The purpose of this study is to examine the stability of a rotor-oil film bearing system assisted by AMB. In this paper, the relationship between the gain of the controller and stability is experimentally investigated.

2. Stabilization method for oil film bearing
2.1 Fluid force model

Figure 1 shows a cross sectional view of an oil film bearing. The oil film force is assumed to be linear with respect to the radial displacement around the static equilibrium. The rotor rotation generates the oil flow. It makes the rotor system unstable at high rotational speeds. According to the Bently-Muszynska model, the lubricant oil flows at an average angular velocity $\lambda\Omega$, where λ and Ω are the fluid circumferential average velocity ratio and the rotational speed of the rotor, respectively. The bearing force Q_{brg} of the oil film bearing in the radial direction is expressed as follows:

$$Q_{\mathrm{brg}} = -c_{\mathrm{d}}(\dot{z}_{\mathrm{b}} - j\lambda\Omega z_{\mathrm{b}}) - k_{\mathrm{d}} z_{\mathrm{b}} \tag{1}$$

where $z_{\mathrm{b}} = x_{\mathrm{b}} + jy_{\mathrm{b}}$ is the displacement of the rotor at the oil film bearing, and c_{d} and k_{d} are the damping and stiffness coefficients, respectively. The cross-coupled stiffness force caused by the oil flow acts as an unstable force. Furthermore, the cross-coupled stiffness force is proportion to the rotational speed, so if it exceeds the damping force at high rotational speeds, self-excitation is induced.

2.2 Stabilization method

In order to cancel the cross-coupled stiffness force $jc_{\mathrm{d}}\lambda\Omega z_{\mathrm{b}}$ and stabilize the rotor system, an AMB force in the same direction as the damping force is generated as shown in Fig. 2. The AMB force F_{AMB} without bias current is a nonlinear force and is described by Eq. (2).

$$F_{\mathrm{AMB}} = k_0 \frac{i_{\mathrm{z}}^2}{g^2} \tag{2}$$

where k_0 is a coefficient of the current-displacement-force-relationship, i_{z} is the sum of the coil currents $i_{\mathrm{x}} + ji_{\mathrm{y}}$ in the X and Y directions, and g is the gap between the electro-magnet and the rotor. In order to generate F_{AMB} in the same direction as the damping force and compensate the nonlinear characteristics of the magnetic attraction force, the coil current is controlled according to Eq. (3).

$$i_{\mathrm{z}} = i_{\mathrm{x}} + ji_{\mathrm{y}} = k_{\mathrm{p}}\left(-j\frac{x_{\mathrm{b}}}{\sqrt{|x_{\mathrm{b}}|}} + \frac{y_{\mathrm{b}}}{\sqrt{|y_{\mathrm{b}}|}}\right) \tag{3}$$

where k_{p} is the proportional gain of the AMB assist. By substituting Eq. (3) into Eq. (2) and neglecting the variation in the gap, we obtain the AMB force as follows;

$$F_{\mathrm{AMB}} = -jk_0 \frac{k_{\mathrm{p}}^2}{g^2} z_{\mathrm{b}} \approx -jk' z_{\mathrm{b}} \tag{4}$$

where k' is a constant coefficient, assuming g is constant. In the next section, the vibration of the rotor is experimentally evaluated by varying the control parameter k_{p}.

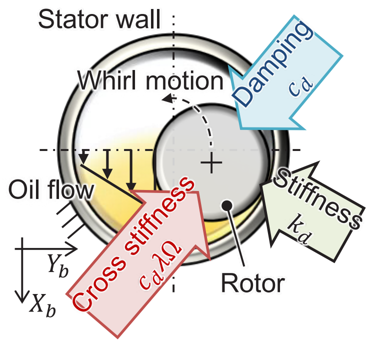


Fig. 1 Cross sectional view of oil film bearing and oil film force

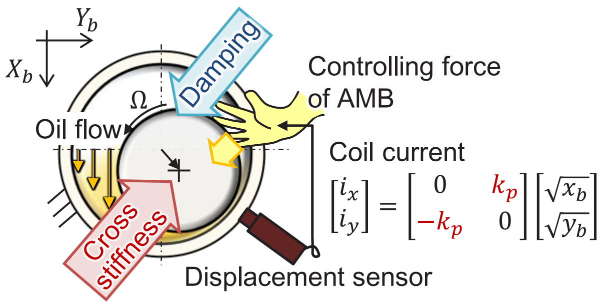


Fig. 2 Concept of the proposed stabilization method

3. Experiment for the stabilization of self-excited vibrations

3.1 Experimental setup

Table 1 and Fig. 3 show the configuration and dimensions of an experimental test rig. The rotor is supported by an oil film bearing and a ball bearing. A radial heteropolar-type magnetic bearing (RMB) is installed between a mass disk and the oil film bearing. Two eddy current displacement sensors are placed just beside the RMB to measure the radial motion of the shaft. The variation in the vibration of the shaft without RMB assist with respect to the rotational speed was measured as shown in Fig. 4. Self-excited vibrations occur at 73 rps. The 1st critical speed of the shaft is around 38 Hz, which is not shown in Fig. 4.

Table 1 Dimensions and specifications of the test rig

Rotor	Shaft	L430 × ϕ10[mm]
	Total mass	2.88 [kg]
	Natural freq.	38 [Hz]
Oil film bearing	Diameter	ϕ10 [mm]
	Length	10 [mm]
	Clearance	50 [μm]
RMB	Type	Hetero, 8 poles
	Inner diameter	ϕ29.5[mm]
	Length	30 [mm]
	Clearance	500 [μm]

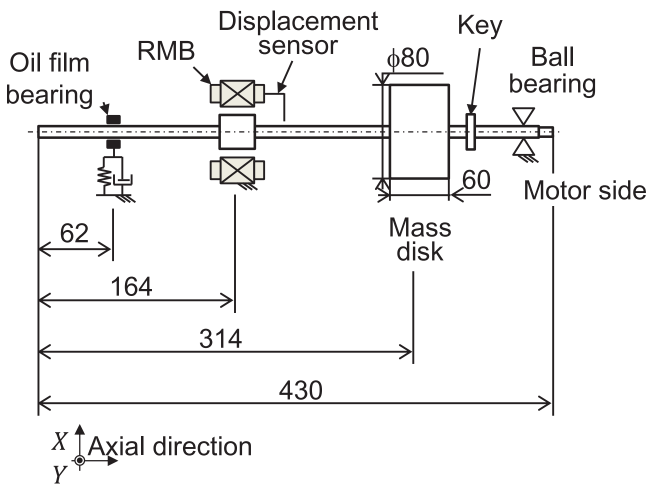


Fig. 3 Test rig

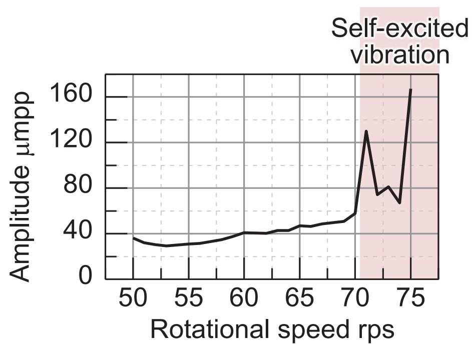


Fig. 4 Shaft amplitude without RMB assist

3.2 Control system and initial control parameter setting

Figure 5 shows a block diagram of the RMB assist. The x and y displacements are used to generate the y and x magnetic forces, respectively. Following Eq. (3), the square root of the absolute displacement multiplied by the gain k_p and the sign of the displacement gives the reference current. Based on the sign of each reference current, one of a pair of amplifiers is selected and switched in each direction. In this assist, non-bias control of the RMB is adopted.

Figure 6 shows the measured peak to peak amplitude of the shaft for various values of k_p from 0 to 180 A/m$^{0.5}$ at a rotational speed of 70 rps, which is just before the rotational speed at which the onset of self-excited vibrations occurs. The damping ratio is positive without RMB assist. Setting k_p to less than 160, the peak to peak amplitude is almost constant and less than 100 μm. In the high speed range above the 1st critical speed, the inertia force is dominant, so the RMB assist generating the stiffness force doesn't change the vibrational amplitude. However, when the value of k_p is larger than 180 A/m$^{0.5}$, the amplitude suddenly goes beyond 400 μm due to the instability of the rotor.

3.3 Suppression of the self-excited vibrations

Setting the gain k_p to 0, 140 and 160 A/m$^{0.5}$, the rotational speed of the rotor changes from 70 to 90 rps. The measured vibration is compared as shown in Fig. 7. Without RMB assist (k_p = 0 A/m$^{0.5}$) and with RMB assist with insufficient gain (k_p = 140 A/m$^{0.5}$), the rotational speed can't go beyond 76 rps, which is due to self-excited vibrations. On the other hand, with sufficient gain (k_p= 160 A/m$^{0.5}$), the vibrational amplitude is around 100 μm from 70 to 90 rps. Thus, the occurrence of self-excited vibrations can be completely suppressed.

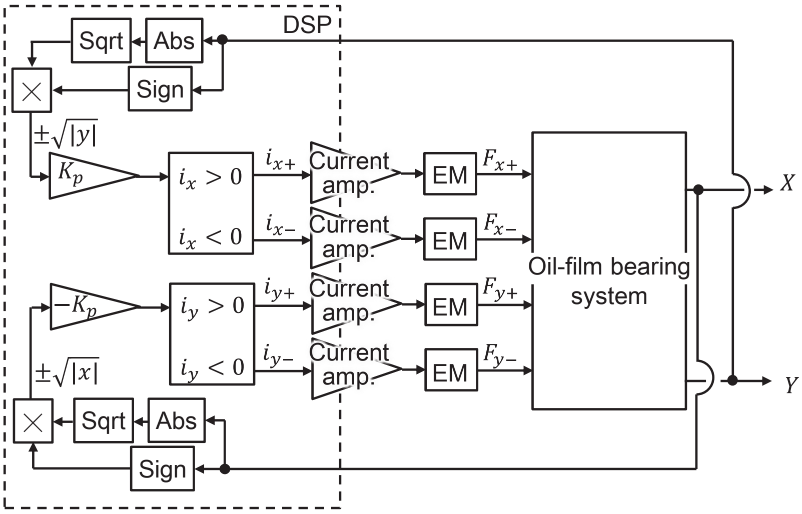


Fig. 5 Block diagram of RMB assist

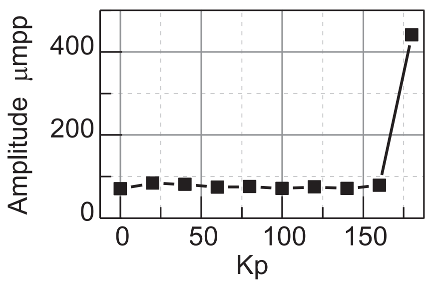


Fig. 6 Shaft amplitude with RMB assist at 70 rps

In order to check the effect of RMB assist, k_p was varied from 0 to 180 A/m$^{0.5}$ at a rotational speed of 75 rps, where self-excited vibrations occur without RMB assist. The amplitude slightly changes between 340 and 400 μm with k_p less than 75 A/m$^{0.5}$ as shown in Fig. 8. In this range of k_p, the rotor vibration becomes a limit cycle and the amplitude is almost constant. With k_p greater than 75 and less than 180 A/m$^{0.5}$, the amplitude is between 60 and 100 μm and the rotor system becomes stable. With the gain k_p greater than 180 A/m$^{0.5}$, the rotor becomes unstable because the RMB assist causes instability in the backward motion. The details are discussed in the next section.

3.4 Discussion

As can be seen in Fig. 8, large k_p causes instability. Theoretically speaking, an excessive value of k_p induces backward motion because the RMB generates a force in the backward direction. Figure 9 shows the FFT results for rotor vibration with k_p = 140, 160 and 180 A/m$^{0.5}$at a rotational speed of 70 rps. The backward component gradually increases as the gain k_p increases. By setting a high gain, such as k_p = 180 A/m$^{0.5}$, the backward force makes the rotor unstable. On the other hand, the forward component, which has the same frequency as the backward one, also increases. An elliptical orbit combining the backward and forward motion is observed. The orbit is generated due to the anisotropy of the stiffness induced by misalignment of the RMB.

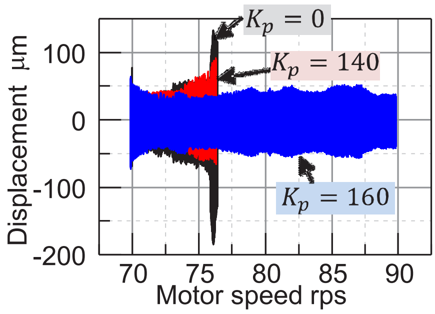


Fig. 7 Rotor displacement with and without the RMB assist

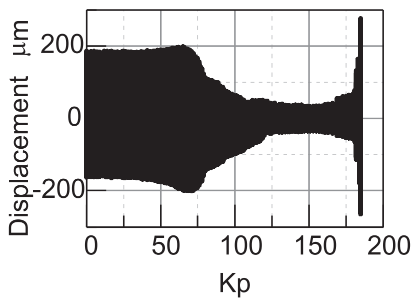
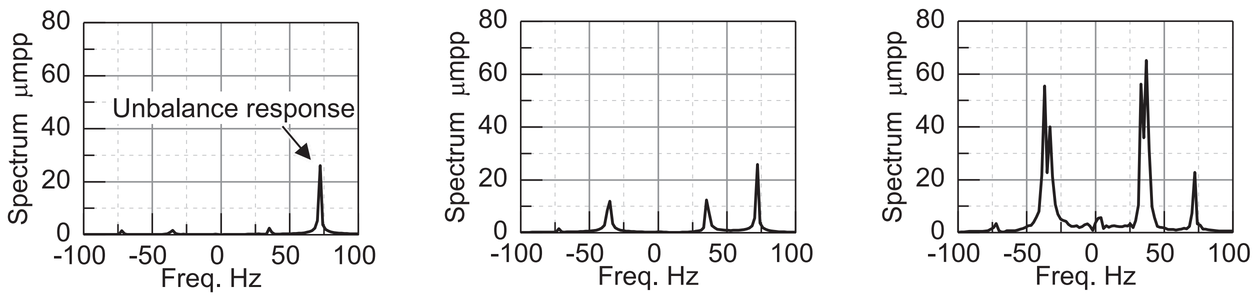


Fig. 8 Rotor displacement with RMB assist at 75 rps



(a) k_p = 140 (b) k_p = 160 (c) k_p = 180

Fig. 9 FFT results at 70 rps with RMB assist

5. Conclusion

In this research, we experimentally investigated the relationship between the RMB assist control parameter and instability. First, with RMB assist higher rotational speeds were achieved than without the assist. Second, at rotational speeds of less than the speed at which we see the onset of self-excited vibrations, the RMB assist had no effect on the rotor vibration because the inertia force dominated the stiffness force of the RMB. Furthermore, the excessive gain caused the instability. Finally, at rotational speeds higher than the onset speed, a small gain was unable to generate sufficient force to stabilize the self-excited vibrations, and self-excitation occurred. Future work will be the application of the proposed method to other fluid induced instabilities such as seal instability, and the development of an automatic tuning method for the gain.

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