VIBRATION CONTROL OF FLEXIBLE ROTOR SUPPORTED BY INCLINATION CONTROL MAGNETIC BEARINGS

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ABSTRACT

A new type of magnetic bearing is introduced to control the bending modes of a flexible rotor. The technique is intended to stabilize a flexible rotor which can experience a range of bending modes. The rotor is assumed to be flexible and supported by magnetic bearings. However, the midspan bearing is not allowed for the fundamental rotor functions. The rotor should run super critical speed. A newly developed magnetic bearing is introduced which is composed of pair of HB type magnetic bearings and can control inclination as well as position of the rotor. The center position of the bearing is controlled by the same directional force of both bearings, while the inclination is controlled by the opposite force of them. The position control capability is used to support the rotor, while the inclination control capability is used to increase damping of the bending modes. An experimental setup is made to confirm the proposed technique. The results are indicating high possibility of reducing the bending vibration by using the proposed pair of HB type magnetic bearing.

INTRODUCTION

Magnetic bearings have noncontacting supports and active control capabilities and have been considered adequate for highspeed flexible rotors (Dussaux, 1990). However, standard control systems are apt to be unstable when the rotating speed increases over the resonant frequency of the bending mode. Several control techniques have been proposed

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and tested; for example, modal control, H_∞ control and sliding mode control. But the technique was mainly used to control the rigid motion of the rotor, while the bending modes were treated as the uncertainty from the nominal model (Herzog and Bleuler, 1992; Fujita, Matsumura and Namerikawa, 1992; Cui and Nonami, 1992). Using this technique, the bending vibrations become robustly stable because they are fundamentally stable modes. For the supercritical rotor however, the bending vibrations should be damped by the magnetic bearings.

In this paper the rotor is assumed to be flexible and both ends of the rotor are supported by magnetic bearings. The midspan disc cannot be controlled directly because of the lack of an actuator. In the previous paper (Okada, et al., 1997), a practical robust modal controller was introduced to control both ends of magnetic bearings to control the first bending mode as well as the rigid modes. The higher bending modes are guaranteed to be robustly stable. First the local PD controller was designed to have enough stiffness and damping at both ends of the rotor. Then the bending mode controller was designed using the H_∞ mixture sensitivity problem. This controller was applied to control a three-disc flexible rotor with good results. However, it is still considered difficult to pass several bending modes and to reach high speed rotation.

In this paper a new type of magnetic bearing is introduced to control bending modes of a flexible rotor. The technique is intended to stabilize a flexible rotor which can experience a range of bending modes. The rotor should run over several critical speeds. A newly developed magnetic bearing is introduced which is composed of a pair of HB type magnetic bearings and can control inclination as well as position of the rotor. The center position of the bearing is controlled by the same directional force of both bearings, while the inclination is controlled by the opposite force of them. The position control capability is used to support the rotor, while the inclination control capability is used to increase damping of the bending modes. An experimental setup is made to confirm the proposed technique. The rotor has the midspan mass to produce the bending vibration. One end of the rotor is supported by a ball bearing while the other end is supported by the proposed magnetic bearings. The results are indicating good reduction of the first bending vibration by using the proposed pair of HB type magnetic bearing.

FLEXIBLE ROTOR AND INCLINATION CONTROL MAGNETIC BEARING

In this paper, a three mass rotor is treated as shown schematically in Fig. 1. One end of the rotor is supported by a ball bearing and the other end is supported by the proposed

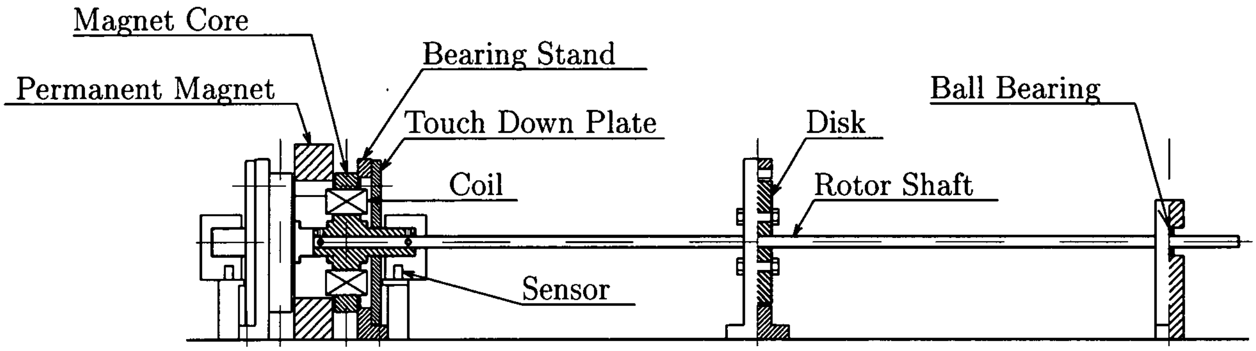


Figure 1. Schematic of rotor and magnetic bearing system

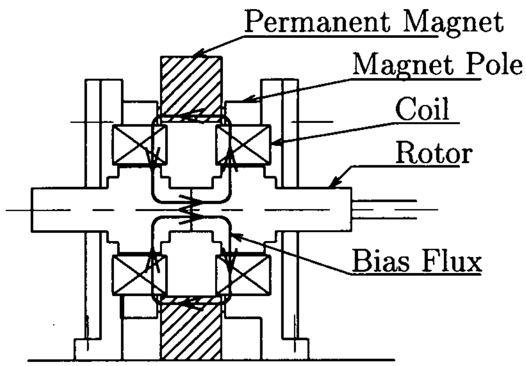


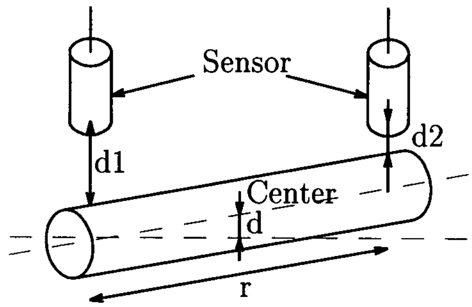
Figure 2. Inclination control magnetic bearing

magnetic bearings. The mass of the bearing disc is 0.39 [kg], while the mass of the center disc is 0.3615 [kg]. The rotor shaft has the diameter of $\phi 8$ [mm] and the length of 700 [mm]. The distances between the center disc and the both ends of bearings are 300 [mm]. A DC motor is attached to the right end of the rotor through rubber coupling.

Scheme of Inclination Control Magnetic Bearing

Scheme of the proposed inclination control magnetic bearing is shown in Fig. 2. The proposed magnetic bearing is composed of a pair of HB type magnetic bearings. The permanent magnet produces the constant bias flux as shown by the arrowed line. Touch down plates are attached on the outer sides of both HB bearings. The average airgap of magnetic bearings is 0.7[mm] while the gap between the rotor and the touch down plate is 0.5[mm].

Four gap sensors are installed on the outer sides of magnetic bearings; two for horizontal (x) and two for vertical (y) directions. The central displacement is determined by averaging the two sensing signals, while the inclination is calculated by the difference



$$
\begin{aligned}
Displacement &= d = \frac{d1 + d2}{2} \\
Angle &= \theta = \frac{d1 - d2}{r}
\end{aligned}
$$

Figure 3. Sensing of displacement and inclination

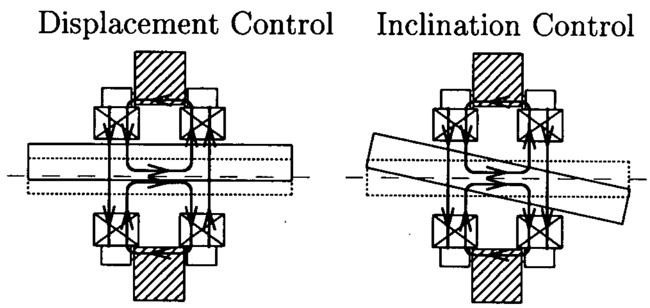


Figure 4. Operation of the magnetic bearings

of them, as shown schematically in Fig. 3. This sensing technique guarantees complete colocation of the sensor and actuator.

Operation of the proposed magnetic bearing is shown schematically in Fig. 4. The attractive force is controlled by addition or subtraction of the constant bias flux and the control flux. The attractive force is linearized by using the bias flux given by the permanent magnet. The linear force is produced by the same directional forces in both magentic bearings as shown in the left graph, while the moment is produced by the out phase force in both magnetic bearings as shown in the right graph in Fig. 4.

CONTROL SYSTEM

The controller used in this paper is the local PD controller. Transfer function of analog PD controller is shown by eqn. (1).

$$
G(s) = K_p + \frac{K_d s}{T_d s + 1} \tag{1}
$$

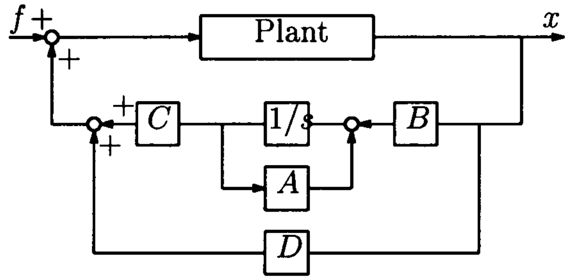


Figure 5. Block diagram of PD control

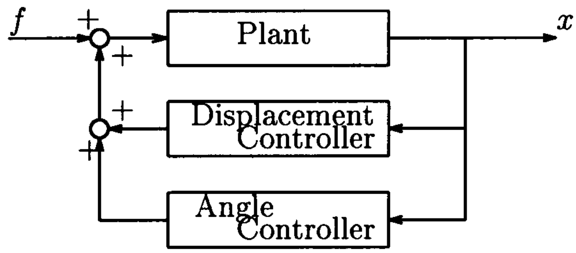


Figure 6. Block diagram of inclination control

where K_p and K_d are the displacement and the velocity feedback gains, T_d is the derivative time constant, respectively. The equivalent digital PD controller is given by,

$$\begin{cases} x_{k+1} &= e^{-\tau/T_d}x_k + u_k \\ y_k &= \dfrac{K_d}{\tau}(e^{-\tau/T_d} - 1)x_k + (K_p + \dfrac{K_d}{\tau})u_k \end{cases} \tag{2}$$

u_k : input signal
x_k : state variable
y_k : output signal

Block diagram is shown in Fig. 5.

Bending vibration control

The displacement PD controller is designed to guarantee the enough stiffness and damping to the central displacement of magnetic bearing. The inclination PD controller, however, is designed to give enough damping to the bending vibration. They are installed individually to x and y directions. The block diagram is shown in Fig. 6.

EXPERIMENTAL RESULTS AND CONSIDERATIONS

To confirm the above control technique, a three mass rotor was constructed. The schematic of the experimental setup is shown in Fig. 2. The rotor was set horizontally with right end of the shaft connected to a motor via a rubber coupling. It is capable of running up to 5,000 [rev./min.].

Four gap sensors are installed to measure the x and y displacements. The signal is put into DSP (TMS320C40) through 16 bit A/D converter. The displacement and the inclination controllers are transformed into digital one using the bilinear transformation. The DSP calculates the command signal to the magnetic bearings with the sampling interval of $\tau = 0.1$[ms]. Then the signal is put out to the power amplifier though 12 bit D/A.

Local PD control

First only the displacement controller is installed to determine the parameters; $K_{pd} = 4.5 \times 10^4$ [N/m], $K_{dd} = 11.34$ [Ns/m], while the inclination controller is $K_{pa} = 0.0$ [N/m], $K_{da} = 0.0$ [Ns/m], $T_d = 0.65$ [ms].

The rotor was excited by a sinusoidal centrifugal force with the frequency of rotation caused by the unbalance weight of the disc. This unbalance response is measured by the gap sensor and recorded as shown in Figs. 7 to 10. Figures 7 and 8 show the responses of the rotor supported only by the displacement PD controlled magnetic bearings. The outer gap response is shown in Fig. 7, while the inner gap response is indicated in Fig. 8. Both responses indicate the effect of fundamental bending vibration at 1,500 [rev./min.]. The midspan disc is touched down to the outer case and the rotor cannot run super critical speed.

Inclination control

In addition to the displacement PD control, the inclination PD control is installed and the responses are shown in Figs. 9 and 10. The bending vibration is well reduced as shown by the outer and inner responses in Fig. 9 and 10, respectively. The displacement controller used is $K_{pd} = 4.5 \times 10^4$ [N/m], $K_{dd} = 39.7$ [Ns/m], and the inclination controller is $K_{pa} = 0.0$ [N/m], $K_{da} = 34.02$ [Ns/m] and $T_d = 0.65$ [ms].

The peak response is well reduced and the rotation is smooth. The rotor can run up to its top speed. This responses indicate the superiority of the proposed inclination control magnetic bearings.

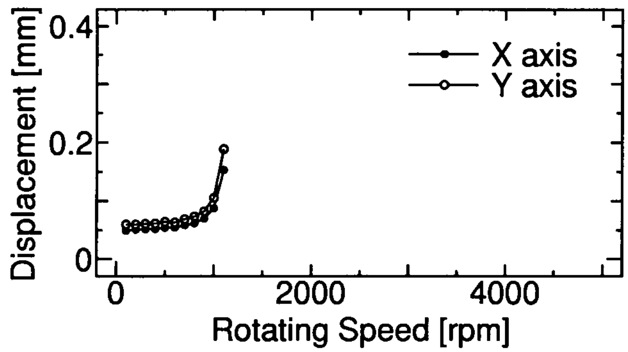


Figure 7. Unbalance response of outside magnetic bearing without the inclination control

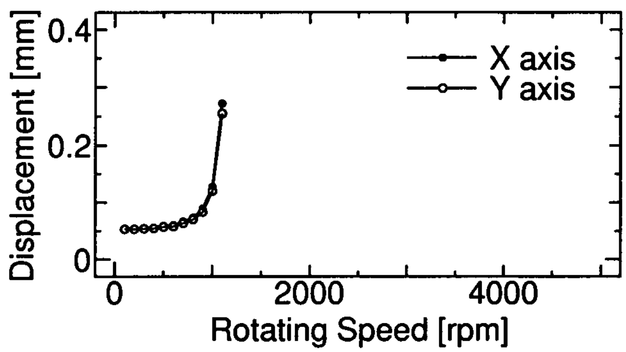


Figure 8. Unbalance response of inside magnetic bearing without the inclination control

CONCLUSIONS

A new type of magnetic bearing is introduced to control bending modes of the flexible rotor. The developed magnetic bearing is composed of a pair of HB type magnetic bearings and can control inclination as well as position of the rotor. The position control capability is used to support the rotor, while the inclination control capability is used to increase damping of the bending modes. An experimental setup of three mass rotor is made to confirm the proposed technique. The results are indicating good reduction of the first bending vibration by using the proposed pair of HB type magnetic bearing. Further work is continuing to apply the proposed magnetic bearing to multi-mass rotor and to reduce several bending resonances.

REFERENCES

Cui, W.-M. and K. Nonami, 1992."H^∞ Control of Flexible Rotor- Magnetic Bearing System," *Proc. of the 3nd Int. Symp. on Magnetic Bearings*, Alexandria, VA, 505-514

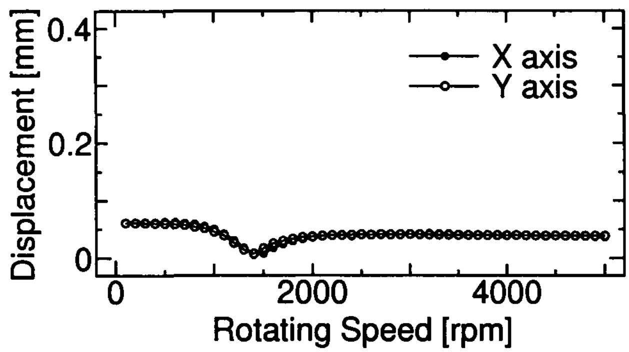


Figure 9. Unbalance response of outside magnetic bearing with the inclination control

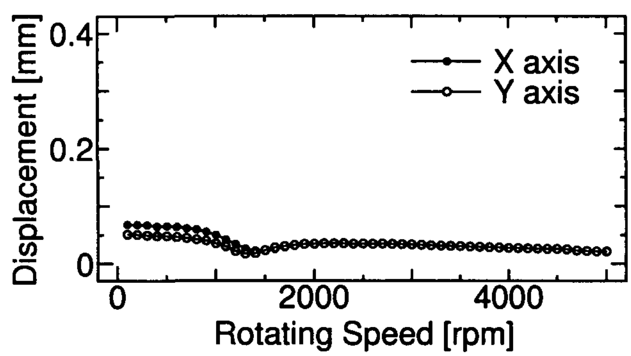


Figure 10. Unbalance response of inside magnetic bearing with the inclination control

Dussaux, M., 1990."The industrial applications of the active magnetic bearing technology," *Proc. of the 2nd Int. Symp. on Magnetic Bearings*, Tokyo, Japan, 33-38

Fujita, M., F. Matsumura and T. Namerikawa, 1992."μ-Analysis and Synthesis of a Flexible Beam Magnetic Suspension System," *Proc. of the 3nd Int. Symp. on Magnetic Bearings*, Alexandria, VA, 495-502

Herzog, R. and H. Bleuler, 1992."On Achievable H^{∞} Disturbance Attenuation in AMB Control," *Proc. of the 3nd Int. Symp. on Magnetic Bearings*, Alexandria, VA, 485-494

Okada, Y., H. Hayashi, T. Saito and Y. Shinoda, 1997."Vibration Control of Flexible Rotor Supported by Magnetic Bearings," *Proc. of Asia-Pacific Vibration Conference '97*, Kyonju, Korea, 2, 991-993