Active Stabilization of a Repulsive Magnetic Bearing Using the Motion Control of Permanent Magnets*

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This paper investigated the control system design of a repulsive magnetic bearing using the motion control of permanent magnets. In the repulsive magnetic bearing system, the radial motions of the rotor were passively supported by repulsive forces between ring-shape permanent magnets; the axial motion was actively controlled by the motion control of the permanent magnets with a pair of voice coil motors. Two control schemes, PD control and state-variable feedback, were applied for the stabilization of the axial motion. The results showed that the dynamic performance of the controlled system could be adjusted more flexibly by state-variable feedback.

Key Words: Magnetic Bearing, Actuator, Positioning, Mechatronics, Motion Control, Permanent Magnet

1. Introduction

There are several methods of supporting a moving or rotating mass by using electromagnetic forces without any mechnical contact[1),(2]. One of the principal methods is to use repulsive forces between permanent magnets. Levitation system using this method has several advantages; it is stable in the direction of repulsive force (normal direction); no energy is required to generate levitation force. It is useful for supporting a mass in combination with mechanical guidances or reducing its load on a conventional bearing. It is also used as a passive support element in apparatuses with active magnetic bearings. However, its use is limited to such applications because of instability in the lateral directions and poor damping in the normal direction.

As a mean of compensating these disadvantages, we proposed to introduce the motion control of permanent magnets into this type of levitation system[(3)-(6)]. Levitation systems with performance of positioning and vibration control in the normal direction can be realized by inserting an actuator between a levitation magnet and its base[(3)]. Lateral motion can be stabilized by moving a magnet for support like an inverted pendulum[(4)]. A repulsive magnetic bearing system was developed by applying the latter principle to suspend a rotating mass (rotor)[(5),(6)]. In the developed system, the radial motions of the rotor were passively supported by repulsive forces between ring-shape permanent magnets; the axial motion was actively controlled by the motion control of the permanent magnets with a pair of voice coil motors.

This paper investigates the control system design of the magnetic bearing system both theoretically and experimentally. Two control schemes are applied for the stabilization of the axial motion. A state-variable feedback scheme is compared with the output feedback scheme that was used in the previous work. It will be shown that the dynamic performance of the controlled system can be adjusted flexibly with the state-variable feedback scheme.

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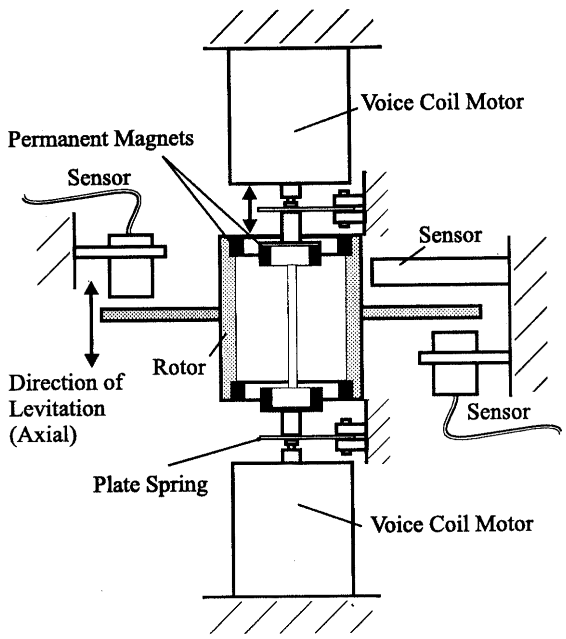


Fig. 1 Repulsive magnetic bearing apparatus

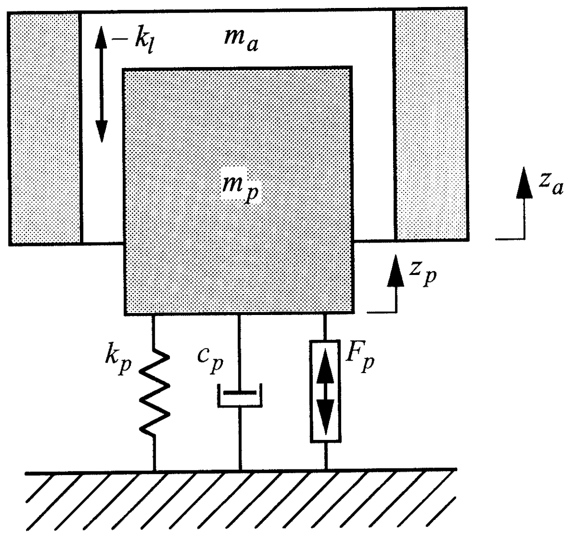


Fig. 2 Basic model (k_l: lateral factor)

2. Modeling

2.1 Structure of a repulsive magnetic bearing

Figure 1 shows a schematic diagram of the developed magnetic bearing apparatus using the motion control of permanent magnets[6]. It is an outer-rotor type. The rotor has two ring-shape permanents at its top and bottom. Two voice coil motors are fixed to the ceiling and the base of the apparatus. Each motor drives a ring-shape permanent magnet for support. Each of the permanent magnets for support is connected to the base through a plate spring to obtain restoring forces without control. The length between the two permanent magnets is kept constant by constraining the movements of the two motors mechanically. Thus, this system can be treated as a scalar input system.

2.2 Modeling

Figure 2 shows a physical model of the system illustrated by Fig. 1. It is assumed for simplicity that the rotor m_a moves only in the axial direction. The total force produced by the voice coil motors is denoted by $F_p(t)$. The gravitational force acting on the rotor and the lateral forces between the permanent magnets are balanced in the equilibrium states. For small deviations from the equilibrium, the equations of motion become

$$m_a\ddot{z}_a=k_l(z_a-z_p), \tag{1}$$

$$m_p\ddot{z}_p=k_l(z_p-z_a)-c_p\dot{z}_p-k_pz_p+F_p(t) \tag{2}$$

where

z_a : displacement of the rotor,
z_p : displacement of the permanent magnets for support,
m_p : mass of the support including permanent magnets driven by the voice coil motors,
k_p : stiffness of the plate springs,
c_p : damping coefficient between the support and the base,
k_l : lateral factor between the permanent magnets.

The voice coil motors are controlled to produce force following a command signal inputted to a driver circuit:

$$F_p(t)=k_au(t), \tag{3}$$

where

$u(t)$: input voltage to the driver circuit,
k_a : gain of the driver circuit.

From Eqs. (1), (2) and (3), a state space model describing the dynamics of the system is obtained as

$$\dot{\boldsymbol{x}}(t)=\boldsymbol{A}\boldsymbol{x}(t)+\boldsymbol{B}u(t), \tag{4}$$

where

$$\boldsymbol{x}=\begin{bmatrix} z_a \\ \dot{z}_a \\ z_p \\ \dot{z}_p \end{bmatrix},\quad \boldsymbol{A}=\begin{bmatrix} 0 & 1 & 0 & 0 \\ a_0 & 0 & -a_0 & 0 \\ 0 & 0 & 0 & 1 \\ -a_1 & 0 & a_1-a_2 & -a_3 \end{bmatrix},\quad \boldsymbol{B}=\begin{bmatrix} 0 \\ 0 \\ 0 \\ b_0 \end{bmatrix},$$

$$a_0=\frac{k_l}{m_a},\ a_1=\frac{k_l}{m_p},\ a_2=\frac{k_p}{m_p},\ a_3=\frac{c_p}{m_p},\ b_0=\frac{k_a}{m_p}.$$

The transfer function from the input to the displacement of the rotor is obtained as

$$G_a(s)=\frac{Z_a(s)}{U(s)}$$

$$=-\frac{a_0b_0}{s^4+a_3s^3+\{a_2-(a_0+a_1)\}s^2-a_0a_3s-a_0a_2}. \tag{5}$$

Equation (5) shows that this system is unstable because the characteristic polynomial has both positive and negative coefficients. Thereby, feedback control is necessary for contactless levitation.

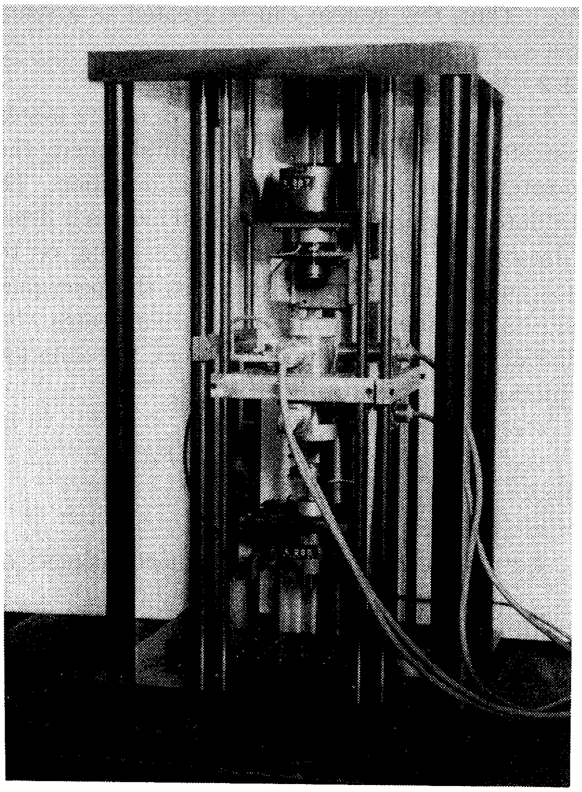


Fig. 3 Photograph of the apparatus

3. Control System Design

3.1 PD control

The PD control is a fundamental control scheme for stabilization. The control input is represented by

$$u(t)=p_d z_a(t)+p_v \dot{z}_a(t)+v(t), \qquad (6)$$

where p_d and p_v is the gains of displacement and velocity feedback, and $v(t)$ is an auxiliary input. From Eqs.(5) and (6), the following equation is obtained.

$$\frac{Z_a(s)}{V(s)}=\frac{a_0 b_0}{t_c(s)}, \qquad (7)$$

where

$$t_c(s)=s^4+a_3 s^3+(a_2-a_0-a_1)s^2 + a_0(b_0 p_v-a_3)s+a_0(b_0 p_d-a_2). \qquad (8)$$

Necessary conditions for achieving stable suspension by PD control are

$$a_3>0, \qquad (9)$$

$$a_2>a_0+a_1. \qquad (10)$$

Thus, the spring and damping elements suspending the permanent magnets must be selected to satisfy[6]

$$c_p>0, \qquad (11)$$

$$k_p>\left(1+\frac{m_p}{m_a}\right)k_l. \qquad (12)$$

When these conditions are satisfied, this system can be stabilized by adjusting the gains p_d and p_v.

3.2 State feedback control

Since the system described by Eq.(4) is controllable, the closed-loop poles can be arbitrarily assigned by state variable feedback. The control input is represented by

$$u(t)=\boldsymbol{Fx}(t)+v(t) = p_d z_a+p_v \dot{z}_a+q_d z_p+q_v \dot{z}_p+v(t), \qquad (13)$$

where

$$F=[p_d \quad p_v \quad q_d \quad q_v].$$

From Eqs.(4) and (13), the following equation is obtained.

$$\frac{Z_a(s)}{V(s)}=-\frac{a_0 b_0}{t_c'(s)} \qquad (14)$$

where

$$t_c'(s)=s^4+(a_3-b_0 q_v)s^3+(a_2-a_0-a_1-b_0 q_d)s^2 + a_0\{b_0(p_v+q_v)-a_3\}s+a_0\{b_0(p_d+q_d)-a_2\} \qquad (15)$$

Let the desired characteristic polynomial denoted by

$$\Delta(s)=(s^2+2\zeta_1\omega_1 s+\omega_1^2)(s^2+2\zeta_2\omega_2 s+\omega_2^2) = s^4+c_3 s^3+c_2 s^2+c_1 s+c_0. \qquad (16)$$

Comparing Eq.(15) with Eq.(16) gives

$$q_v=\frac{1}{b_0}(a_3-c_3), \qquad (17)$$

$$q_d=\frac{1}{b_0}(a_2-a_0-a_1-c_2), \qquad (18)$$

$$p_v=q_v+\frac{1}{b_0}\left(\frac{c_1}{a_0}+a_3\right), \qquad (19)$$

$$p_d=q_d+\frac{1}{b_0}\left(\frac{c_0}{a_0}+a_2\right). \qquad (20)$$

4. Experiment

4.1 Experimental apparatus

Figure 3 is a photograph of the developed experimental apparatus. The values of the parameters of the apparatus are listed in Table 1.

Table 1 Parameters of the apparatus

m_a	0.353 kg
m_p	0.350 kg
k_p	1.25×10^5 N/m
c_p	19.3Ns/m
k_l	2.46×10^3 N/m
k_a	3.92N/V

Figure 4 shows an outlook of the rotor. It has two ring-shape permanent magnets with inner and outer diameters of 24 mm and 32 mm. All the permanent magnets are made of SmCoB materials. The radial and axial motions of the rotor are detected by eddy-current gap sensors.

The ring-shape permanent magnets for support have a 7-mm inner diameter and a 12-mm outer diameter. They are driven by a pair of voice coil

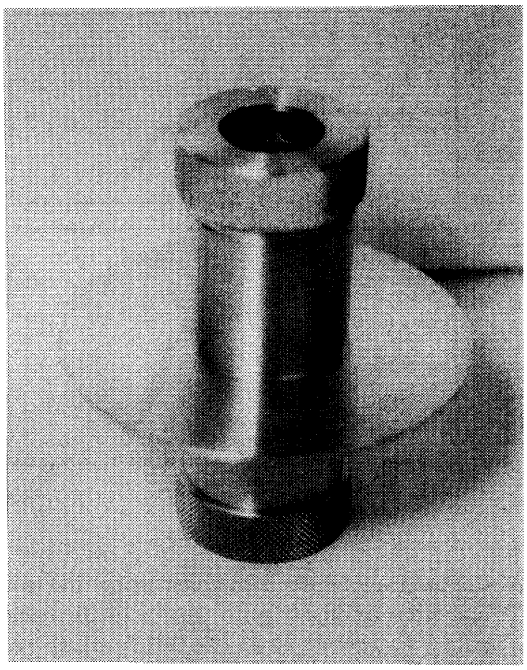
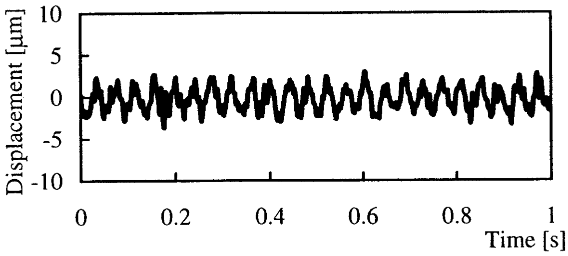
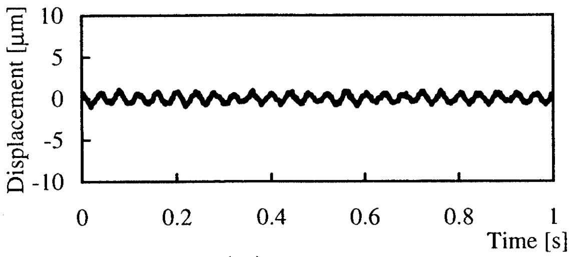


Fig. 4 Photograph of the rotor



(a) Magnet for support



(b) Rotor

Fig. 5 Levitation accuracy of a system using PD control

motors with a stoke of 10 mm and a maximum output force of 9.8 N. The displacement of the movers in the axial direction is also detected by an eddy-current gap.

These signals are inputted to a DSP-based digital controller through A/D converters. The controller calculates control input according to Eqs. (6) and (13). The output signal is converted to analog with a D/A converter. Two current-output amplifiers drive the voice coil motors according to the output signal.

4.2 PD control

Figure 5 shows the movements of the rotor and

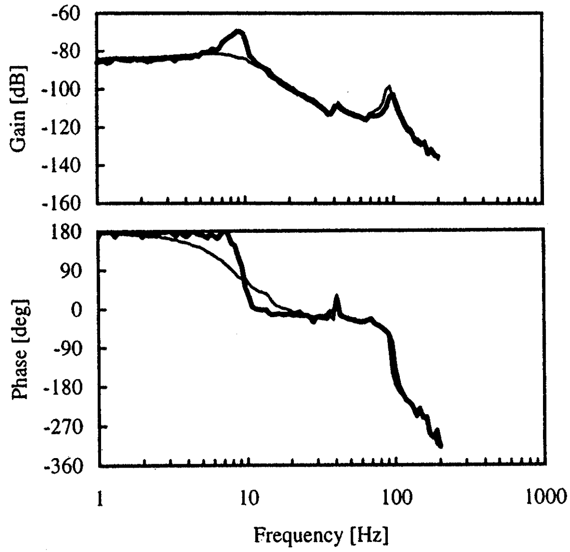


Fig. 6 Frequency responses of systems using PD control (a) ——: $p_d=5.95\times10^4$ [V/m], $p_v=0.95\times10^2$ [Vs/m], (b) ——: $p_d=5.95\times10^4$ [V/m], $p_v=3.79\times10^2$ [Vs/m]

the magnets for support when the rotor levitates with a PD controller whose parameters are

$$p_d=6.80\times10^4\,[\text{V/m}],\quad p_v=4.36\times10^2\,[\text{Vs/m}].$$

The deviation of the rotor from the equilibrium position is kept within ±1 μm.

Figure 6 shows frequency responses of the rotor displacement $z_a(t)$ to the auxiliary input $v(t)$. The gains are selected as

(a) $p_d=5.95\times10^4$ [V/m], $p_v=0.95\times10^2$ [Vs/m] (thick line),

(b) $p_d=5.95\times10^4$ [V/m], $p_v=3.79\times10^2$ [Vs/m] (thin line).

The resonance peak at about 10 Hz is suppressed by increasing the velocity gain. However, the damping characteristics at higher frequencies (about 100 Hz) are made worse.

4.3 State feedback control

Figure 7 shows the movements of the rotor and the magnets for support when the rotor levitates with a state-variable feedback that is designed as

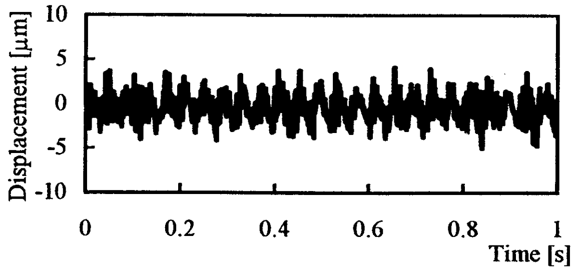
$$\omega_1=94.2\,[1/s],\quad \omega_2=565\,[1/s],\quad \zeta_1=0.6,\quad \zeta_2=0.3.$$

The deviation of the rotor from the equilibrium position is almost same as that of the PD-controlled system shown by Fig. 4. In contrast, the motion of the magnets for support includes higher-frequency components than that of the PD-controlled system.

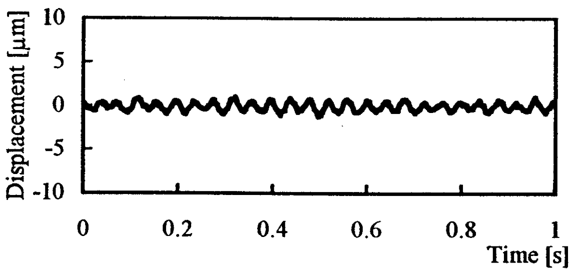
Figure 8 shows frequency responses of the rotor displacement $z_a(t)$ to the auxiliary input $v(t)$. The poles are selected as

(a) $\omega_1=75.4$ [1/s], $\omega_2=565$ [1/s], $\zeta_1=0.2$, $\zeta_2=0.1$ (thick line),

(b) $\omega_1=75.4$ [1/s], $\omega_2=565$ [1/s], $\zeta_1=0.6$, $\zeta_2=0.1$ (thin line).



(a) Magnet for support



(b) Rotor

Fig. 7 Levitation accuracy of a system using state feedback

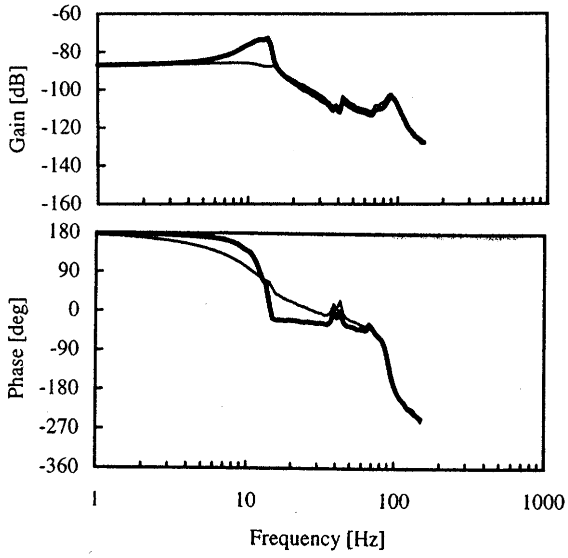


Fig. 8 Frequency response of systems using state feedback (a) ——: $\omega_1=75.4$ [1/s], $\omega_2=565$ [1/s], $\zeta_1=0.2$, $\zeta_2=0.1$ (b) ——: $\omega_1=75.4$ [1/s], $\omega_2=565$ [1/s], $\zeta_1=0.6$, $\zeta_2=0.1$

The damping characteristics in a low-frequency range (about 10 Hz) are improved by increasing ζ_1 without worsening the higher-frequency characteristics.

Figure 9 compares the frequency response of a PD-controlled system (thick line) with that of a state-feedback-controlled system (thin line) ; both the systems maximize the displacement feedback gain P_d. The maximal physically realizable gain is

(a) PD control : $p_d=1.79\times10^5$ [V/m],

(b) State feedback : $p_d=3.30\times10^5$ [V/m].

These results show that the dynamic performance can be adjusted more flexibly by using the state-feedback control scheme.

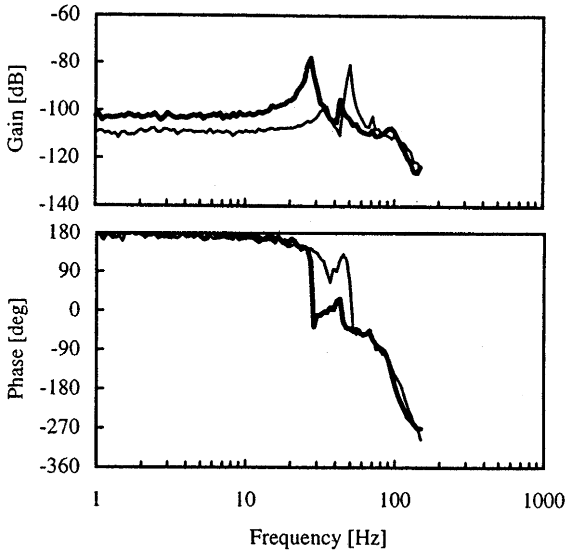


Fig. 9 Comparison between systems using PD control and state feedback as to the maximal realizable gain. (a)——: $p_d=1.79\times10^5$ [V/m], $p_y=4.13\times10^2$ [Vs/m], $q_d=0$, $q_v=0$ (b)——: $p_d=3.30\times10^5$ [V/m], $p_v=9.10\times10^2$ [Vs/m], $q_d=1.00\times10^4$ [V/m], $q_v=2.85\times10^2$ [Vs/m]

5. Conclusion

The control system design of a repulsive magnetic bearing using the motion control of permanent magnets was discussed. PD control and state-variable feedback were applied to stabilize the system in the axial direction. The experimental results showed that the dynamic performance of the controlled system could be adjusted more flexibly by the state feedback control scheme.

Other control schemes including integral action are also applied to the developed repulsive magnetic bearing[7]. Research on another repulsive magnetic bearing using piezoelectric actuators is under way[8].

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