Experimental study on the Adjustability of Radial Stiffness in a Repulsive Magnetic Bearing Device

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ABSTRACT

This paper studies the ability to adjust the stiffness in the radial direction on a non-contact levitated rotor by using an independent motion control of permanent magnets. The method of stabilization in the axial direction, by moving a magnet for support like an inverted pendulum, is applied. The repulsive forces in the radial direction depend not only on the size and physical characteristics of magnets but also on the relative positioning of magnets to each other in the axial direction. This work shows the principles and the one of necessary methods of motion control for successful adjustability of the radial stiffness.

INTRODUCTION

There are several methods to support a moving or rotating mass by using magnetic forces without any mechanical contact [1-2]. One of the principal methods is to use the repulsive forces of permanent magnets. In this type of magnetic bearing system the object is levitated by the repulsive forces between permanent magnets. Such a system does not need any energy to generate the levitation force and is stable in the direction of repulsive forces (FR), but unstable in the normal direction to the FR. In the previous works, experimental devices, using PD and also state feedback control [3 - 4] was developed where the levitated object was a cylindrical rotor and permanent magnets were ring-shaped. While the permanent magnets for support are mechanically connected in chose works, the independent motion control of permanent magnets has been proposed [5 - 6]. The repulsive forces in radial direction can be changed by adjusting of the relative position of the magnets for support to each other. Since we use two pair of ring-shape permanent magnets (Fig. 1), such adjustment is achieved by controlling the motion of magnets for support independently. In this work, the feasibility of such stiffness adjustment is studied experimentally.

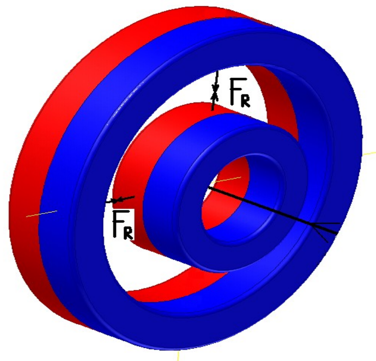

FIGURE 1: Positioning of ring shape permanent magnets

MECHANICAL CONSTRUCTION

Figure 2 shows a schematic diagram of the developed magnetic bearing apparatus using the motion control of permanent magnets. It is an outer-rotor type and includes one ring-shape permanent magnet at the each end. Each voice coil motor drives an inner permanent magnet for support. The displacement of rotor is detected by sensors 1 and 2. Since all motions in the radial directions are passively supported by repulsive forces, we assume for the simplicity, that the rotor moves only in the axial direction.

MODELING AND SIMPLIFICATION

Figure 2 shows a physical model of the system illustrated by Figure 3. where

$Fp1\ (t),\ Fp2\ (t)$: Force generated by VCM's
$Cp1,\ Cp2$: Damping due to the friction inside of VCM's
$kp1,\ kp2$: Stiffness of springs inside of VCM's
ma : Mass of rotor
$mp1,\ mp2$: Left and right mass driving by theVCM's
$kl1,\ kl2$: Lateral factor's between the magnets

The total forces produced by voice coil motors are F_{p1} and F_{p2}. The gravitational force acting on the rotor and the radial forces between the permanent magnets are balanced in the equilibrium states.

Considering the symmetry, we assume that:

$$kl1 = kl2\ (\equiv k_l) \tag{1}$$

$$mp1 = mp2\ (\equiv m_p) \tag{2}$$

$$cp1 = cp2\ (\equiv c_p) \tag{3}$$

$$kp1 = kp2\ (\equiv k_a) \tag{4}$$

Also it is assumed that the input signal to the both amplifier is same. Then we can simplify the physical model as shown by Figure 4. Table 1 shows the physical parameters of the magnetic bearing apparatus.

TABLE 1: Physical parameters**.**

Variable	Value	Units
m_a	0.16	Kg
m_p	0.170	kg
k_p	8500	N/m
k_l	3000	N/m
k_i	10	N/A
c_p	7.5	Ns/m

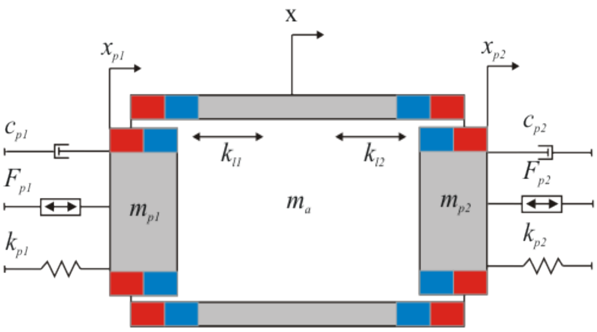

FIGURE 2: Physical Model of the magnetic bearing apparatus.

CONTROL DESIGN

To achieve the levitation several control schemes are applicable. We compare three different control methods, PD, I-PD and the single I-PD. The Figures 5, 6 and 7 show the diagrams for the PD, the I-PD and single I-PD control, respectively.

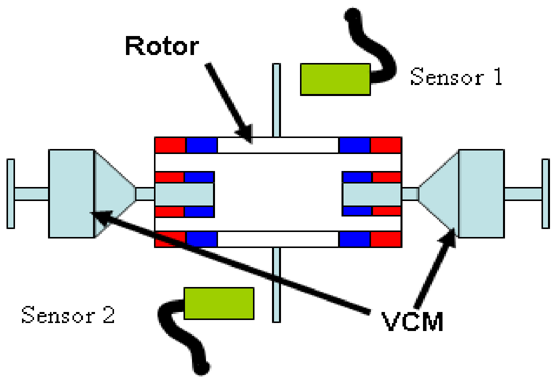

FIGURE 3: Repulsive magnetic bearing apparatus. The outer magnets inner diameter is 24 mm and outer diameter is 32 mm. The ring shape permanent magnets for support have a 7mm inner and 12 mm outer diameter.

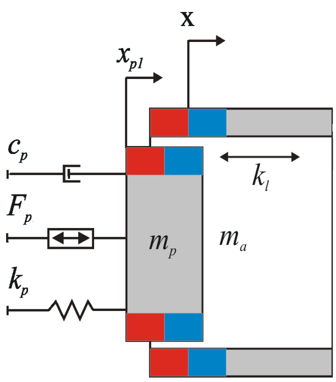

FIGURE 4: Simplified physical model.

Following the equation of motion for the simplified system:

$$m_a \ddot{x} = k_l (x - x_p) \tag{5}$$

$$m_p \ddot{x}_p = k_l (x_p - x) - c_p \dot{x}_p - k_p x_p + k_i * i(t) \tag{6}$$

From (5), (6) and included the local close loop control of the both VCM's the state space model describing the dynamics of the system is obtained as:

$$\dot{x}(t) = Ax(t) + Bu(t) \tag{7}$$

where

$$A_s = \begin{bmatrix} 0 & 1 & 0 & 0 \\ \frac{2k_p - k_i * xp_d}{m_p} & -\frac{c_p - k_i * xp_v}{m_p} & 0 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{k_l}{m_a} & 0 & \frac{k_l}{m_a} & 0 \end{bmatrix}, B_s = \begin{bmatrix} 0 \\ \frac{k_i}{m_p} \\ 0 \\ 0 \end{bmatrix} \tag{8}$$

$$\boldsymbol{x}_s = \begin{bmatrix} x_p \\ \dot{x}_p \\ x \\ \dot{x} \end{bmatrix},$$

Since the system described by (7) and (8) is controllable, the closed-loop poles can be arbitrary assigned by state feedback. The closed-loop gains for the all control schemes of the whole system are selected through the numerical simulations and later fine adjustment while experimental operation.

The PD control input is represented by

$$u(t) = K * x_s(t) \tag{9}$$

$$= p_d x + p_v \dot{x} \tag{10}$$

where

$$K = [0 \quad 0 \quad p_d \quad p_v]. \tag{11}$$

The values of the matrix K for the closed-loop control become as follow:

$p_d = -1320$ [A/m]
$p_v = -16$ [As/m].

The I-PD control input is represented by

$$u(t) = IF \int (u(t) - x(t))dt - Kx_s(t) \tag{12}$$

$$= IF \int (u(t) - x(t))dt - p_d x - p_v \dot{x} \tag{13}$$

where

$$K = [0 \quad 0 \quad p_d \quad p_v] \tag{14}$$

The values of the matrix K and integral factor IF for the I-PD control become as follow:

$IF = -15000$
$p_d = -2000$ [A/m]
$p_v = -30$ [As/m].

Since in case of the single I-PD the control input is same as for I-PD but due the fact that the control input is only to the one of two VCM applied we have to consider the changed k_l factor. However the control input is same as (12) to (14) and the corrected values are:

$IF = -15000$
$p_d = -3000$ [A/m]
$p_v = -50$ [As/m].

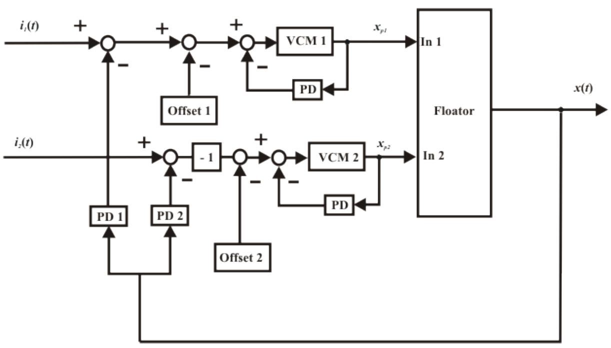

FIGURE 5: Diagram of PD Control

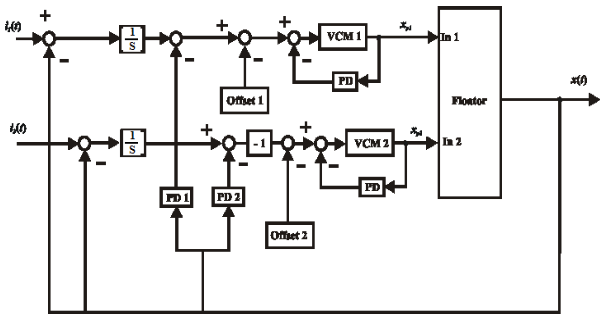

FIGURE 6: Diagram of I-PD Control

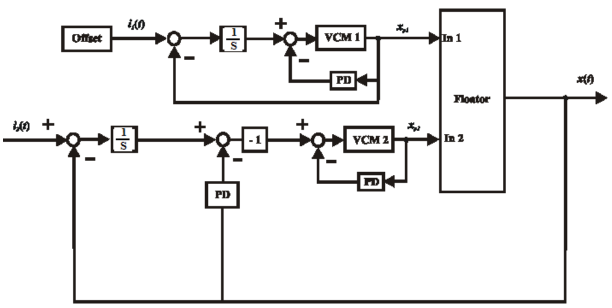

FIGURE 7: Diagram of single I-PD Control

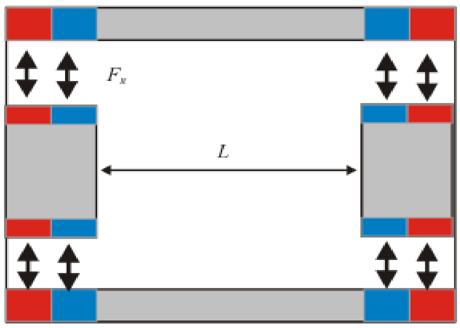

FIGURE 8: Full overlapping of magnets delivers the maximum amount of repulsive forces

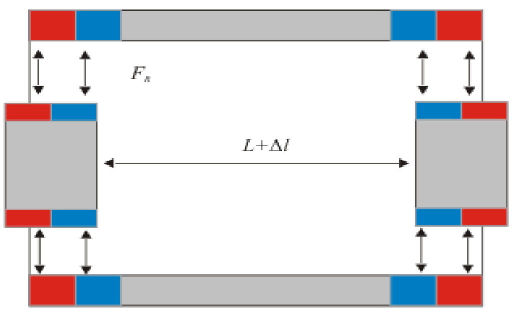

FIGURE 9: Smaller repulsive forces as a result of smaller overlap

ADJUSTMENT ON RADIAL STIFFNESS

The amount of the repulsive forces is directly proportional to the lap of the magnet pair. When the overlap is full the repulsive forces are maximal as shown in Figure 8. Since we can control the motion of magnets independently, we can change the distance L between two inner magnets and therefore make the overlap area smaller. As a result, we achieve the decreasing of repulsive forces as shown in Figure 9. The maximal possible ΔL is mainly dependent on the following factors: the control scheme, the maximal possible coil current for the VCM and also of the magnet design. In our experiments we could realize the ΔL of maximal 1.5 mm without any drawback for the levitation. For the successful increasing of the distance L between the magnets for support we changed the "Offset 1" and "Offset 2" values by using PD and I-PD control methods, and by setting of the "Offset" value by using the single I-PD control. Adding an offset we observed the position of floator also in axial direction to make sure that it is not moved from the initial one

EXPERIMENT

The feasibility of stiffness in the radial direction was investigated. For decreasing the overlap area is necessary to increase the distance between the magnets for support. Therefore we added a constant offset signal with the same amount but with the opposite sign to each voice coil motor. After increasing the distance in the range of 0 to 1.5 mm, the displacement in the radial direction, by adding different masses on the point "M" in Fig. 10, have been measured and compared. Figures 11, 12 and 13 show the results in detail and the Tables 2, 3 and 4 show the summarized overview of the the spring constant denoted by K [N/m].

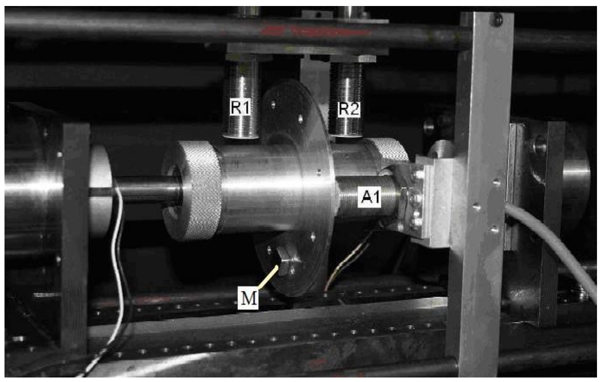

FIGURE 10: Magnetic bearing apparatus. The sensors A1 and A2 (A2 is not in the picture) detect the motion in the axial direction and sensors R1 and R2 in the radial direction

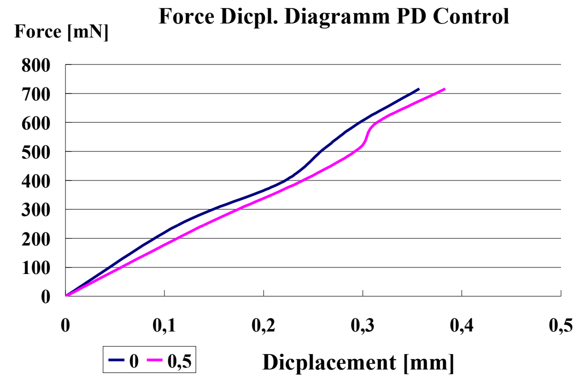

FIGURE 10: Force – Displacement diagram for the PD control and △L of 0 and 0.5 mm

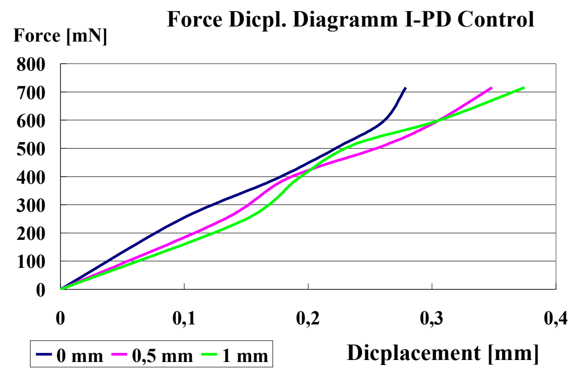

FIGURE 11: Force – Displacement diagram for the I-PD control and △L of 0, 0.5 and 1 mm

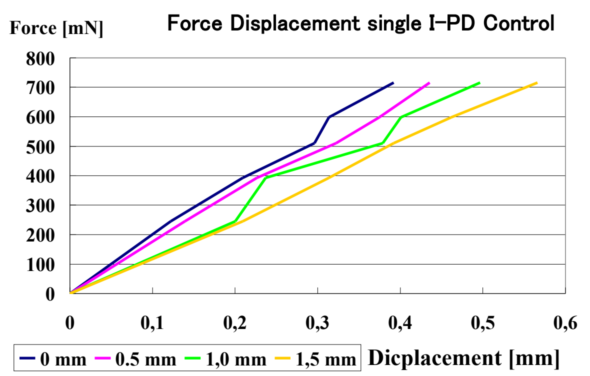

FIGURE 12: Force – Displacement diagram for the I- PD control and △L of 0, 0.5, 1 and 1.5 mm

TABLE 2: The average Spring Constant K for PD Control

$\triangle L$[mm]	0,0	0,5
K [N/mm]	2004,88	1868,19

TABLE 3: The average Spring Constant K for I-PD Control

$\triangle L$ [mm]	0,0	0.5	1,0000
K [N/mm]	2568,76	2055,01	1911,64

TABLE 4: The average Spring Constant K for single I-PD Control

$\triangle L$[mm]	0,00	0,50	1,00	1,50
K [N/mm]	1826,7	1644,0	1442,1	1264,6

CONCLUSIONS

The adjustability on radial stiffness, was studied. The experimental results show that the stiffness can be adjusted, also by using different control methods. In the current configurations, the range of adjustment is around 15-30%. However the adjustability can be improved by introducing more sophisticated control methods, by modifying the design of permanent magnets and by changing of operating mode of VCM. Research in such improvement is under way.

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