SINGLE AXIS CONTROLLED ATTRACTION TYPE MAGNETIC LINEAR BEARING

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Abstract - This paper presents a new type of magnetic linear bearing with active control only in one direction of the one magnetically levitated table. This architecture of linear bearing uses two pairs of permanent magnets working in attraction mode to ensure stable levitation of the bearing table along the y-axis direction and around x, y and z-axis simultaneously and a control system composed of two electromagnets, a gap sensor and a controller to keep the table in a fixed z position. The principle, the dynamic model for z direction table motion and the control system for this bearing are presented. Finally, by experiments conducted in a prototype, it is shown the effectiveness of the principle here presented.

Key words: magnetic linear bearing, mechatronics, controlled bearing

1. Introduction

Currently, one important segment of industry, the microelectronics industry, are looking for an alternative machine element for high precision linear or rotary motion that can operate inside a vacuum chamber, where ICs of latest generation are nowadays produced. For this purpose, the magnetic bearing is one promising solution. There are many researches concerning magnetic bearings but most of them treats the bearing in which at least 5 d.o.f (degrees of freedom) of the object are controlled (for example, Youcef and Reddy, 1992; Gangbing and Ranjan, 1996; Marcio et al., 1996 and Schweitzer, 1992). Since the control of each d.o.f, requires a sensor, an actuator and a controller, the entire system becomes complex in terms of the design of its mechanical / electrical part and the control system. Considering this, this paper presents a new principle of magnetic linear bearing, in which only 1 d.o.f. of a table, along its moving direction, here named *z*-axis direction, is actively controlled. The remaining degrees of freedom in other directions are restricted only by the action of permanent magnets. Different from similar type of bearing proposed by other authors (for example, Ohji et al., 1996), here permanent magnets will work in attraction mode, in order to avoid the problem of demagnetization (Campbell, 1994). The active control, at least in 1.d.o.f, is necessary since it is impossible to reach the stability only by permanent magnets. This is a consequence of Earnshaw´s principle (Earnshaw, 1939). There are researches showing the possibility of constructing magnetic bearings without any active control by using superconductor materials (for example, Marion-Péra et al., 1994) but, up to the present moment, such materials only behave as superconductors below 77K temperature degrees, and it will take some time to improve this kind of materials so that they become useful.

2. The principle of the new bearing

Figure (1) shows the schematics of the proposed magnetic linear bearing. A permanent magnet is fixed to each extremities of a moving table that is positioned between two stationary actuators. Each actuator is a combination of an electromagnet and a permanent magnet. The polarity of each magnet is as described in Fig. (1). An attraction force acts between each pair of magnets. This configuration assures stable equilibrium of the table along the *y* direction. As described later, assuring a minimum length and width to the table, the two pairs of magnets assure the table angular stability around the *x*, *y* and *z*-axis respectively. The table stability along the *z* direction is assured by a control loop, which is composed of a non-contact type gap sensor, a controller and two electromagnetic actuators.

3. Magnetic stiffness and table rotation stability analysis

Magnetic stiffness K_ϕ and K_θ, against table rotation around the *x* and *y*-axis respectively, are assured by attraction force between permanent magnets attached in the extremities of the bearing table (length *l*) and those, attached to a fixed base Fig. (2). K_ϕ and K_θ are described by Eqs. (1) and (2) (Delamare, 1994).
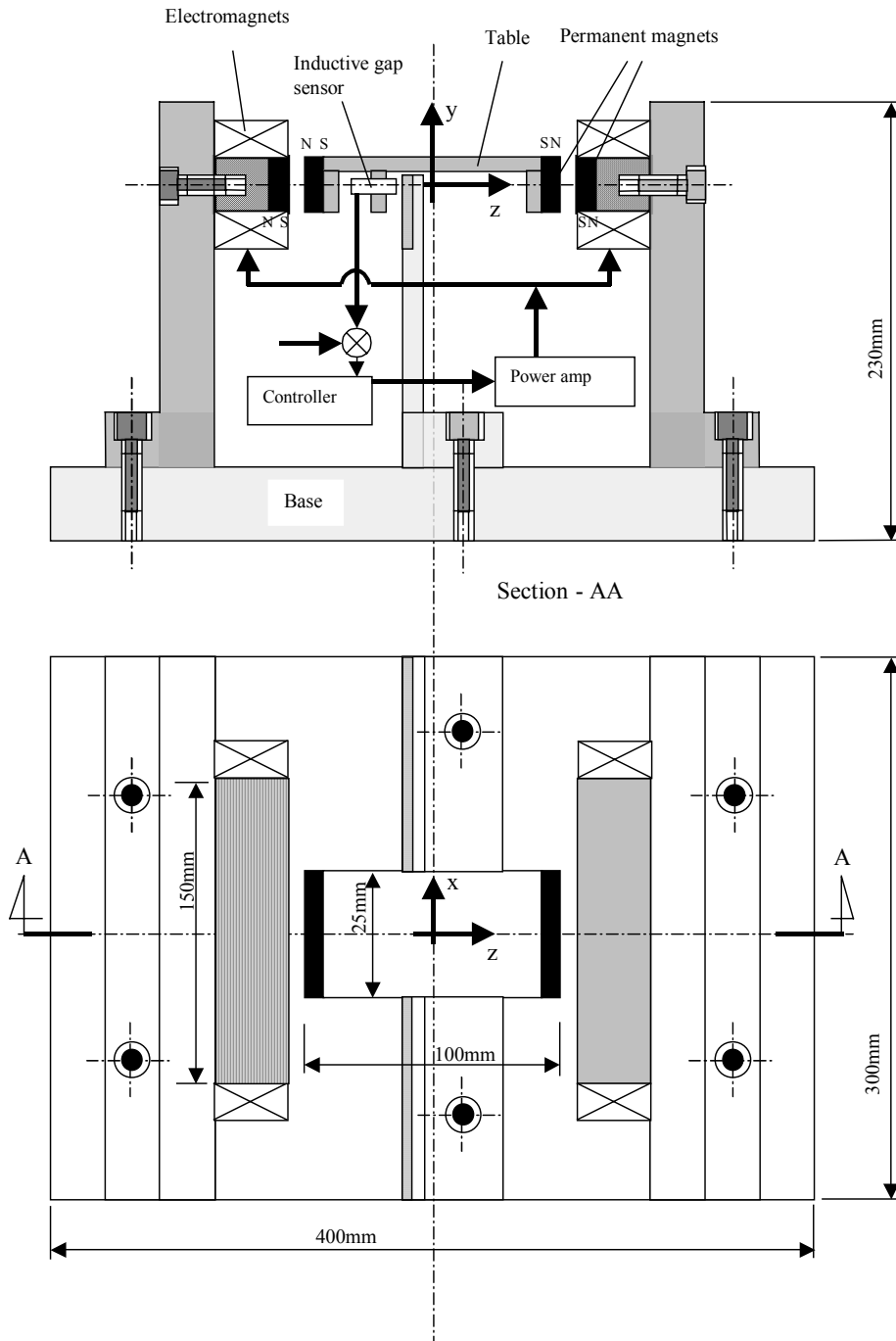
Figure 1. Configuration of the proposed magnetic linear bearing

$$K_{\phi} = l \cdot \left(\frac{l}{2} \cdot K_y + F_z \right) \tag{1}$$

$$K_{\theta} = l \cdot \left(\frac{l}{2} \cdot K_x + F_z \right) \tag{2}.$$

Equations (1) and (2) show that the magnetic stiffness K_{ϕ} and K_{θ} are proportional to magnets distance l, the stiffness K_x in x direction, the stiffness K_y in y direction and the magnetic force F_z in z direction. Equations (1) and (2) also show that high stiffness K_{ϕ} and K_{θ} will be achieved, in the magnetic bearing passive portion, by using, for example, permanent magnets with high magnetization and attached as far as possible from each other. This minimum length of the bearing table is described as follow.
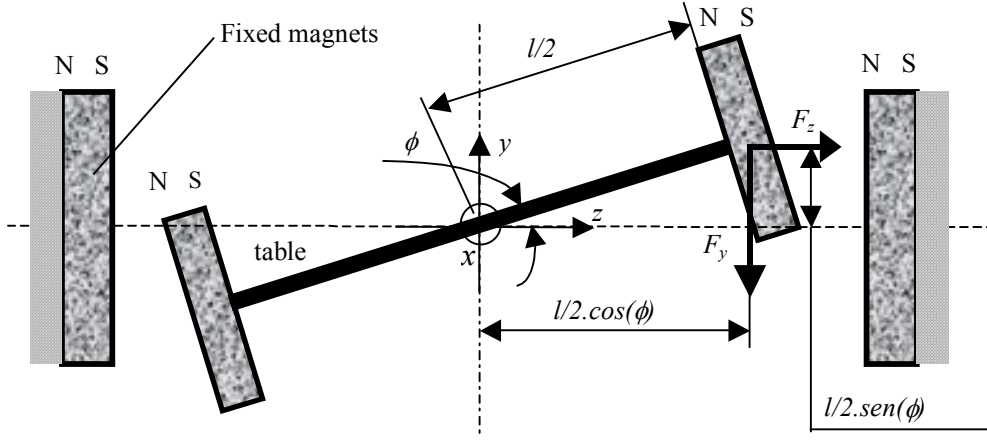
Figure 2. Table inclined by an angle ϕ around the *x*-axis, forces acting in the permanent magnets center

As shown later, the magnetic stiffness K_y is positive and, therefore, the bearing table has stable equilibrium in terms of translation along *y* direction. On the other hand, in the *z* direction, although the corresponding stiffness K_z is negative, Fig. (8b), the stability is assured by the control loop. However, the stability in terms of the table rotations, ϕ and θ, around the *x* and *y*-axis respectively, must be analyzed as follow.

Consider the configuration of Fig. (3). The distance between the permanent magnets is *l* and its width is *b*. In front of each magnet there is another one working in attraction mode. Considering the pairs of magnets at the right side of Fig. (3), when the table is tilted by an angle ϕ around the *x*-axis, as shown in the figure, the gap amount increases at the upper side and decreases at the lower side. Thus, the magnetic attraction force F_z at the lower side becomes larger than the corresponding magnetic F_z force at the upper side. The opposite occurs with the pair of magnets at the left side of the table. However, the tilting of the table also results in Δ_y displacement of the magnet attached to the table, relative to the magnets fixed to the base. These displacements generate a magnetic force F_y in the opposite direction of the displacements. This will force the table back to its original position before tilting. Figure (3) depicts a tilting around a generic point *0* located at the center of the table. However, the table is stable in terms of translation along the *y*-axis direction, as mentioned before and, the momentum do not depends of the pole *0* position on the *z*-axis (Beer.J, 1991). In such conditions and observing Fig. (3), the following are obtained:

$$M(\phi) = -F_z(l\,\mathrm{sen}(\phi) - b\cos(\phi)) + F_y(l\cos(\phi) + b\,\mathrm{sen}(\phi)) \tag{3}$$

Now, differentiating Eq. (3) in relation to ϕ angle, considering small enough angular tilting ϕ and working with Delamare´s equations, results:

$$K_\phi = K_z \frac{b^2}{2} - lF_z + K_y \frac{l^2}{2} + bF_y \tag{4}$$

The necessary and sufficient condition for the stable equilibrium of the bearing table around the *x*-axis is that the angular stiffness K_ϕ must be positive, that is:

$$K_\phi > 0 \tag{5}$$

Under this condition, Eq. (4) becomes:

$$\left(K_z \frac{b^2}{2} - lF_z + K_y \frac{l^2}{2} + bF_y \right) > 0 \tag{6}$$

The Earnshaw´s principle, for a pair of block type permanent magnets, shows that:

$$K_y + K_z = 0 \tag{7}$$

However, the angular tilting ϕ is assumed be enough small, so that:

$$F_y(\phi \to 0) = 0 \tag{8}$$

By Fig. (3), one can get:

$$z(\phi \to 0) = \frac{l}{2} \tag{9}$$

$$F_z = K_z.z \tag{10}$$

Now, substituting Eqs. (7) ~ (10) into Eq. (6), results:

$$\left(l^2 - \frac{b^2}{2}\right) > 0 \tag{11}$$

Now, solving Eq. (11) for l, one can reach the following relation that assures stable equilibrium of the bearing table for tilting around the x-axis.

$$l > \frac{b.\sqrt{2.}}{2} \tag{12}$$

Now, Fig. (4) depicts the table inclination around y-axis. Comparing Figs. (3) and (4), and considering that the magnets length is a, the condition to assure stable equilibrium for inclination around the y-axis is obtained in a similar way Eq. (12) was obtained, resulting in:
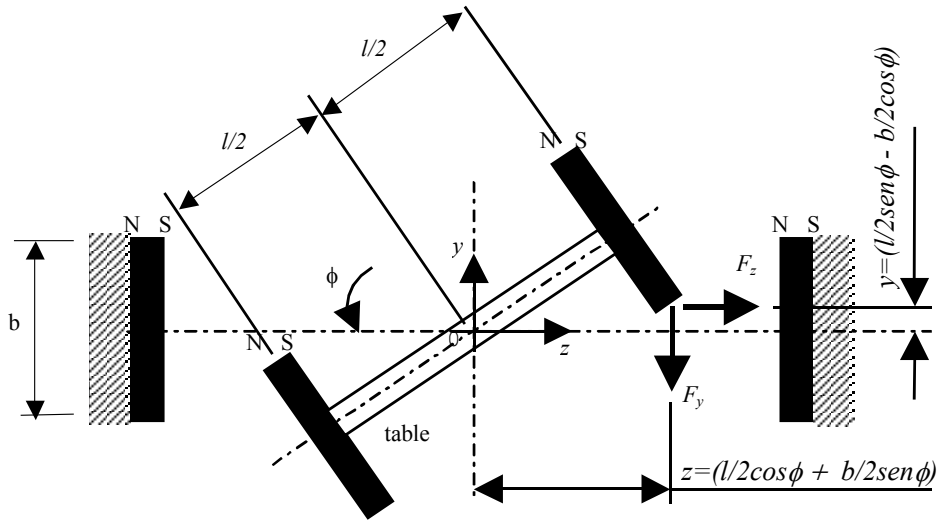
$$l > \frac{a.\sqrt{2}}{2} \tag{13}$$

Figure 3. Table inclined by an angle ϕ around x-axis

4. System modeling and control

The control system considering the table z position is modeled assuming some simplifications: (a) the table remains symmetric around the x, y and z-axis, (b) the displacements are small and occur around the equilibrium position and (c) the magnetic attraction force and the electromagnetic force in the z direction can be linearized with respect to the nominal operating point of equilibrium (i_o , z_o). The dynamic model of the proposed magnetic linear bearing is presented in Fig. (5). As shown in Fig. (5) the same current goes through the electromagnet coils, so the electromagnetic attraction and repulsion forces are obtained in each actuator side by mounting the coils with reversed polarity. The magnetic force f_m and the electromagnetic one f_{em} were linearized with respect to the displacement z and to the electric current i as follow:
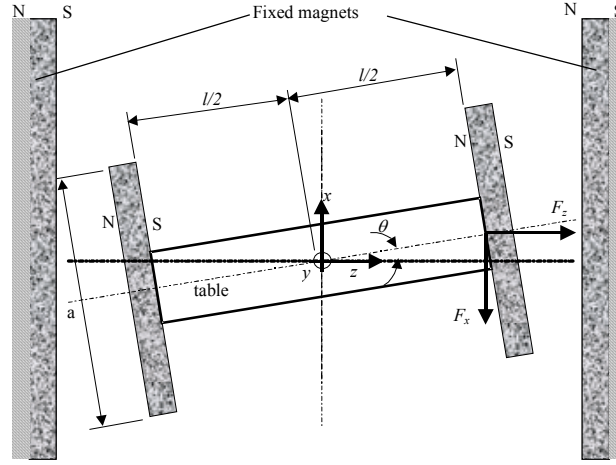
Figure 4. Table inclined by an angle θ around y-axis

$$f_m(t) = k_h z(t) \tag{14}$$

$$f_{em}(t) = k_t i(t) \tag{15}.$$

Where, t is the time k_h and k_t are respectively the magnetic and the electromagnetic constants. In Eq. (15) the electromagnetic constant k_t is function of the nominal displacement z_o. On the other hand, considering the use of electromagnets with inductance L and resistance R, the dynamic of the electromagnetic coil is given by the following equation:

$$L\frac{di}{dt} + Ri(t) = v(t) \tag{16}$$

Using Eqs. (14) ~ (16), the open loop transfer function of the system G(s) is obtained

$$G(s) = \frac{X(s)}{V(s)} = \frac{2k_t}{LM} \cdot \frac{1}{s^3 + \dfrac{R}{L}s^2 + \dfrac{k_h}{M}s + \dfrac{Rk_h}{LM}} \tag{17}$$

Here, z and M are respectively, the gap deviation from nominal operation point (table z direction) and the mass of the table, including the magnets.

In this system, only one gap sensor has been used, and the measured variable was the table z-axis direction. The system described by Eq. (17) is stabilized by a proportional – integral – derivative (PID) – type controller given by:

$$G_c(s) = \left(k_P + \frac{1}{T_i s} + \frac{T_d s}{\tau s + 1} \right) \tag{18}$$

Here, k_p is the proportional gain of the controller, and the T_i and T_d are, respectively, the time constant of the integral and derivative element. Each controller term has a specific effect on the bearing behavior, i.e., the proportional term (P element) makes the current change proportional to the size of the error signal. This results in a bearing behavior analogous to a spring. The force returning the shaft to the center position in the bearings increases proportionally with the error signal of the table z position. So, a bearing system with simple proportional control will oscillate and become unstable because there is no mechanism to provide damping to the system. The required damping can be introduced using derivative term (D element). The derivative control produces a force that is proportional to the velocity of the table z position. This results in the bearing removing energy from the table. Actually, the derivative term works as a shock absorber. However, a controller with only proportional and derivative terms will always have a steady state offset from the set point. This is due to the fact that the proportional and derivative controllers only deliver a non-zero output if there is a position error. In order to solve this position error an integrator term (I element) is used. The integrator produces output, which increases at a rate proportional to the size of the error signal over time. Thus, the longer the error signal is non-zero, the larger the integrator output. The integrator will hold its output constant if no error signal is present. For steady state conditions, the integrator increases the output until the error signal is zero (Aström and Hägglund, 1988). The block diagram of the control system for the z-axis direction of the bearing table is shown in Fig.(6).
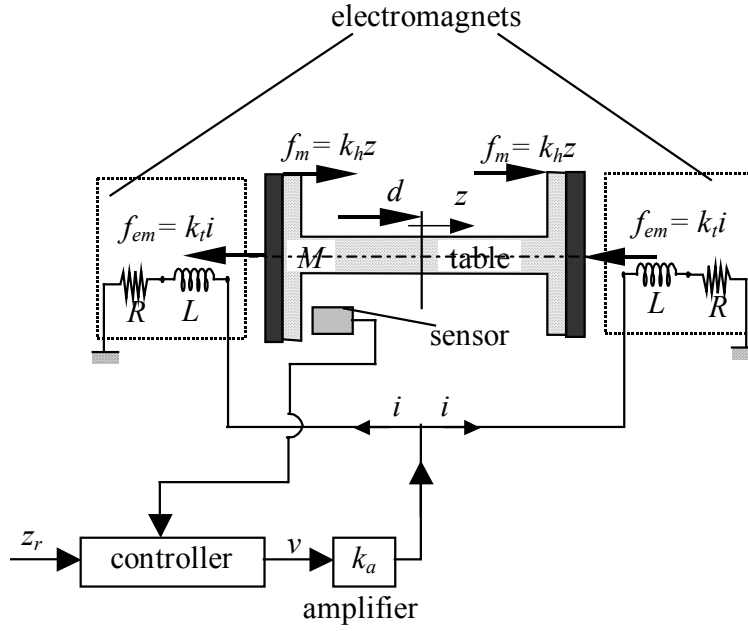
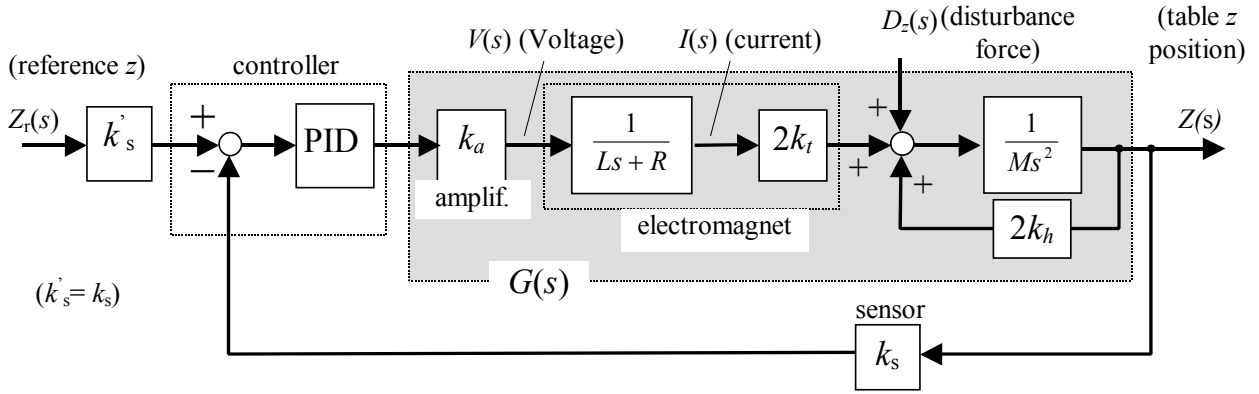Figure 5. Magnetic linear bearing dynamic model

Figure 6. Magnetic bearing control system block diagram

5. Results of experiments

As the first step in the development of the bearing, two block type permanent magnets, made of FeBa (J=0.2T, magnetization in the z-axis direction, and dimensions $a_1 = a_2 = 10mm$, $b_1 = b_2 = 25mm$, $l_1 = 100mm$ and $l_2 = 25mm$ as shown in Fig. 7) was selected because of construction convenience. Using these magnets, the magnetic forces F_y e F_z (x and y, according to Fig. 7) were measured experimentally and simulated according equations developed by Akoun and Yonnet, 1984. Results are shown in Fig. (8a) and (8b). Figure (8a) shows that the magnetic stiffness K_y is positive, so this is the reason magnetic bearing table is stable in y direction. This figure also shows that the K_y stiffness intensity, for a nominal operation point of $1,5x10^{-3}$ m, is approximately 400N/m. By Fig. (8b) one can observe that the corresponding stiffness K_z is negative and this is the reason because magnetic bearing table is unstable in z axis direction. This figure also shows that the K_z stiffness intensity is approximately −875 N/m.

Electromagnets used in the prototype have core dimensions of 10x35x100 mm and coils with 850 turns each. It's electromagnetic force constant was experimentally measured and a constant value of 0,8 N/A was obtained for a nominal gap of $1,5x10^{-3}$ m.

Using the PID controller mentioned above, some experiments were conducted on the prototype shown in Fig. (9). Parameters of the bearing were measured experimentally and parameters of the controller defined by simulations based on Eqs. (17) and (18). All parameters and their respective values are listed in Tab. (1).

Figure (10) shows the position of the table in z direction, with no disturbance being applied to the table and without moving it along x direction. A continuous vibration of about 7μm maximum displacement can be observed. This vibration was mainly caused by electrical noises from the sensor, from the electromagnetic actuators and from the

Table 1. System parameters

System	Symbol	Value
Magnetic constant (N/m)	k_h	4×10^3
Electromagnetic constant (N/A)	k_t	0.8
Mass of table (Kg)	M	0.25
Sensor gain (V/m)	$k_s = k_s$'	5×10^3
Amplifier gain	k_a	10
Nominal gap (m)	z_0	8×10^{-4}
Inductance (H)	L	$12.5\times10^-$
Resistance (Ω)	R	40

Controller	Value
k_P	100
T_i (s)	2×10^{-3}
T_d (s)	1.25

electronic circuits of the controller. Figure (11) shows the response to impulsive forces applied to the table in the z direction. The table displaces from its nominal operating position, due to the forces, but the original position is restored rapidly. This means that the bearing controller is capable of reject low frequencies disturbance signals applied to the table and thus, dumping vibrations caused by external forces.
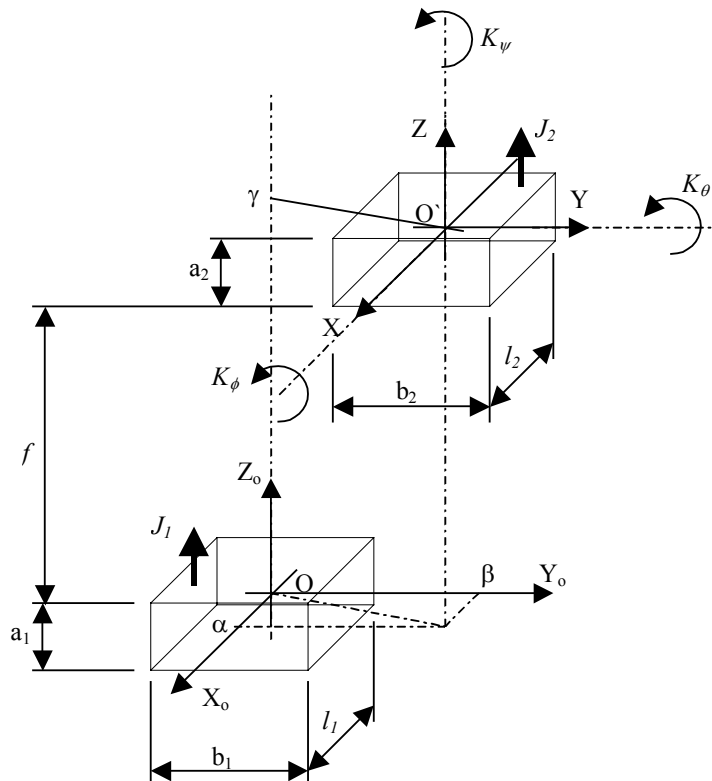

Figure 7. Two block type permanent magnets

The controller was also equipped with an entrance for a reference signal (reference z position). Figure (12) shows the response to a stepwise reference input of 6×10^{-5}m. In less than 0.2 s, the table reaches the commanded position and remains in this new position with a precise positioning of 10µm. This result shows a capability of this bearing to execute fast and precise positioning of the table. By this, the bearing can, for example, compensate for systematic motions errors of the table that occurs in the z direction, along its travel.

Figure (13) shows the experimental and theoretical stiffness K_y (Akoun and Yonnet, 1984), of the prototype in the y direction. The stiffness curve aspect shows a prominent nonlinear characteristic and increase as the shift β of the table

increases (Fig. 7). By this figure, the maximum stiffness K_y intensity reached in the prototype is approximately 1000 N/m.

The stiffnesses against table rotation around x, y and z-axis was not measured because of the necessity of developing specific measurement system, i.e., a system to apply a known moment to the table and measure its angular displacement. This will be an activity for future works. However, using Eq. (1) at least the intensity of the stiffness K_ϕ around the x-axis can be calculated. So, by Figs. (1), (8b) and (13) the experimental values for l, F_z and K_y can be obtained. So, putting these values into Eq. (1) the intensity of the stiffness K_ϕ is 4,27 Nm/rad.

Although these stiffnesses intensities are not sufficient for practical applications, they can be increased, for example, by using rare earth permanent magnets.
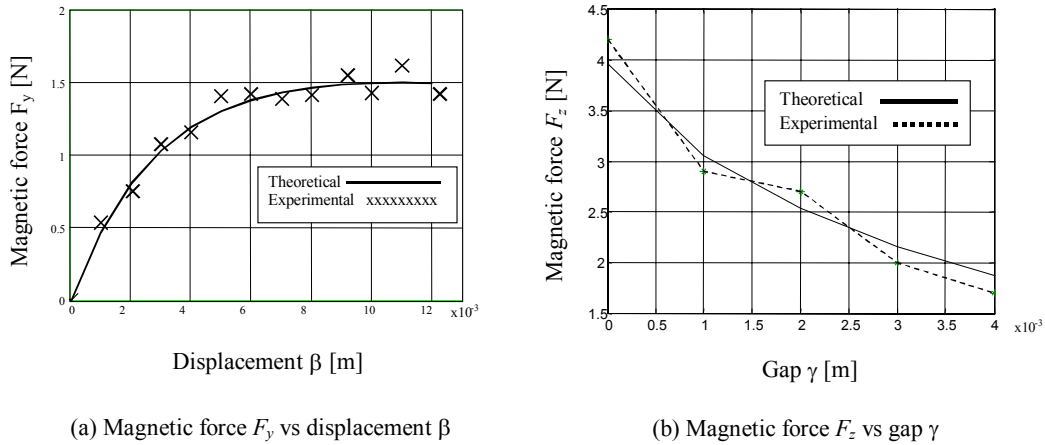

(a) Magnetic force F_y vs displacement β (b) Magnetic force F_z vs gap γ

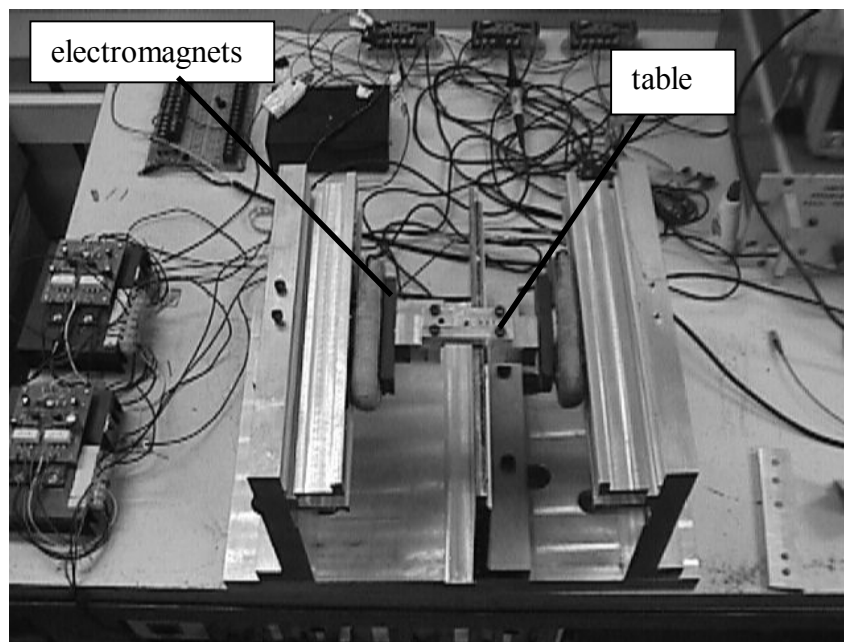Figure 8. Magnetic forces

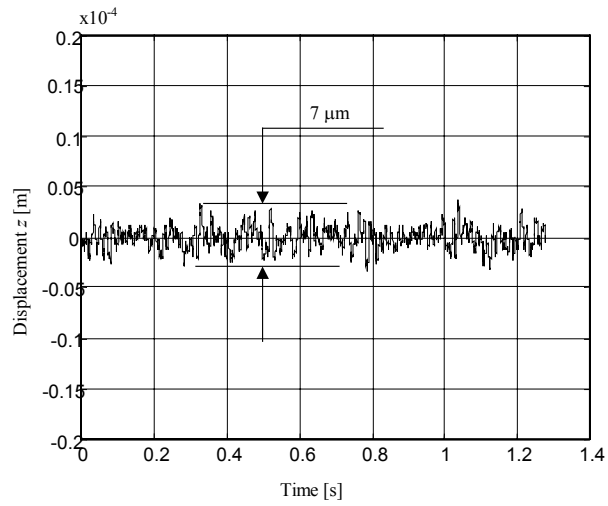Figure 9. Prototype of the linear magnetic bearing

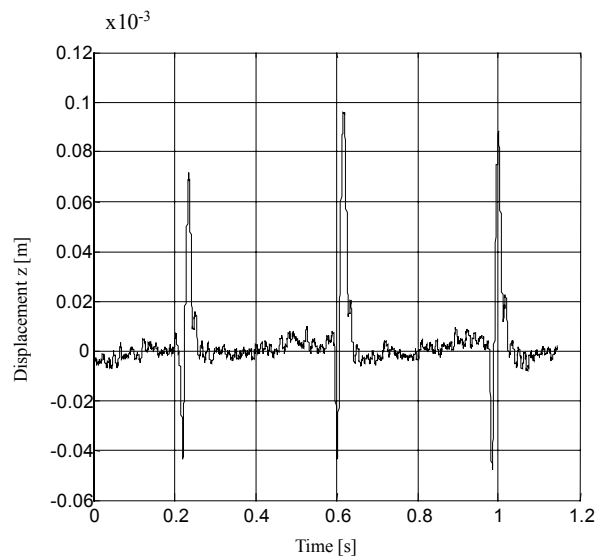Figure 10. Vibration of the table in *z*-axis direction
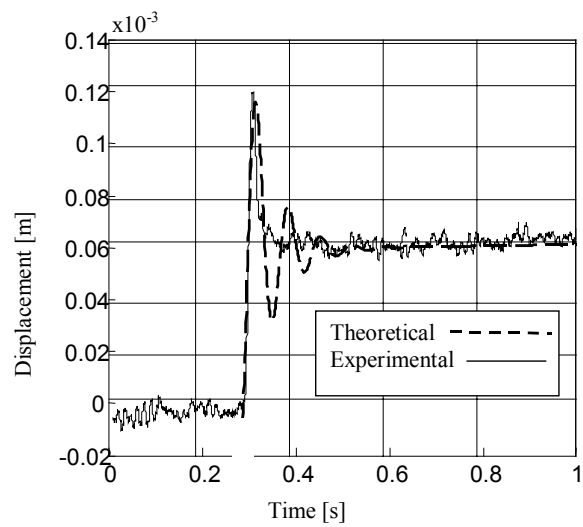

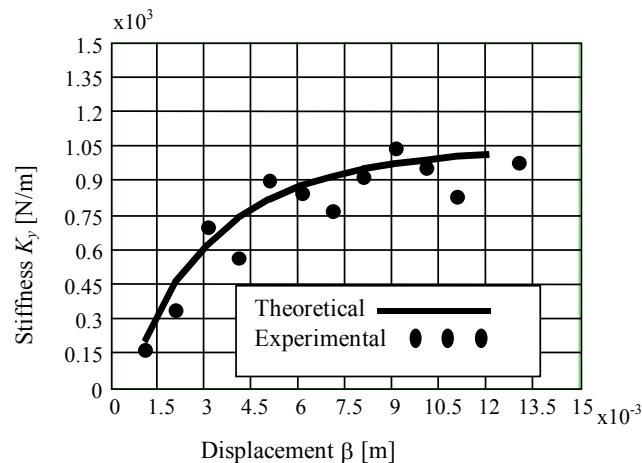
Figure 11. Impulse response

Figure 12. Step response

Figure 13. Bearing stiffness in *y*-axis direction

6. Conclusions

This paper has presented a novel concept of a magnetic linear bearing with active control in only 1.d.o.f of a magnetically levitated table, namely the motion in the direction orthogonal to the table travel direction. The remaining degrees of freedom are restricted only by the action of permanent magnets that works in attraction mode. The principle of this linear bearing was described, and the most relevant points to be considered in its design were described. Finally by experiments, it was shown that the bearing is capable of: 1) achieving a stable suspension of the table; 2) keeping the table in a fixed *z*-axis position with accuracy better than 7μm; 3) executing a fast and precise positioning of the bearing table. In the present work, all the experimental results were shown for the stationary bearing table i.e., without moving the table along the *x* direction. So, better adjust of controller parameters, improvement in the electromagnetic actuators characteristics, better arrangement of the bearing table architecture to improve the magnetic stiffness in *y* axis direction and also around *x*, *y* and *z* axis and experiments conducted moving the bearing table will be a theme for future works.

7. References

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