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SINGLE AXIS ACTIVE MAGNETIC BEARING SYSTEM WITH MECHANICAL DAMPERS FOR HIGH SPEED ROTOR

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Abstract

We realized a single axis active magnetic bearing system for high speed rotor(60,000rpm) by applying mechanical dampers. Test machine's rotor is $M = 0.76$ kg, $I_Z = 4.3 \times 10^{-4}$ kgm^2, and $I_X = I_Y = 1.8 \times 10^{-3}$ kgm^2. Naturally, this system is mounting position free. Though we have well known eddy current damper and oil damper, the mechanical damper that we developed is a simple and unique one. In this paper, we show that a single axis active magnetic bearing system with the mechanical dampers has good features if the system parameters are selected systematically and carefully. Furthermore, the dynamic behavior of rotor suspended by this magnetic bearing system is analyzed and test result of this system is shown.

1. Introduction

Though magnetic bearings are known from forty years ago, very limited number of products have been developed so far. The majority of application is of special order products used in scientific application or space development[1]. In recent years, the magnetic bearing for wafer transfer-machine is developed in semiconductor manufacturing industry which are also of special order products[2]. The turbo molecular pump is only one exception that succeeded in mass production[3]. However,its share in the entire market is as low as approximately 10 %.

It is true that magnetic bearings have many distinctive features which are not attained by ordinary ball bearing. However, practical application of these bearings is greatly hindered by the fact that production cost is very high and that size of the circuit and structure is too big.

The most important matter to make these magnetic bearings popular is [How to design small size and low cost magnetic bearings without loosing original features of those].

Currently, we developed a single axis active magnetic bearing which satisfied the purpose said above. This is equipped with a simple and unique mechanical damper made of viscoelastic material, (though we have well known eddy current damper and oil damper[4][5].)

The following four features are distinctive:

1) It has enough stability from 0rpm till ultra high speed rotation(60,000rpm) like five axes active magnetic bearing.
2) It has sufficient spring constant in every direction for mounting position free.
3) Power consumption of the magnetic bearing is very small.
4) It is small sized and low cost magnetic bearing.

This system seems best suited for such an application in which a rotor is simply turned at high speed such as a chopper, pump, and compressor.

2. Character

The character of this magnetic bearing system is shown below.

1) **Character of rotor**

Mass of rotor 0.76 Kg

Moment of inertia of rotor
(Z is rotational axis) $I_z = 4.3 \times 10^{-4}$ Kgm^2
$I_x = I_y = 1.8 \times 10^{-3}$ Kgm^2

Rotational speed 60,000rpm

2) **Charactor of magnetic bearing**

Structure
Two passive radial magnetic bearings with mechanical damper and One active axial magnetic bearing

Steady current of Axial magnets 0～0.2 A

Resonance points
Toransrational mode 50 Hz
Tilting mode 57 Hz

3) **Motor**
High frequency induction motor Max. 150 W
Run-up time (60,000rpm) 2 min.

4) **Size**

Structure φ 100×180mm
(Circuit-box 250×130×430mm)

3. Entire construction

Single axis active magnetic bearing system with mechanical dampers is shown fig.1. The rotor is supported by passive radial magnetic bearings A and B, and by active axial magnetic bearing C. Acceleration up to 60,000 rpm is driven by means of a high frequency induction motor D.

With passive magnetic bearing A, three permanent magnets are provided at both rotor side and stator side on the concentric circle, and they are repulsing each other. This sort of magnetic bearing is stable in radial direction and is unstable in axial direction. Three permanent magnets are magnetized in axial direction and are laminated in the order of NS, SN, NS.

The stator which holds permanent magnets of passive magnetic bearing A acts as mechanical damper X and moves freely in radial direction only. With this configuration, a viscoelastic material 1 is put between main stator and movable stator 2. When the rotor is vibrated in radial direction, mechanical damper X is vibrated in radial direction, and its energy is absorbed by viscoelastic material 1 to damp vibration of the rotor.

Passive magnetic bearing B is basically identical with magnetic bearing A. One of differences is such that stator portion is divided into two segments of fixed part and moving damper part Y. Rotor side holds three permanent magnets, fixed part of stater side holds two and moving damper part Y holds one. Two permanent magnets are provided on both rotor side and fixed part on the concentric circle so that they may repulse each other with a similar manner as observed with magnetic bearing A. This sort of bearing is stable in radial direction and unstable in axial direction. Permanent magnets remained at rotor side and at moving damper part Y absorb each other in axial direction. This moving damper Y is of mechanical damper and can move in radial direction only. When the rotor is vibrated in radial direction, it is vibrated according to permanent magnet force, and its energy is absorbed by viscoelastic material 3 put between main stator and movable stator 4 with a similar manner as observed with mechanical damper A.

Active axial magnetic bearing C consists of two axial electromagnets 5 of similar configuration, an armature disk 6, an axial position sensor 7, and an electromagnetic control circuit(not shown fig.1). As for axial direction damping, amplitude and phase of electromagnetic force is being controlled.

Since bias magnetic flux is applied to the axial direction electromagnet by the permanent magnet, little electric current should be flowed through the electromagnetic coils in steady state.

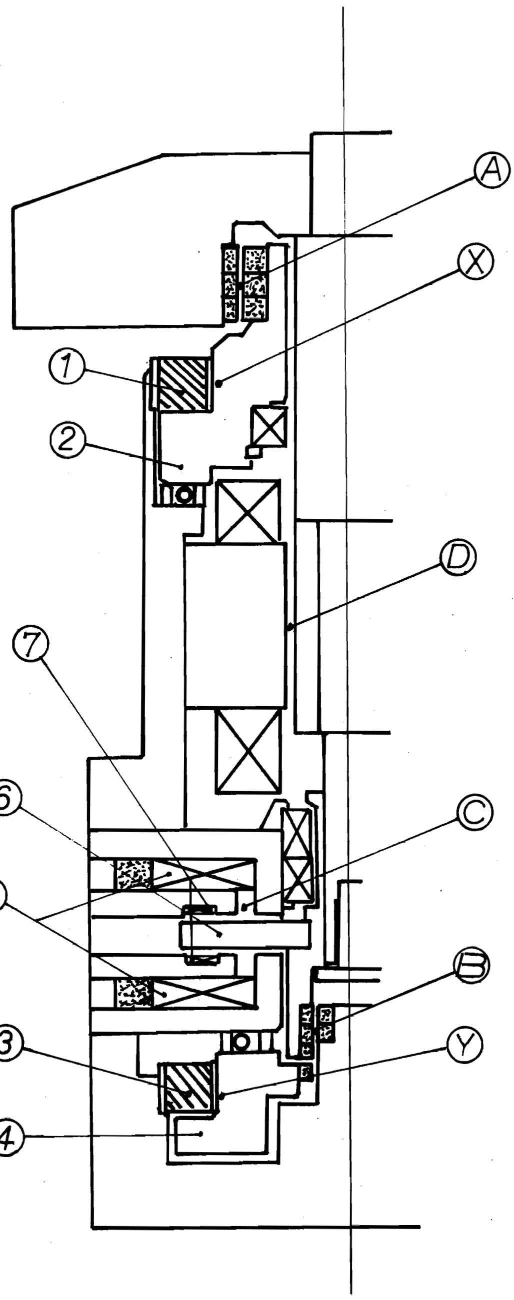

Fig.1 Entire construction of single axis active magnetic bearing system with mechanical dampers

4. Modeling and numerical calculation

4.1 Radial equations of motion

It is discussed only radial motions because main theme in this paper is passive radial magnetic bearings with mechanical dampers.

A model shown which is considered in this investigation is shown fig.2. Principal axes of coordinates are X, Y, Z, θ_X, θ_Y, and θ_Z. Coordinates at points A and B are X_a, Y_a, X_b, and Y_b. Definition of each parameter is shown in Table 1.

The following assumptions are made.

1) Displacement in θ_X and θ_Y direction is smaller than L_a and L_b respectively.
2) Movement of the damper is limited to a plane perpendicular to the axis of rotation.
3) Gravity acts in the direction of -X.
4) The rotor is rigid.

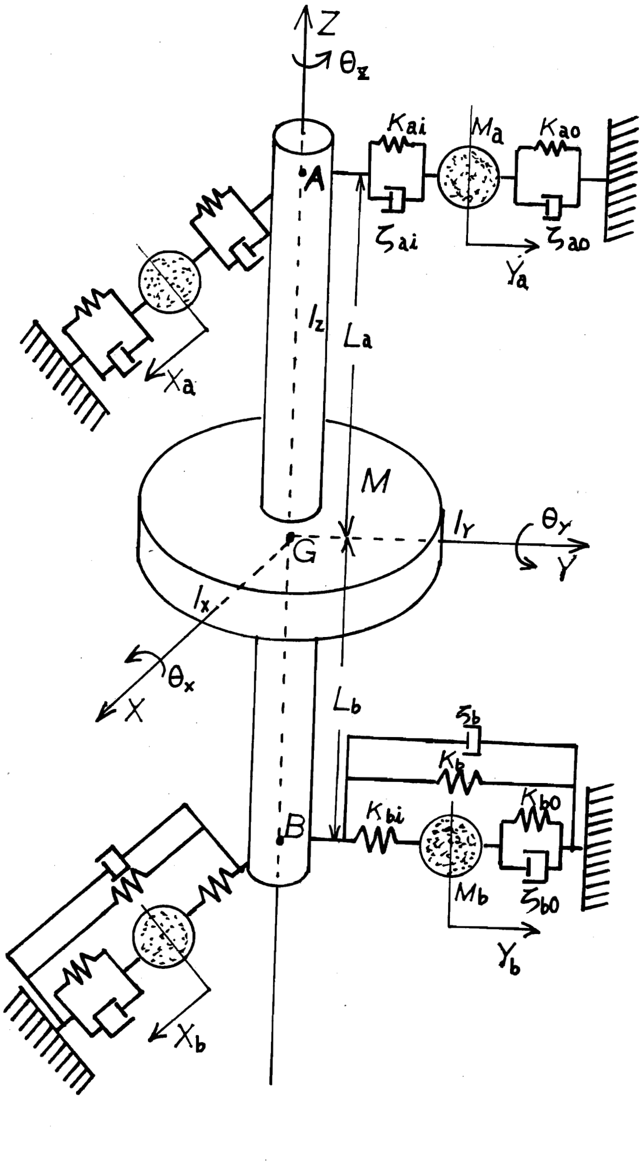

Fig.2 A model of passive radial magnetic bearings with mechanical dampers

Symbol	Definition
M	Mass of rotor
I_z	Moment of inertia of rotor (z, x, and y direction)
I_x I_y	
L_a	Distance between center of gravity and point A or B
L_b	
K_{ai}	Spring constant at point A i:Inside o:Outside
K_{ao}	
ζ_{ai}	Damping coefficient at point A i:Inside o:Outside
ζ_{ao}	
M_a	Mass of damper A
K_b	Spring constant at point B :Directly coupled with stator i:Inside o:Outside
K_{bi}	
K_{bo}	
ζ_b	Damping coefficient at point B :Directly coupled with stator o:Outside
ζ_{bo}	
M_b	Mass of damper B
ω	Rotor angular speed
g	Acceleration of gravity

Table 1 Definition of parameters

Radial equations of motion are as follows.

(1)

$$MX'' = -Mg - X(K_{ai}+K_b+K_{bi}) - \theta_y(L_aK_{ai} - L_b(K_b+K_{bi})) + X_aK_{ai} + X_bK_{bi} + 2\zeta_{ai}(K_{ai}ML_b/(L_a+L_b))^{1/2}(-X' - \theta_y' L_a + X_a') + 2\zeta_{bi}(K_bML_a/(L_a+L_b))^{1/2}(-X' + \theta_y' L_b)$$

(2)

$$MY'' = -Y(K_{ai}+K_b+K_{bi}) + \theta_x(L_aK_{ai} - L_b(K_b+K_{bi})) + Y_aK_{ai} + Y_bK_{bi} + 2\zeta_{ai}(K_{ai}ML_b/(L_a+L_b))^{1/2}(-Y' + \theta_x' L_a + Y_a') + 2\zeta_{bi}(K_bML_a/(L_a+L_b))^{1/2}(-Y' - \theta_x' L_b)$$

(3)

$$\begin{aligned} I_y\theta_y'' = & -X(L_aK_{ai} - L_b(K_b+K_{bi})) - \theta_y \\ & (L_a^2K_{ai} + L_b^2(K_b+K_{bi})) \\ & + X_aL_aK_{ai} - X_bL_bK_{bi} + I_z\omega\theta_x' \\ & + 2\zeta_{ai}(K_{ai}ML_b/(L_a+L_b))^{1/2} \\ & (-X' - \theta_y'L_a + X_a')L_a - 2\zeta_{bi} \\ & (K_bML_a/(L_a+L_b))^{1/2}(-X' \\ & + \theta_y'L_b)L_b \end{aligned}$$

(4)

$$\begin{aligned} I_x\theta_x'' = & -Y(L_aK_{ai} - L_b(K_b+K_{bi})) - \theta_x \\ & (L_a^2K_{ai} + L_b^2(K_b+K_{bi})) \\ & - Y_aL_aK_{ai} + Y_bL_bK_{bi} - I_z\omega\theta_y' \\ & - 2\zeta_{ai}(K_{ai}ML_b/(L_a+L_b))^{1/2} \\ & (-Y' + \theta_x'L_a + Y_a')L_a + 2\zeta_{bi} \\ & (K_bML_a/(L_a+L_b))^{1/2}(-Y' \\ & + \theta_x'L_b)L_b \end{aligned}$$

(5)

$$\begin{aligned} M_aX_a'' = & -M_ag - X_a(K_{ao}+K_{ai}) - X_a'2\zeta_{ao} \\ & (M_aK_{ao})^{1/2} + \theta_yL_aK_{ai} + XK_{ai} - 2\zeta_{ai} \\ & (K_{ai}ML_b/(L_a+L_b))^{1/2}(-X' \\ & - \theta_y'L_a + X_a') \end{aligned}$$

(6)

$$\begin{aligned} M_aY_a'' = & -Y_a(K_{ao}+K_{ai}) - Y_a'2\zeta_{ao} \\ & (M_aK_{ao})^{1/2} - \theta_xL_aK_{ai} + YK_{ai} - 2\zeta_{ai} \\ & (K_{ai}ML_b/(L_a+L_b))^{1/2}(-Y' \\ & + \theta_x'L_a + Y_a') \end{aligned}$$

(7)

$$\begin{aligned} M_bX_b'' = & -M_bg - X_b(K_{bo}+K_{bi}) - X_b'2\zeta_{bo} \\ & (M_bK_{bo})^{1/2} - \theta_yL_bK_{bi} + XK_{bi} \end{aligned}$$

(8)

$$\begin{aligned} M_bY_b'' = & -Y_b(K_{bo}+K_{bi}) - Y_b'2\zeta_{bo} \\ & (M_bK_{bo})^{1/2} + \theta_xL_bK_{bi} + YK_{bi} \end{aligned}$$

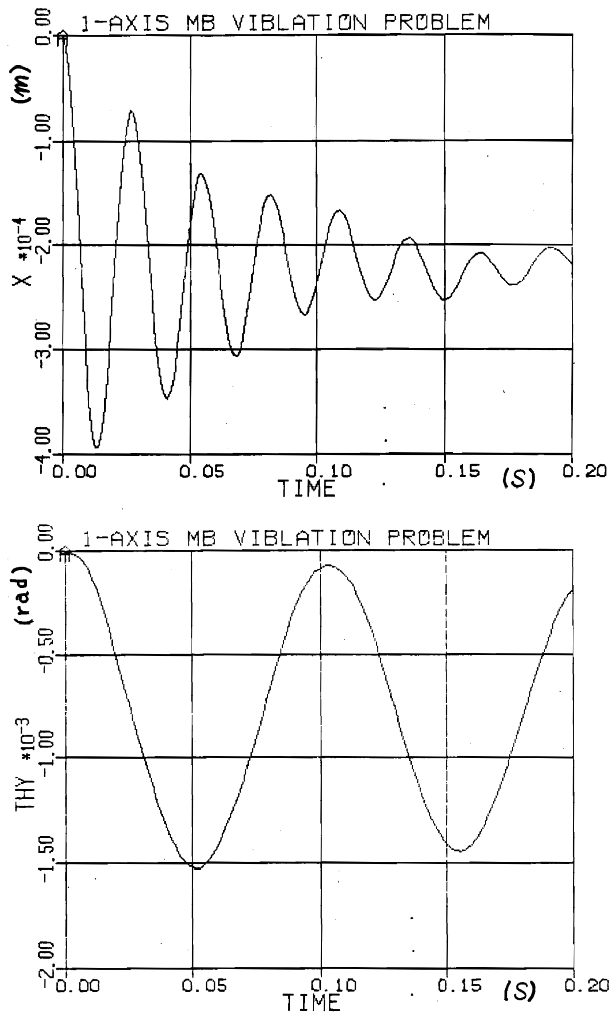

Fig.3 Rotor's motions of X and θ_Y direction at $\zeta_{ao}=0.14$, $\zeta_{bo}=0.14$, $\zeta_{ai}=0.01$, $\zeta_b=0.01$, and $\omega=2\pi\times10^3$ rad/s

4.2 Numerical calculation

Dynamic characteristics of the rotor are studied by numerical calculation using equations of motion obtained in 4.1.

The result from numerical calculations is shown below.

1) If spring constants of force supporting rotor are determined by maintaining the relation of $L_b/L_a = K_{ai}/(K_b + K_{bi})$, spring constants of force supporting rotor are nearly equal to those of force supporting damper, and mass of damper is nearly equal to effective mass of rotor respectively, damping capacity of the system becomes optimum.
2) The limit of rotational speed depends on the amount of decrease in resonance frequency and damping capacity of tilting mode due to gyro effect when rotational speed increases.

An example of caliculation is shown in fig.3.

5. Test result

In order to confirm the effect of mechanical damper, we compared real rotor's motion with the result of numerical calculation by using the parameters that were measured. (Only spring constants of the force of permanent magnets are designed values.[6])

As is shown in fig. 4a and 4b, similarity between two viblation-forms is pretty good.

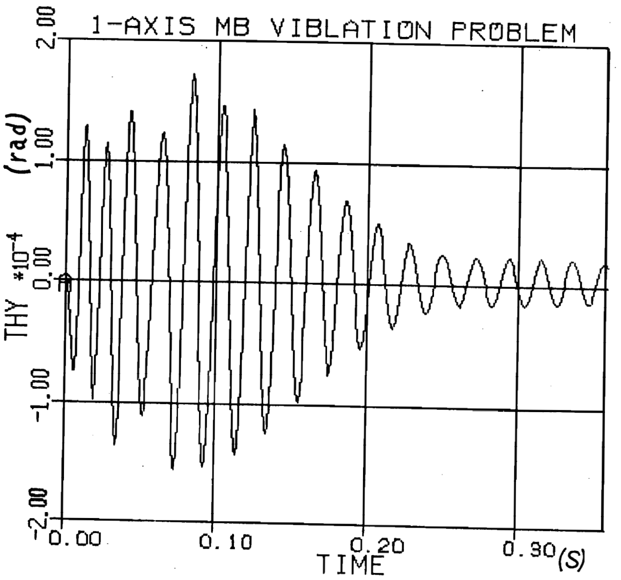

Fig.4a Rotor's motion of θ_y direction
in calculation
at $\zeta_{ao}=0.11$, $\zeta_{bo}=0.11$, $\zeta_{ai}=0.01$, $\zeta_b=0.01$, $K_{ai}=6.5\times10^4$ N/m, $K_b=K_{bi}=1.2\times10^4$ N/m, $K_{ao}=2.2\times10^4$ N/m, $K_{bo}=7.2\times10^4$ N/m and $\omega=0$

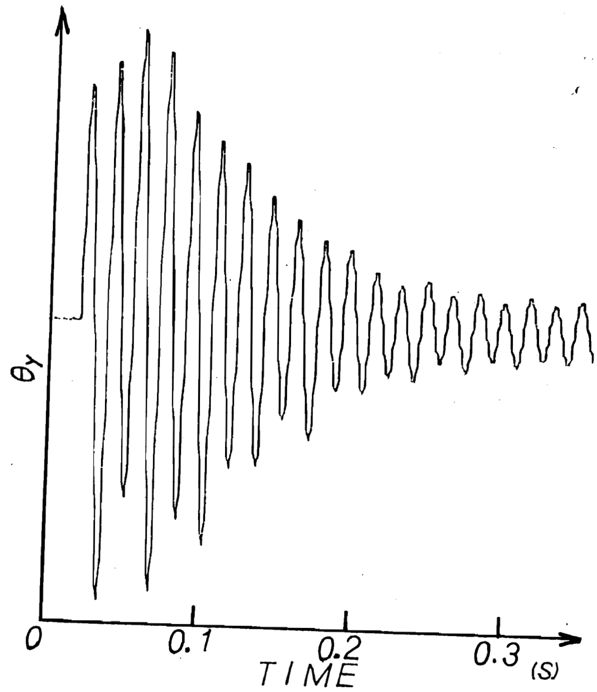

Fig.4b Rotor's motion of θ_y direction
in measurement

6. Conclusions

It has become clear that we can design a mechanical damper optimumly by using the model mentioned above and selecting the parameters systematically and carefully, though most of people have considered tuning in mechanical damper is very difficult. And we think this method is one of the answers for designing small size and low cost magnetic bearing without loosing original features of it.

7. References

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