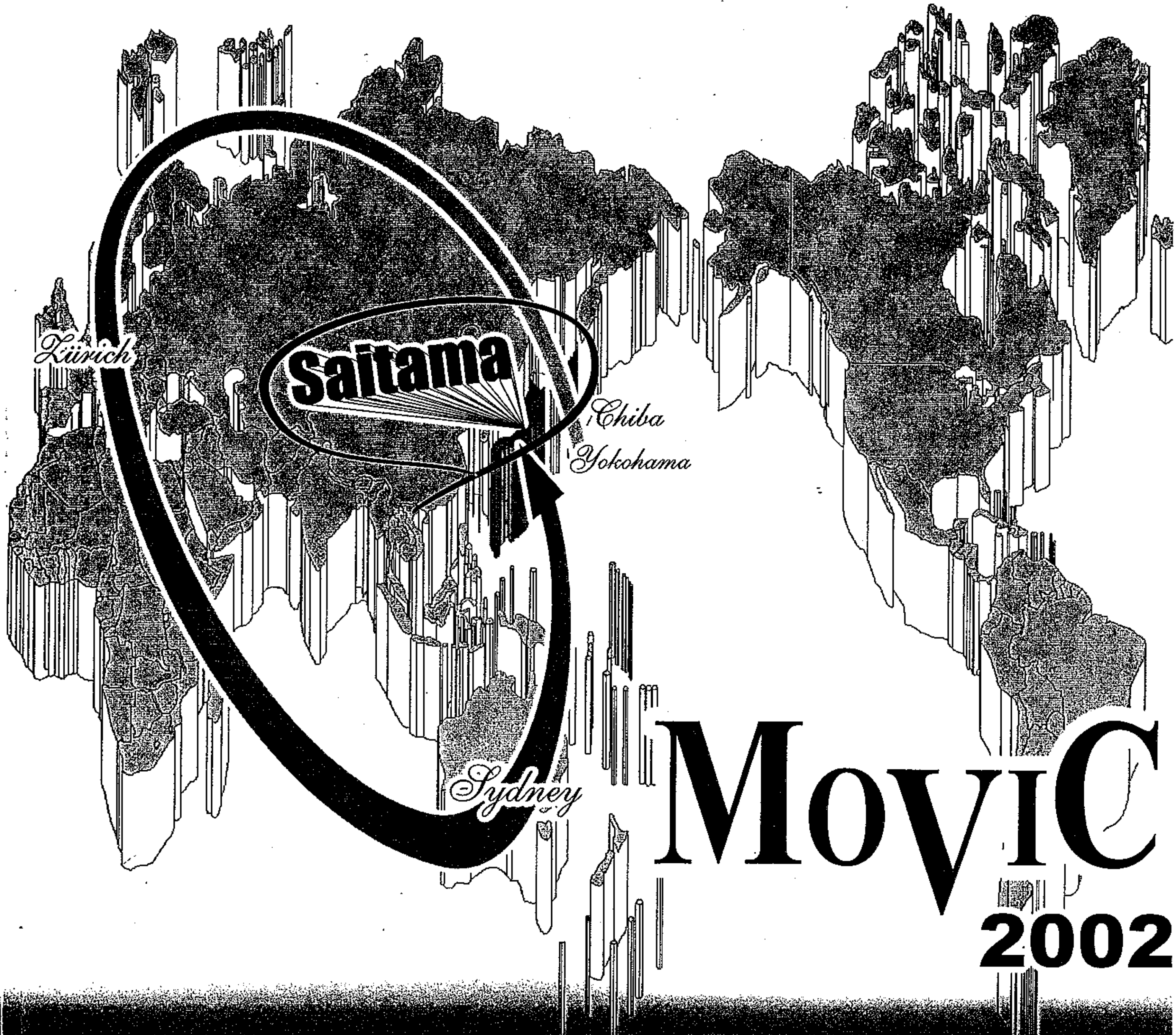
No.02-201

The Sixth International Conference on

Motion and Vibration Control

August 19-23, 2002
Saitama, Japan



MoViC 2002

Proceedings Volume 2

Edited by
Takeshi MIZUNO and Yoshihiro SUDA

The Dynamics, Measurement and Control Division of
Japan Society of Mechanical Engineers

The 6th International Conference on Motion and Vibration Control, Saitama, August 19-23, 2002

VIBRATION CONTROL WITH LINEAR ACTUATED PERMANENT MAGNET

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ABSTRACT

This paper describes a new vibration control method. The distinctive feature of this method is the use of linear actuators and permanent magnets. Linear actuators actuate the magnets and control attractive forces which are used for reducing vibration through adjustments of the air gap between magnets and an object. To examine the performance of the proposed vibration control method, we consider a system modeled on an actual experimental device. Numerical simulations are carried out and the effectiveness of the proposed method is determined.

1. INTRODUCTION

In the process of plating, coating or rolling of steel sheets, vibration in conveyance often becomes a problem, as sheets are very flexible. As a countermeasure, a vibration suppresser with mechanical contacts is not suitable in such a process, Objects are easily damaged due to their material makeup such as iron plate which has just been rolled, coated, or plated. Therefore a noncontact suppression mechanism is more suitable for controlling the steel sheets. Problems such as deformation, peeling, and uniformless products are minimized. Noncontact vibration control methods which use attractive forces of electromagnets have already been proposed in many papers[1]-[5]. The principal weakness of these methods is that the control range is very constricted, because the attractive force of the magnet varies in inverse proportion to the square of air gap length. If the vibration amplitude of the object is large, it becomes impossible to control the object using electromagnets.

This paper proposes a vibration control method using permanent magnets and linear actuators. The key to the proposed method is the force control mechanism. A linear actuator drives a permanent magnet and varies the air gap between the magnet and the object. The variation in the size of the air gap changes the attractive force. Since the control range is almost the same as the actuator stroke, we can expect the vibration control range to be correspondingly wide.

In this paper, we study the feasibility of the proposed method. The outline of the proposed method is introduced and the aim of the system is shown. The principle of the control mechanism is explained, and an experimental system is introduced and modeled. Following that, the experimental system is analyzed according to linear control theory, and a controller is designed based on the results. Numerical simulations are carried out to demonstrate the properties of the control method and its feasibility.

2. BASIC CONCEPTION OF VIBRATION CONTROL FOR STEEL SHEET

A schematic illustration of a steel sheet plating or coating process is shown in Fig. 1. The steel sheet is fed from the right side of the figure and is directed upwards by a roller moving clockwise. While the steel sheet is being fed into the solution bath, plating or coating is carried out. After the plating process is completed in this way, the steel is seasoned or cooled in the vertical feed. In the seasoning process, the steel is especially sensitive to deformation. Consequently vibration control in the seasoning process is very important.

The aim of the proposed system is to reduce vibration caused by the roller feed mechanism in the plating

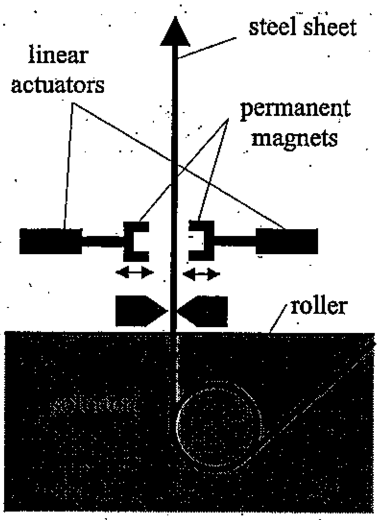

Fig.1 Illustration of Plating Process

process. Two permanent magnets and linear actuators located on opposing sides of the steel plate are used for vibration control. The magnets are actuated, by the actuator, in the horizontal direction. When the left magnet is positioned closer to the steel and the right magnet further away, a leftward force is generated. Similarly a rightward force can be generated. This force control mechanism has previously been proposed in magnetic levitation systems[6]

The strategy for controlling vibration in the steel sheet is as follows: A sensor measures the displacement of the steel sheet. Based on the sensor information, a controller calculates the force required to suppress the vibration. This force is created by driving the magnet and adjusting the air gap size. Thus the proposed vibration control system is realized.

3. EXPERIMENTAL SYSTEM

An experimental system to examine the performance of the proposed vibration control method was devised. This system was modeled in order to analyze the linear control theory and to synthesize the control system.

3.1 Experimental System

A photograph of an experimental system is shown in Fig. 2 The outline of the system is illustrated in Fig. 3. As the first step toward realization of the proposed method, an experimental design was created as shown in the Fig. 3.

The lever supported by two springs is the controlled object to suppress vibration. These springs can be replaced and an arbitrary stiffness can be determined.

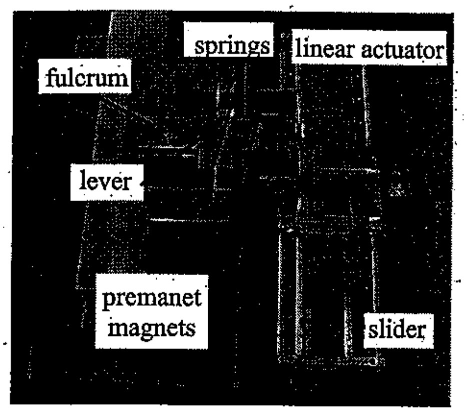

Fig.2 Photograph of Experimental System

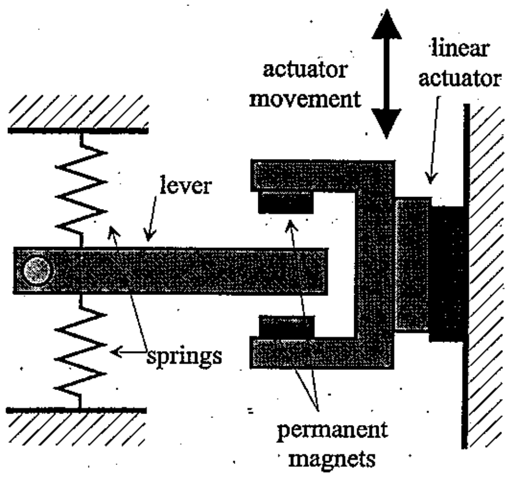

Fig.3 Illustration of Experimental System

The control force is created by two permanent magnets which are actuated by a linear actuator. Two iron plates are installed in the lever. They are positioned so that the face to the permanent magnets as attractive forces act on the lever. The linear actuator is a type of linear DD motor which is called a Megathrust motor produced by NSK. The actuator has a stroke length of 400 mm, a stroke speed of up to 1800 mm/s, and a precision of 0.001 mm.

3.2 MODELING OF SYSTEM

Modeling of the experimental system is needed in order to confirm stability, calculate the feedback gains, and permit a numerical simulation. In the model, the motion of the lever is assumed to be translational, even though its motion is rotational, The positive direction is the upward direction in Fig. 3

The symbols used in the model are: z_0 and z_1 are displacements of the lever and the actuator, d_0 is the air gap width when the lever is centered between the

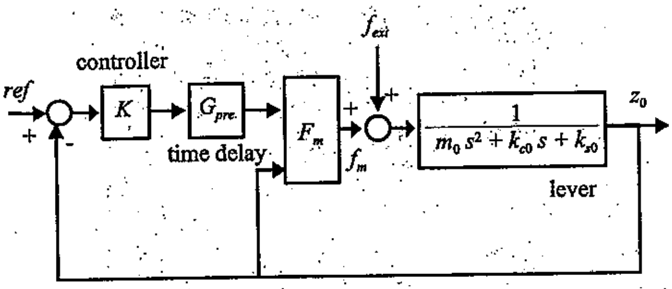

Fig.4: Block Diagram in Case System Input is Magnet Position

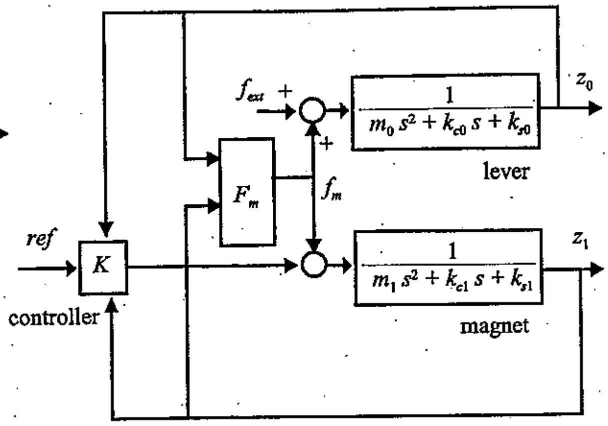

Fig.5: Block Diagram When System Input is Actuator Force

magnets, k_{s0} and k_{s1} are spring constants, k_{c0} and k_{c1} is damping coefficients, m_0 is the equivalent mass of the lever, m_1 is the mass of moving part together with the permanent magnet (sum of the magnet and the slider), f_m is the attractive force, and f_a is the force of the actuator. The attractive force is determined as follows:

$$f_m = \frac{k}{(d_0 - z_0 + z_1)^2} - \frac{k}{(d_0 + z_0 - z_1)^2} \tag{1}$$

where, k is the constant of the magnet. The equation for the motion of the lever is

$$m_0\ddot{z}_0 = fm - k_{s0}z_0 - k_{c0}\dot{z}_0 \tag{2}$$

The equation for the motion of the permanent magnet is

$$m_1\ddot{z}_1 = -fm - k_{s1}z_1k_{c1}\dot{z}_1 + f_a \tag{3}$$

The system model is analyzed for two cases: One is when the system input is defined by the position of the magnets, and the other is when the input is defined by the force of the actuator.

Position input system When the system input is defined by position of the magnets, the model is represented by using Eq. (1) and (2). The system block diagram is shown in Fig. 4. In Fig. 4, f_{ex} represents the external disturbance and is the source of the vibration. As the disturbance is a compulsive force caused by the feed of the steel into the original system, it may be added to the external force for the lever.

The block G_{pre} in the figure represents the delay of the actuator. Even if the position input system is considered, a delay from the input signal to actual change of the position of the magnet exists due to overcoming the inertial force of the mass of the moving parts, the inductance of motor coil, and so on.

As one of the aims of modeling the system is to make a controller based on a linear control theory, a linear model of the system is required. However, in this system, the attractive force of the magnet is only nonlinear. It may be linearlized as

$$\tilde{f}_m = k_m z_0 - k_m z_1 \tag{4}$$

where k_m is constant. The linearlized state space model whose input is displacement of the magnetic position and output is the displacement of the lever is:

$$\dot{x}_1 = A_1x_1 + b_1u_1 \tag{5}$$
$$y = c_1x_1 \tag{6}$$

where x_1 is state vector $x'_1 = (z_0 \quad \dot{z}_0)$, input $u_1 = z_1$ and

$$A_1 = \begin{pmatrix} 0 & 1 \\ \frac{k_m - k_{s0}}{m_0} & -\frac{k_{c0}}{m_0} \end{pmatrix},\ b_1 = \begin{pmatrix} 0 \\ -\frac{k_m}{m_0} \end{pmatrix},\ c_1 = (1 \quad 0)$$

The system represented by this model is stable when $k_0 > k_m$, and it is controllable and observable.

Force Input System When the input of the system is defined by the force of the actuator, the model is represented by Eq. (1), (2) and (3). The system input is the actuator force f_a and the outputs are the displacements of the lever and the magnets. The block diagram is shown in Fig. 5. For Fig. 5, an external force is also input at the point of the force for the lever. The slider for the actuator is assumed to be supported by a spring whose stiffness is stable enough for the system without a need for active control. The spring is omitted from Fig. 3.

A linearlized state space model of the system is:

$$\dot{x}_2 = A_2x_2 + b_2u_2 \tag{7}$$
$$y = C_2x_2 \tag{8}$$

where x_2 is state vector $x'_2 = (z_0 \quad \dot{z}_0 \quad z_1 \quad \dot{z}_1)$, input $u_2 = f_a$

$$A_2 = \begin{pmatrix} 0 & 1 & 0 & 0 \\ \frac{k_m - k_{s0}}{m_0} & -\frac{k_{c0}}{m_0} & -\frac{k_m}{m_0} & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{k_m}{m_1} & 0 & \frac{k_m - k_{s1}}{m_1} & -\frac{k_{c1}}{m_1} \end{pmatrix}, \quad b_2 = \begin{pmatrix} 0 \\ 0 \\ 0 \\ -\frac{1}{m_1} \end{pmatrix}$$

and $C_2 = \begin{pmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{pmatrix}$.

The proper spring constant k_1 and damping factor k_{c1} makes the system stable without active feedback. Numerical simulations were carried out on just such a spring-and-damper system. The system proved to be controllable and observable with or without the spring and the damper.

4. NUMERICAL SIMULATION

Numerical simulations were carried out on the models. A digital controller was used for examinations. Simulations with quantified value input and/or discrete time inputs were examined.

4.1 Controller

The controller for the vibration control system is a regulator with the LQ theory. The feedback gains are calculated by MATLAB on the state space model. The state variables are assumed to be gained precisely without delay.

The feedback rule is, when the system input is the magnet position,

$$z_1 = -G_{pre}(k_p z_0 + k_d \dot{z}_0) \qquad (9)$$

where, k_p and k_d are feedback gains. G_{pre} is a transfer function of time delay. When the system input is the actuator force, the feedback rule is

$$f_a = -(k_{p0} z_0 + k_{d0} \dot{z}_0 + k_{p1} z_1 + k_{d1} \dot{z}_1) \qquad (10)$$

where, k_{p0}, k_{d0}, k_{p1}, k_{d1} are feedback gains.

Parameters used in simulation are: $m_0 = 1$, $m_1 = 3$, $d_0 = 0.1$, $k_{s0} = 70$, $k_{c0} = 0.1$, $k_{s1} = 200$, $k_{c1} = 0.7$, $k = 0.01$.

4.2 Magnet Position Input System

Simulations were carried out with the lever initially set to 0.01 and so that it vibrates freely. Then, after 1 unit of time, the active vibration control starts. The magnets' positions are assumed to be saturated at ± 0.04, and the feedback gains are: $k_p = -9.28$ and $k_d = -0.68$.

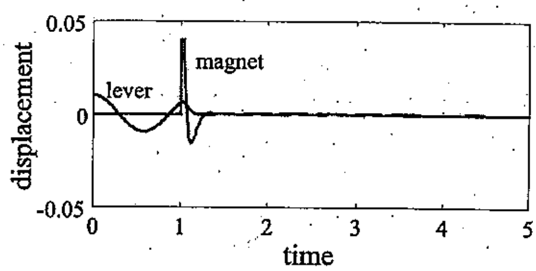

Fig.6 Simulation Result (Position Input, $G_{pre} = 1$)

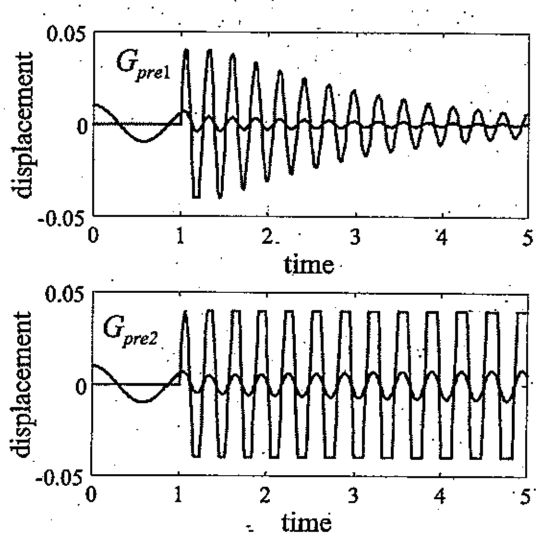

Fig.7: Simulation Result (Position Input, $G_{pre1} = \frac{1}{0.06s+1}$ and $G_{pre2} = \frac{1}{0.07s+1}$)

The results for the $G_{pre} = 1$ case, which is the system without delay, are shown in Fig. 6. The movements of the lever and the magnet are recorded. As shown in the figure, the vibration is suppressed rapidly after active control.

The result of $G_{pre1} = \frac{1}{0.06s+1}$ and $G_{pre2} = \frac{1}{0.07s+1}$ are shown in Fig. 7. The upper figure is for G_{pre1} and the lower figure for G_{pre2}. As shown in these figures, the grater the time delay the more unstable the system.

Next the effect of quantified input to the system is examined. Digital controller outputs quantified the values. Because this is not good for system stability, the tolerance should be known. The control input is quantified with 0.01 interval. The value is a relatively large value. The result is shown in Fig. 8. As shown in the figure, if the control input has a quantified value, its stability can be maintained.

The digital controller outputs a discrete value sampled on a fixed time interval. The result for a sampling

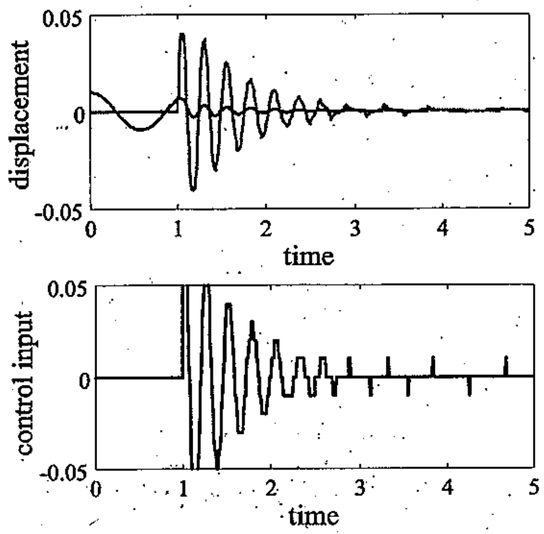

Fig.8: Simulation Result (Position Input, $G_{pre} = \frac{1}{0.05s+1}$, Quantified Input)

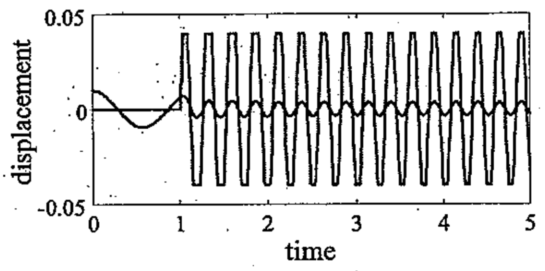

Fig.9: Simulation Result (Position Input, $G_{pre} = \frac{1}{0.05s+1}$, Sampling time = 0.005)

time of 0.009 is shown in Fig. 9. As seen in the figure, the system is stable. The result for a sampling time of 0.01 that is unstable. Thus sampling time is a stability variable.

4.3 Actuator Force Input System

This simulation is carried out in a similar manner to the position input system simulation. The lever is initially set to 0.01 and vibrates freely. In this simulation, however, the magnet also vibrates in harmony with the lever. After 3 time units, the active control starts, and the results are recorded until time unit 10. The actuator force is assumed to be saturated at ± 0.4. The feedback gains are calculated from the LQ theory as $k_{p0} = 21.8$, $k_{d0} = -4.77$, $k_{p1} = 29.9$ and $k_{d1} = 12.7$.

The result of the continual control is shown in Fig. 10. The upper figure shows the displacement of the lever and the magnet, while the lower figure shows the force of the actuator. It is observed that initial error

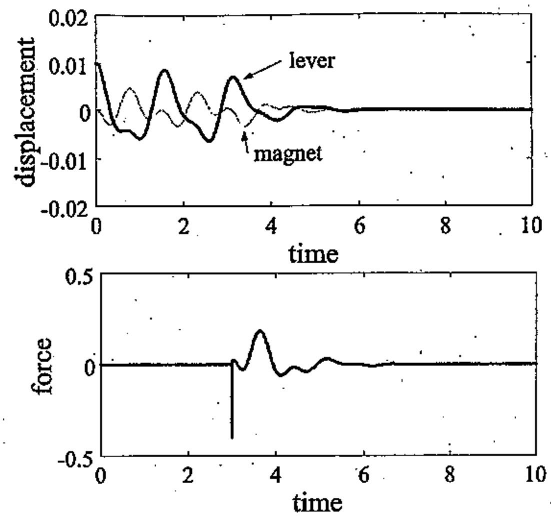

Fig.10 Simulation Result (Force Input)

of the lever position causes vibration of both the lever and the magnet. After active control, the vibration converges to the origin.

When the force of the actuator has a quantified value with an interval of 0.05, the result is shown in Fig. 11. As shown in the readout for the actuator force, the force varies as a step function. Because the interval 0.05 is relatively large, the system converges towards the origin. However residual vibration is observed. The reason is that the quantified input can not control it. The adjustment of feedback gains can improve the amplitude of the vibration.

When the force of the actuator has a discrete value adding to a quantified value, the results are shown in Fig. 12. The sampling time is 0.1. This sampling time is large in comparison with the position input system. The spring which supports the magnets to make the system stable may affect the system stability.

5. CONCLUSION

A novel vibration control method which uses a linear actuator and a permanent magnet has been proposed. An experimental system has been introduced and modeled. The vibration control system has been modeled for two types of the system input. Both models have been verified to be controllable and observable. From these numerical simulations, it has been proven that both position and force control system can suppress vibration. The actuator force control system has been shown to be more robust than the position control system. Consideration of the time delay for the system has been a major factor in stabilizing the

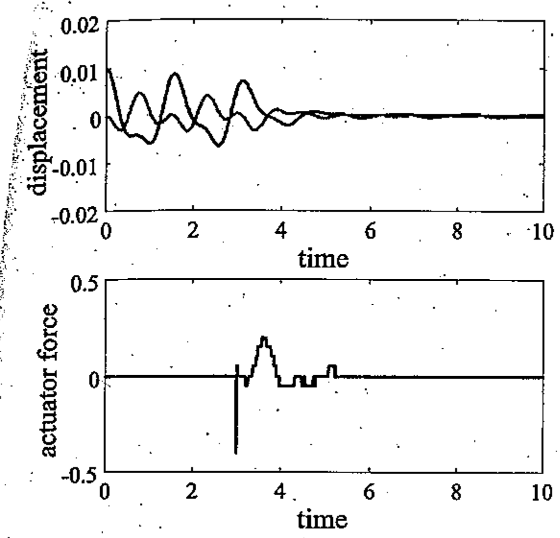

Fig.11 Simulation Result (Quantified Force Input)

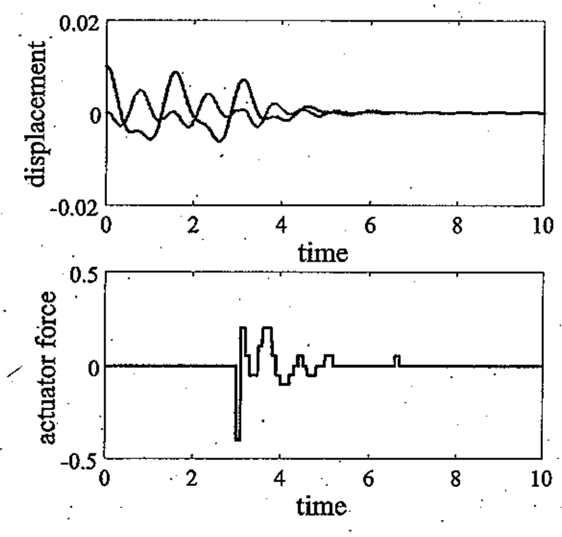

Fig.12: Simulation Result (Discrete and Quantified Force Input)

system especially for position control system model. The importance of estimating the delay in positioning of the actuator has also been demonstrated.

Further studies involving identification of the parameters of the experimental device and its examination will be ongoing.

ACKNOWLEDGMENTS

The author gratefully acknowledges the efforts of correcting proofs for this paper by Prof. David Greene.

REFERENCES

1. Matsuda K., Yoshihashi M., Okada Y. and Tan ACC: Self-Sensing Active Suppression of Vibration of Flexible Steel Sheet, *Trans. ASME, J. Vib. Acoust.* Vol.118, 469-473, (1996)
2. Fujita M., Matsumura F. and Namerikawa T.: μ-Analysis and Synthesis of a Flexible Beam Magnetic Suspension System, *Proc. 3rd Int. Symp. Magnetic Bearings*, 495-504 (1992)
3. Stephens L.S, Timmerman M.A. and Casemore, M.A.: Gain Scheduled PID Control for Elecromagnetic Vibaration Absorbers, *Proc. 6th Int. Symp. Magnetic Bearings*, 331-340, (1998)
4. Betschon F. and Schöb R.: On-Line=Adapted Vibration Control, *Proc. 6th Int. Symp. Magnetic Bearings*, 362-371, (1998)
5. Shinko Electric co, LTD,: Electromagnet Vibration Control System for Steel Sheet (in Japanese), *http:// www.shinko-elec.co.jp /NewsRelease /new_18.htm*
6. Oka K. and Higuchi T.: Magnetic Levitation System by Reluctance Control: Levitation by Motion Control of Permanent Magnet *Int. J. Applied Electromagnetics in Materials*, Vol.4, 369-375, (1994)