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A Study of Vibration Suppression Using Permanent Magnet and Actuator

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Abstract 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. First the principle of the proposed vibration suppression system will be explained. Second a prototype vibration control system will be introduced and modelled. Some numerical and experimental examinations will be carried out. The results support the proposed vibration suppression mechanism which is feasible.

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 suppressor with mechanical contacts is not suitable in such a process. Because 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 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 con-

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stricted, 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 modelled. Following that, the experimental system is analysed according to linear control theory, and a controller is designed based on the results. Numerical simulations and experimental examinations are carried out to demonstrate the properties of the control method and its feasibility.

2 Proposed Vibration Suppression Mechanism

A schematic proposed system 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.

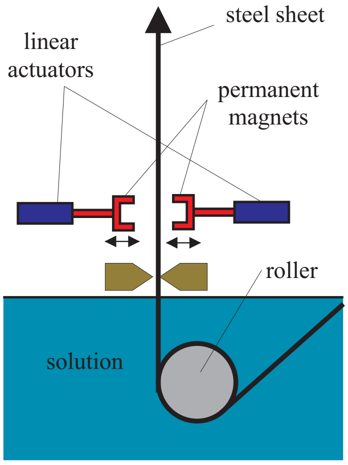

Fig. 1 The steel sheet is coming from the right, turns around the rotor and is pulled up. In the upwards process, the solution would be dried. A mechanical vibration suppression system damage the steel sheet.

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

An experimental system to examine the performance of the proposed vibration control method was devised. This system was modelled in order to analyse the linear control theory and to synthesise 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 vibration body supported by two parallel plate springs is the controlled object to suppress vibration. These springs can be replaced and an arbitrary stiffness can be determined. The control force is created by two permanent magnets which

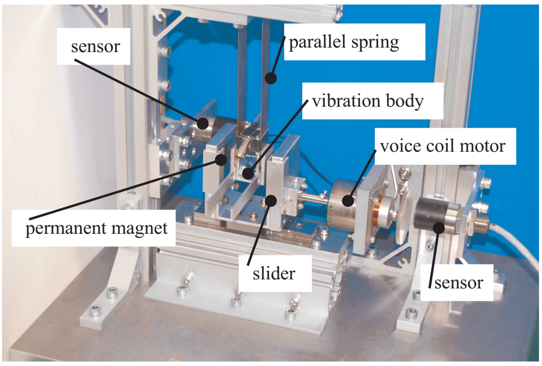

Fig. 2 An experimental device consists of two sensors, a voice coil motor (VCM), a slider, two permanent magnets, a vibration body, and supporting parallel springs. The VCM actuates the slider and two sensors sense the position of the slider and the vibration body.

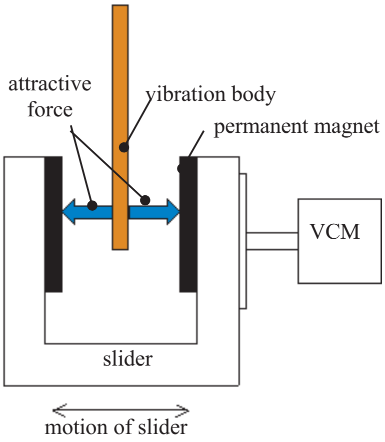

Fig. 3 The VCM actuates the slider and changes the air gaps between the vibration body and permanent magnets. It controls the attractive forces acting on the body.

are located in slider actuated by a linear actuator. Two iron plates are installed in the vibration body. They are positioned so that the face to the permanent magnets as attractive forces act on the body. The linear actuator is a voice coil motor which is called a VCM. The actuator has a stroke length of 15 mm.

3.2 Modelling of System

Modelling 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 vibration body is assumed to be translational, as it supported by parallel springs. The positive direction is the rightward direction in Fig. 3

The symbols used in the model are: z_0 and z_1 are displacements of the body and the slider, d_0 is the air gap width when the body is centered between the magnets, k_{s0} and k_{s1} are parallel spring constants, k_{c0} and k_{c1} is damping coefficients, m_0 is the equivalent mass of the body, m_1 is the mass of slider together with the permanent magnet, 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 body 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

$$m_1 \ddot{z}_1 = -fm - k_{s1} z_1 - k_{c1} \dot{z}_1 + f_a \tag{3}$$

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 body and the magnets. The block diagram is shown in Fig. 4. For Fig. 4, an external force is also input at the point of the force for the body. 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:

$$\dot{x} = Ax + bu \tag{4}$$

$$y = Cx \tag{5}$$

where x is state vector $x = (z_0 \quad \dot{z}_0 \quad z_1 \quad \dot{z}_1)\prime$, input $u = f_a$

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

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

The proper spring constant k_1 and damping factor k_{c1} makes the system stable without active feedback. Numerical simulation and experimental examinations 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.

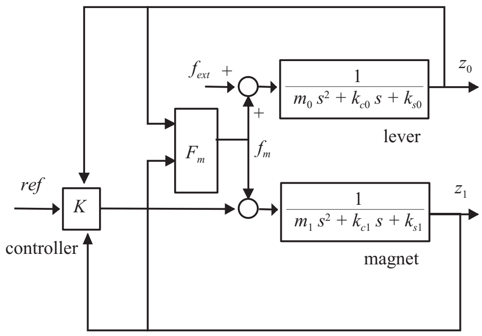

Fig. 4 There are two blocks that indicate the movement of the slider and the vibration body. Vibration body is assumed that it is supported by springs and a second order system. Fm indicates the attractive force function.

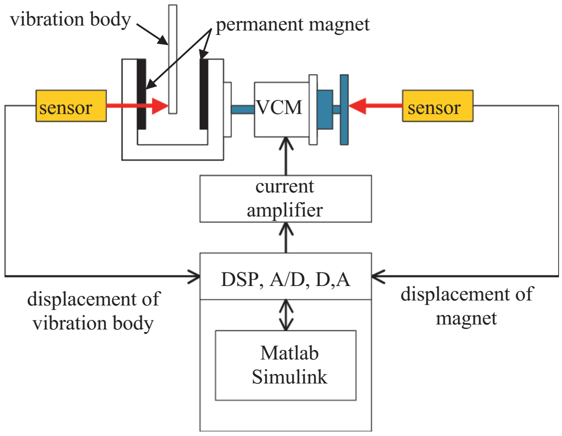

Fig. 5 There are two feedback loops in the system. One is the loop for the vibration body and another is the loop of the permanent magnet. They are inputted to the DSP controller. As the VCM is driven by a current amplifier, the system input can be considered as the force for the magnet motion.

4 Examination

4.1 System Setup

For vibration suppression control, a digital controller was used for examinations. The system structure is shown in Fig. 5 The controller is DSP. The two sensor feedback signals are convert to digital values by A/D converters and they are inputted to the controller. The controller calculates appropriate value of the actuator force and the value is outputted to the current amplifier through a D/A converter.

The controller for the vibration control system is a regulator with two PD feedback loops. 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,

$$f_a = -(k_{p0}z_0 + k_{d0}\dot{z}_0 + k_{p1}z_1 + k_{d1}\dot{z}_1) \tag{6}$$

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

Parameters used in simulation are aligned to the actual experimental device and these values are $m_0 = 0.1$[kg], $m_1 = 0.366$[kg], $d_0 = 0.007$[m], $k_{s0} = 41.7$[N/m], $k_{c0} = 0.05$[Ns/m], $k_{s1} = 0$, $k_{c1} = 0$, $k = 3.67 \times 10^{-7}$[Nm^2], $k_m = 4.29$[N/m].

4.2 Numerical Simulation

Two simulations were carried out with the lever initially set to 3 [mm] and so that it vibrates freely. One simulation was done without feedback control. In this simulation, the magnet was set to stand still to the original point. The result of the simulation is shown in Fig. 6. And another simulation is with feedback control and the result is shown in Fig. 7 The feedback gains are $k_{p0} = 500$, $k_{d0} = 50$, $k_{p1} = 800$

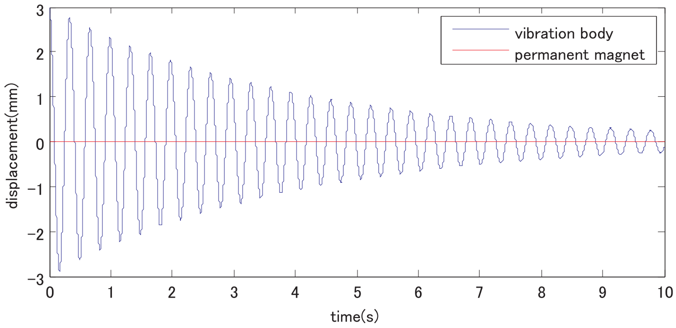

Fig. 6 Numerical simulation result without feedback control

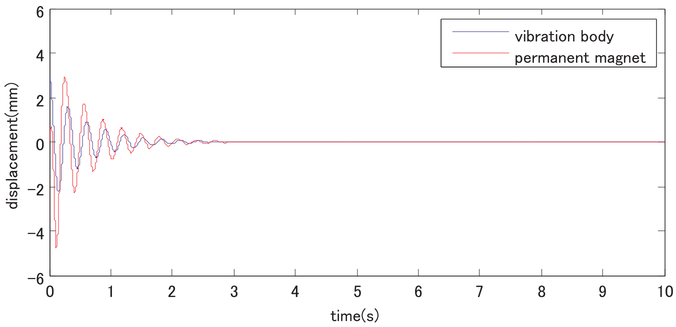

Fig. 7 Numerical simulation result with feedback control

and $k_{d1} = 10$. During the both simulation, the signals of the actuator force and the displacement of the vibration body are recorded.

As shown in Fig. 6 and Fig. 7, both systems converge to the original position. The convergence time with feedback control is, however, much shorter than without feedback control. The proposed vibration suppression mechanism is proved to be effective.

4.3 Experimental Examination

For experiment, we carried out same examinations as numerical simulations. One experiment was done without feedback control. The permanent magnet was controlled as the position in the origin. Another experiment was with feedback control and the gains were the same as they were in the simulation. The results are shown in Fig. 8 and Fig. 9.

In the experiment without feedback control, almost same result was observed as the simulation. The vibration converged to the original position, however it takes

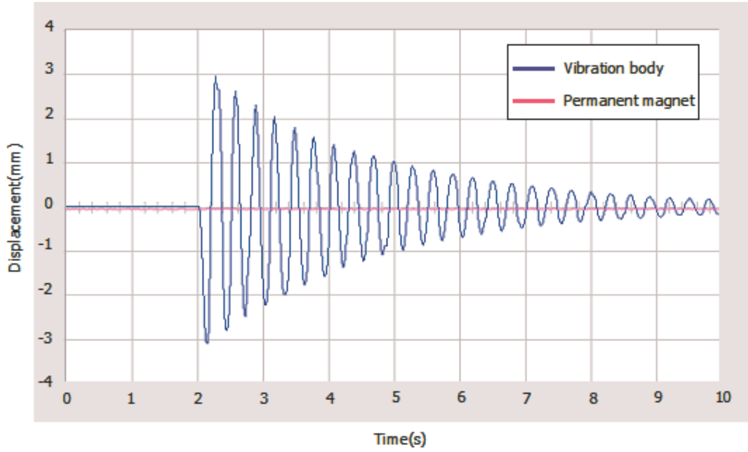

Fig. 8 Experimental examination without feedback control

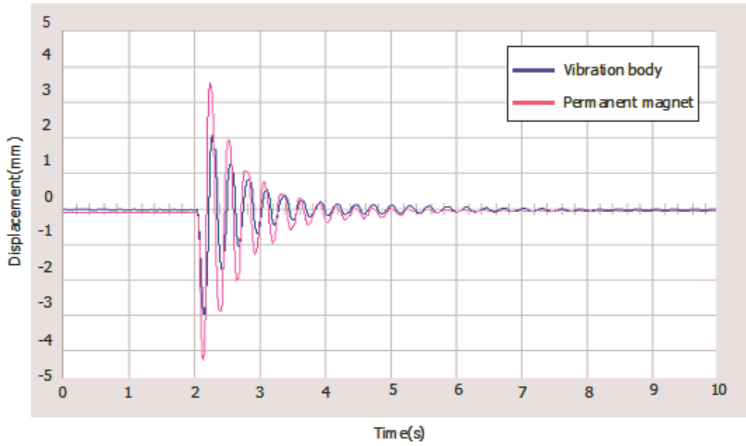

Fig. 9 Experimental examination with feedback control

much time as shown in Fig. 8. As shown in Fig. 9, the result with feedback control indicates the feedback control is effective for vibration suppression. The performance is, however, relative low, as we can see the convergence time is relative longer than the simulation result in Fig. 7. It may be caused by the omitted vibration mode of the parallel springs and the delay of the response of the actuator.

Both results of the simulation and the experiment support the feasibility of the proposed vibration suppression mechanism.

5 CONCLUSION

A vibration control method which uses a linear actuator and a permanent magnet has been proposed. An experimental system has been introduced and modelled. The vibration control system has been modelled and the model have been verified to be controllable and observable. From numerical simulations, it has been proven that the force feedback control system can suppress vibration. Same results has been obtained in experimental examinations. However the performance was relative low because of spillover the vibration of the supported springs and lack of the actuator response. As the result, the proposed vibration suppression mechanism has been proven to be feasible.

Further studies involving consideration of the external disturbance force and investigation of more robust high performance controller will be ongoing.

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