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Experimental Consideration on Vibration Suppression Control of a Magnetically Levitated Thin Steel Plate Using Sliding Mode Control

by

Shinya HASEGAWA[*1], Yasuo OSHINOYA[*2] and Kazuhisa ISHIBASHI[*3]

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Abstract

We have proposed a magnetic levitation control system for a sheet steel and confirmed the realization by digital control experiment. However, because of the strong nonlinearlity of the attractive force of an electromagnet and various uncertainties in circuit current such as changes in resistance due to the heat generation of the electromagnet, the stability of levitation is not sufficiently ensured. As one of the effective control methods for solving this problem, sliding mode control, which enables easy handling of nonlinear models, is attracting attention, and an attempt to use this sliding mode control in electromagnetic bearing control has been reported. In this study, we aim to develop a noncontact support system for thin steel plates with high robustness using sliding mode control, which is tolerant to factors such as disturbance with respect to control signals and the external force of the system. We applied a 1-degree-of-freedom model and a continuous model for the modeling of sheet steel. Then, experiments were carried out under several conditions, and the obtained results were compared with the optimal control results. As a result, it was verified that the suppressive effect of the sliding mode control on disturbance is sufficient and the application of the continuous model enables the construction of a system with robustness to the disturbance of the external force.

Keywords: Steel Plate, Electromagnetic Levitation, Disturbance, Elastic Vibration, Continuous Model, Optimal Control, Sliding Mode Control

1. Introduction

Thin steel plates are widely used as materials for automobiles, electric appliances, cans and other products in current industries. The surface quality required for thin steel plates is becoming higher due to various demands on the thin steel plates. However, because contact conveyance using a roller is mainly adopted in the conveyance process of a thin-steel-plate production line, the problem of surface quality deterioration arises. In recent years, as a countermeasure for this problem, research on the noncontact conveyance system with the application of electromagnetic levitation technology has been active [(1)-(3)]. To date, our research group has constructed an electromagnetic levitation control system with which the relative distance between an electromagnet and a steel plate is maintained at a constant value, aiming to prevent the steel plate from falling from the conveyer or coming into contact with the electromagnet during electromagnetic levitation conveyance [(4)]. However, because of the strong nonlineality of the attractive force of the electromagnet and the various uncertainties in the circuit current such as changes in the resistance due to heat generation of the electromagnet, the stability of levitation is not sufficiently ensured. As one of the effective control methods for solving this problem, sliding mode control, which enables easy handling of nonlinear models, is attracting attention, and an attempt to use this sliding mode control in electromagnetic bearing control has been reported[(5)(6)]. Sliding mode control is highly robust, due to which any disturbance satisfying the matching condition, i.e., disturbance and modeling errors existing in the same channel as the control input, can theoretically be completely removed [(7)]. Accordingly, when sliding mode control is applied to electromagnetic levitation control of thin steel plates, it is expected not only to show excellent control performance against the deterioration of the levitational effect due to the nonlinearlity of the electromagnetic attractive force, but also to ensure robustness against various uncertainties in the circuit current which cannot be eliminated by applying a general linear control theory, thereby enabling the construction of more stable control systems. Under such circumstances, in this study, we simulated uncertainties included in the circuit current by forcibly inputting disturbance into control signals of an electromagnet used for the levitation of thin steel plates, and investigated the effect of sliding mode control in suppressing such disturbances.

Another major problem in electromagnetic levitation control of thin steel plates is the lack of levitation stability due to elastic vibration of the thin steel plate, which is a flexible body. Assuming an actual electromagnetic levitation conveyance process, we can consider a case in which a thin steel plate undergoes elastic vibration due to an unexpected external force other than that from the supporting electromagnets. However, no attempt at controlling the

＊1 Graduate Student, Course of Mechanical Engineering
＊2 Associate Professor, Department of Prime Mover Engineering
＊3 Professor, Department of Mechanical Engineering, School of Engineering II

artificially produced elastic vibration in a thin steel plate during its levitation has been reported. Therefore, we installed an electromagnet to produce disturbance in an experimental apparatus, and artificially applied an external force to the thin steel plate during levitation by electromagnetic attractive force, and investigated the effect of suppressing the elastic vibration. We also performed a comprehensive investigation wherein these disturbances were applied simultaneously, assuming an actual industrial process.

To model a thin steel plate, we used a continuous model considering the vibration mode, and a 1-degree-of-freedom (1DOF) model with which a control system can be easily designed. Then, discussion is made from the viewpoint of a performance evaluation of the sliding mode applied to these two models. In this study, we designed a control system using discrete-time sliding mode control [8] with which the spillover of higher order electric vibration modes, that had previously been neglected, is controlled by the suppression of chattering. In addition, an optimal control was also adopted to compare the control performance, and experiments were performed for various cases.

2. System for control experiment

Figure 1 shows an outline of the control system and experimental apparatus [4]. The object of electromagnetic levitation is a rectangular zinc-coated steel plate (SS400) with length a = 800 mm, width b = 600 mm, and thickness h = 0.3 mm. To accomplish noncontact support of a rectangular thin steel plate using 5 pairs of electromagnets (Nos. 1-5) as if the plate was hoisted by strings, the displacement of the steel plate is measured using five eddy-current gap sensors. Here, the electric circuits of paired electromagnets are connected in series, while an eddy-current gap sensor is positioned between the two magnets of each pair. The detected displacement is converted to velocity using digital differentiation. In addition, the current in the coil of the electromagnets is calculated from the measured external resistance, and a total of 15 measured values are input into the digital signal processor (DSP) via an A/D converter to calculate the control law. A control voltage is output from the D/A converter into a current-supply amplifier to control the attractive force of the 5 pairs of electromagnets in order that the steel plate is levitated below the surface of the electromagnets by 5 mm. In this

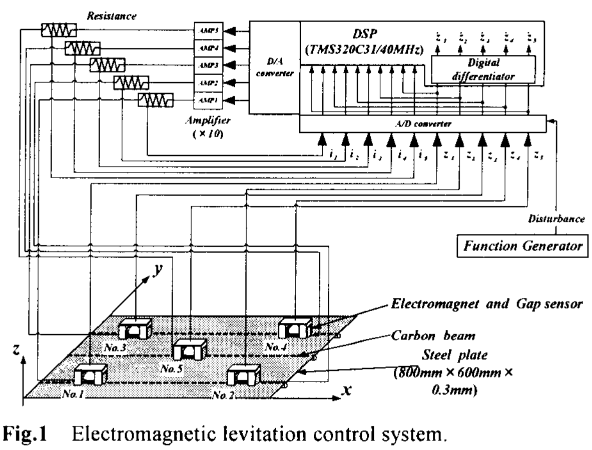


Fig.1 Electromagnetic levitation control system.

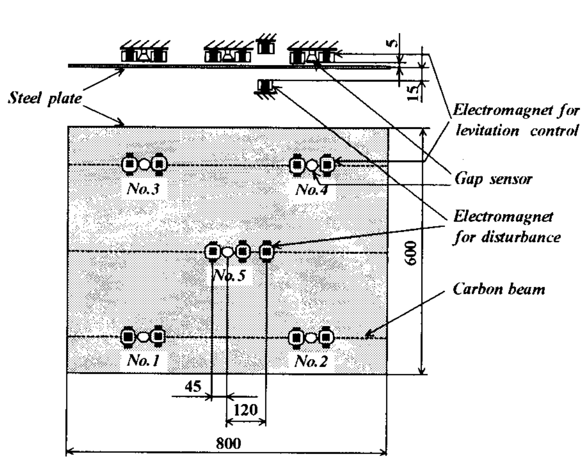


Fig.2 Experimental apparatus.

study, for the basic examinations, the steel plate was reinforced by three pipes in order to suppress the elastic vibrations in the y-axis direction. The dotted lines in the steel plate shown in Fig. 1 are the pipes made of light carbon fiber. The total weight of the carbon pipes is about 6% of the steel plate mass. It was confirmed that the influence of attaching a pipe to the steel is negligible when low-order modes, such as first- and second-order modes, are considered. In addition, as shown in Fig. 2, two electromagnets for the generation of disturbance are placed at the antinode position of the 1st elastic mode vibration, so that the two electromagnets sandwich the steel plate from top and bottom sides.

3. Modeling of steel plate

3.1 1DOF model

In a 1DOF model, independent control is carried out, in which information on detected values of displacement, velocity and coil current of the electromagnet under study at one position are feedback only to the same electromagnet. Therefore, as shown in Fig. 3, the steel plate is divided into 5 hypothetical masses and each part is modeled as a lumped constant system.

When a steel plate is supported by the static electrostatic attractive force of an electromagnet, an equilibrium state exists in which the steel plate is levitated at a constant height. Designating the displacement of the steel plate in the vertical direction of this state as z_n (n corresponds to the numbers 1-5 in Fig. 1), the equation of motion is given as follows.

$$m_{ln}\ddot{z}_n = 2f_{ln} \qquad (1)$$

Where m_{ln}: mass of the steel plate into five [kg], z_n: vertical displacement [m], and f_{ln}: dynamic magnetic force [N].

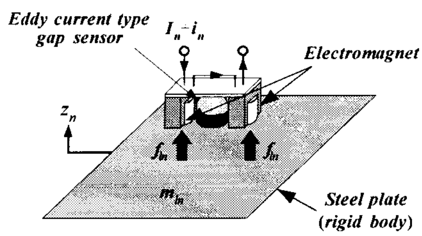


Fig.3 Theoretical model of levitation control of the steel plate.

If deviation around the static equilibrium state is very small, the characteristic equations of the electromagnet are linearized as

$$f_{ln} = \frac{F_n}{Z_0} z_n + \frac{F_n}{I_n} i_n \tag{2}$$

$$\frac{d}{dt} i_n = -\frac{L_{eff}}{L_z} \cdot \frac{I_n}{Z_0^2} \dot{z}_n - \frac{R_z}{2L_z} i_n + \frac{1}{2L_z} v_n \tag{3}$$

$$L_z = \frac{L_{eff}}{Z_0} + L_{lea} \tag{4}$$

where F_n: magnetic force of the coupled magnets in the equilibrium state [N], Z_0: gap between the steel plate and electromagnet in the equilibrium state [m], I_n : current of the coupled magnets in the equilibrium state [A], i_n: dynamic current of the coupled magnets [A], L_z: inductance of one magnet coil in the equilibrium state [H], R_z: resistance of the coupled magnet coils [Ω], v_n: dynamic voltage of the coupled magnets [V], L_{eff}/Z_0: effective inductance of the one magnet coil [H], and L_{lea}: leakage inductance of the one magnet coil [H].

3.2 Continuous model [4]

Continuous model which expresses the motion of the steel plate by means of equations in which the original elastic vibrations of the plate is considered. At an equilibrium levitation state, magnetic forces are determined so as to balance with an elastic force and gravity. The equation of small vertical motion around the equilibrium state of the steel plate subjected to magnetic forces is expressed as follows.

$$\rho h \frac{\partial^2}{\partial t^2} z + \frac{Ch^3}{12} \frac{\partial}{\partial t} \nabla^4 z + D\nabla^4 z$$

$$= \sum_{n=1}^{5} f_{cn}(t)\{\delta(x - x_{a1n})\delta(y - y_{a1n}) + \delta(x - x_{a2n})\delta(y - y_{a2n})\}$$

$$+ w(t)\delta(x - x_{d1n})\delta(y - y_{d1n}) \tag{5}$$

$$\nabla^4 = \frac{\partial^4}{\partial x^4} + 2\frac{\partial^4}{\partial x^2 \partial y^2} + \frac{\partial^4}{\partial y^4}$$

Where ρ: density [kg/m^3], h : thickness [m], C: internal damping coefficient [Ns/m2], D: $Eh^3/12(1-v^2)$ [Nm], v: Poisson ratio, E : Young's modulus [N/m^2], $f_{cn}(t)$: dynamic magnetic force at the n-th coupled magnets [N], t: time [s], $z(x,y)$: vertical displacement [m], x, y, z: coordinate axes indicated in Fig.1[m], x_{a1n}, x_{a2n}, y_{a1n}, y_{a2n}: location of the n-th coupled magnets (n=1,2,3,4,5) [m], $\delta(\ )$: Dirac delta function [l/m], $w(t)$: dynamic magnetic force at the disturbance [N], x_{d1n}, y_{d1n}: position of the electromagnets for disturbance [m].

The characteristic equations of the electromagnets can be derived in much the same as 1DOF model, that is

$$f_{cn} = \frac{F_n}{Z_0} z(x_{sn}, y_{sn}) + \frac{F_n}{I_n} i_n \tag{6}$$

$$\frac{d}{dt} i_n = -\frac{L_{eff} I_n}{L_z Z_0^2} \frac{d}{dt} z(x_{sn}, y_{sn}) - \frac{R_z}{2L_z} i_n + \frac{1}{2L_z} v_n \tag{7}$$

where x_{sn}, y_{sn} :position of the n-th sensor[m].

4. State equation

4.1 State equation for 1DOF model

Using the state vector, the equations (1) ~ (4) are written as the following state equations:

$$\dot{z}_{ln} = A_{ln} z_{ln} + B_{ln} v_{ln} \tag{8}$$

$$z_{ln} = [z_{ln} \quad \dot{z}_{ln} \quad i_{ln}]^T$$

$$A_{ln} = \begin{bmatrix} 0 & 1 & 0 \\ \frac{2F_n}{m_z Z_0} & 0 & \frac{2F_n}{m_z I_n} \\ 0 & -\frac{L_{eff}}{L_z} \cdot \frac{I_n}{Z_0^2} & -\frac{R_z}{2L_z} \end{bmatrix}$$

$$B_{ln} = \begin{bmatrix} 0 & 0 & \frac{1}{2L_z} \end{bmatrix}^T$$

4.2 State equation for continuous model

The vertical displacement of the plate can be expanded to an infinite series of a space-dependent eigenfunction $\phi_i(x,y)$ as shown in Fig.4 multiplied by the time-dependent normal coordinate. The eigenfunctions of the plate are assumed to be products of the elastic beam eigenfunctions of the x- and y-coordinates. The function of x-coordinate $X_{mm}(x)(mm=1,2,\ldots)$ satisfies the free-free boundary condition, and the function of the y-coordinate is expressed in rigid modes(parallel and rotational motions) $Y_1(y), Y_2(y)$ only. In addition, since the number of sensors used in this experiment is 5, we selected $M = 5$ for the control in which consideration is given to the 5th mode (1st elastic mode). The mode-expansion equations are given as follows.

$$z(x, y) = \sum_{i=1}^{M} \phi_i(x, y)\xi_i(t) \tag{9}$$

$$\phi_i(x, y) = X_{mm}(x) \cdot Y_{nn}(y) \quad (mm, nn = 1, 2, \cdots)$$

$$X_1(x) = 1, \quad X_2(y) = \frac{\sqrt{3}}{a}(2x - a)$$

$$Y_1(y) = 1, \quad Y_2(y) = \frac{\sqrt{3}}{b}(2y - b)$$

$$Y_{nn}(y) = cos\frac{\lambda_{ynn}}{b} y + cosh\frac{\lambda_{ynn}}{b} y$$

$$+ \frac{\sin\lambda_{ynn} + \sinh\lambda_{ynn}}{\cos\lambda_{ynn} - \cosh_{ynn}} \left(\sin\frac{\lambda_{ynn}}{b} y + \sinh\frac{\lambda_{ynn}}{b} y \right)$$

$$cosh\lambda_{ynn} \cdot cos\lambda_{ynn} = 1 \quad (nn = 3, 4, \cdots)$$

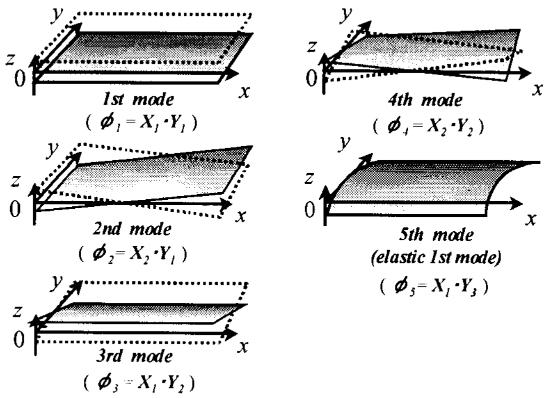


Fig.4 Mode shape of the magnetically levitated steel plate for the continuous model.

$$f_{ynn} = \frac{1}{2\pi}\left(\frac{\lambda_{ynn}}{b}\right)^2 \sqrt{\frac{D}{\rho h}}$$

Where a: length of the plate, b: width of the plate.

State variables of the system are normal coordinates of vertical displacement of the plate ξ_M, The control input of the system is the dynamic voltages of the magnets v_c. Output variables of the system are vertical displacements z_c. Using the state, control and output vectors, the forgoing eqs. (5)~(7) are written as following state and output equations:

$$\dot{\xi}_M = A_{\xi M}\xi_M + B_{\xi M} v_c + G_{\xi M} w \tag{10}$$

$$z_c = C_{\xi M}\xi_M \tag{11}$$

$$\xi_M = \begin{bmatrix} \xi_1 \cdots \xi_M & \dot{\xi}_1 \cdots \dot{\xi}_M & i_1 \cdots i_5 \end{bmatrix}^T$$

$$v_c = \begin{bmatrix} v_1 \cdots v_5 \end{bmatrix}^T$$

$$z_c = \begin{bmatrix} z_1 \cdots z_5 & \dot{z}_1 \cdots \dot{z}_5 & i_1 \cdots i_5 \end{bmatrix}^T$$

here, details of matrices $A_{\xi M}$, $B_{\xi M}$, $C_{\xi M}$ and $G_{\xi M}$ are omitted due to space limitations.

5. Control theory

5.1 Optimal control in the discrete time system [9]

In this study, a control system is constructed using a discrete time system; therefore, the evaluation function of a continuous system is digitized, and the optimal control law is obtained based on the optimal control theory of the discrete time system. Here, the following discrete time system is considered.

$$z_d(i+1) = \Phi z_d(i) + \Gamma v_d(i) \tag{12}$$

$$\Phi = \exp(AT_s),\ \Gamma = \int_0^{T_s} [\exp(A\tau)] d\tau B$$

In the case of a 1DOF model: $A=A_{In}$, $B=B_{In}$.

In the case of a continuous model: $A=A_{\xi 5}$, $B=B_{\xi 5}$.

Here, the evaluation function of the discrete time system is expressed as follows.

$$J_d = \sum_{i=0}^{\infty}\left[z_d(i)^T Q_d z_d(i) + v_d(i)^T r_d v_d(i)\right] \tag{13}$$

Where Q_d and r_d are weighting coefficients.

$$M = \Phi^T M \Phi + Q_d - \Phi^T M \Gamma \left(r_d + \Gamma^T M \Gamma\right)^{-1} \Gamma^T M \Phi \tag{14}$$

$$v_d^o = -F_d z_d \tag{15}$$

$$F_d = \left(r_d + \Gamma^T M \Gamma\right)^{-1} \Gamma^T M \Phi \tag{16}$$

Where M is the solution of the algebraic matrix, the Riccati equation, and T_s is a sampling interval (= 1ms in the experiment). MATLAB command " lqrd " was used to solve eq. (14) and the digital controller was designed by using SIMLINK in the DSP.

5.2 Sliding mode control in the discrete time system [8]

Similar to the previous section (5.1), the discrete time system of eq. (12) is considered. Here we designate the switching hyperplane as S_d, and express the switching function of input as

$$\sigma(i) = S_d z_d(i) \tag{17}$$

The equivalent control input is given as

$$v_{eq}^s(i) = -(S_d\Gamma)^{-1} S_d(\Phi - I) z_d(i) \tag{18}$$

Substituting eq. (18) into eq. (12), the equivalent control system can be expressed as

$$z_d(i+1) = \left\{\Phi - \Gamma(S_d\Gamma)^{-1} S_d(\Phi - I)\right\} z_d(i) \tag{19}$$

here, S_d should be selected such that the system represented by eq. (19) becomes stable. In this study, we used a method of utilizing the zero point of the system, and applied the optimal control theory of the discrete time system to obtain S_d.

$$S_d = \left(r_d^s + \Gamma^T M^s \Gamma\right)^{-1} \Gamma^T M^S \Phi_\varepsilon \tag{20}$$

Where r_d^s is weighting coefficients for control input. M^s is a solution of eq. (14).

Next, a control input which converges the state into a hyperplane and generates the sliding mode is considered. In this study, we design a sliding mode control for the discrete time system, wherein chattering is suppressed . The sliding mode control law used to satisfy this condition is described below.

$$\left.\begin{aligned} v_d^s(i) &= v_{eq}^s(i) + v_{nl}^s(i) \\ v_{eq}^s(i) &= -(S_d\Gamma)^{-1} S_d(\Phi - I) z_d(i) \\ v_{nl}^s(i) &= -\{\alpha(i) + \beta(i)\}\operatorname{sgn}\{\sigma(i)\} \end{aligned}\right\} \tag{21}$$

$$\alpha(i) = \eta\frac{\|\sigma(i)\|}{\|S_d\Gamma\|},\quad 0 < \eta < 2,\quad \beta(i) \geq F_{max} \tag{22}$$

F_{max}: Maximum value of disturbance.

6. Control experiment

6.1 Specifications of the experimental apparatus

The specifications of the system are shown as follows: m=1.08kg, Z_0=5mm, $F_1 \sim F_4$=1.85N, F_5=3.21N, $I_1 \sim I_4$=0.51A, I_5=0.67A,

R_z=20.6Ω, ρ=7500kg/m^3, E=217GPa, ν=0.3, C=2.49×10^8 Ns/m, L_z=0.225H, L_{eff}=1.8×10^{-4}Hm, L_{lea}=0.1891H, T_s=1ms, x_{s1}= x_{s3}= 185mm, x_{s2}= x_{s4}= 615mm, x_{s5}= 400mm, y_{s1}= y_{s3}= 83mm, y_{s2}= y_{s4}= 517mm, y_{s5}= 300mm.

6.2 Setting the comparative standard for control systems

In this study, we examined the vibration-suppression performance during levitation of a flexible steel plate for a total of 4 cases: optimal control applied to the 1DOF model and to the continuous model, and sliding mode control applied to each of the two models. To compare the performance of each control system, a certain standard must be determined. Therefore, carbon-fiber pipes are attached to the steel plate, as shown in Fig. 5, and standard conditions under which the levitation performance of a rigid-body steel plate with suppressed elastic vibration can be considered to be a rigid body are adopted. For the rigid steel plate levitated using each control method, an external force (calculated from the coil current of the electromagnet), as shown in Fig. 6, is applied using the electromagnet for disturbance, as shown in Fig. 2, and the parameters of each control system are set such that the standard deviation of the displacement at the center of the steel plate is within an error range of ±5% for 1.0 × 10^{-5} m during the application of any control method.

The weight matrices of optimal control (eq. (13)) are

- 1DOF model: $Q_d^{1,opt} = I$, $r_d^{1,opt} = 1$.
- Continues model: $Q_d^{c,opt} = I$, $r_d^{c,opt} = I$.

The weight matrices (eq. (13)) used for determining the switching function Sd in sliding mode control, and parameters in the nonlinear input terms in eq. (22) are

- 1DOF model: $Q_d^{1,smc} = I$, $r_d^{1,smc} = 1$, $\eta = 0.05$, $\beta = 3$.
- Continues model: $Q_d^{c,smc} = I$, $r_d^{c,smc} = I$, $\eta = 0.05$, $\beta = 3$.

In the following sections, we discuss the cases in which the flexible steel plate described in Sec. 2 (Figs. 1 and 2) is levitated using control systems designed with the above-described parameters.

6.3 Vibration-suppression performance when disturbance is within circuit current of electromagnet

When a steel plate is levitated using an electromagnet, the change in the resistance due to heat generated in the electromagnet as well as power source noise significantly affects the control performance. When an uncertainty is included in the control system, it is very important in practice to design a control system that accommodates such uncertainties. Therefore, here we investigate the effect of the suppression of uncertainties included in the same channel as control signals, namely, a disturbance which satisfies the matching condition of sliding mode control.

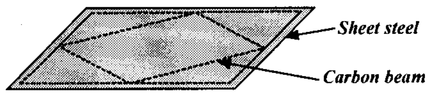


Fig.5 Rigid body steel plate.

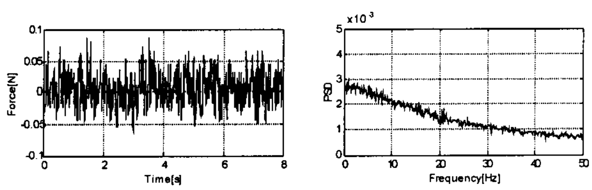


Fig.6 Time history and power spectrum of disturbance from electromagnet.

To simulate the uncertainty included in the circuit current of an electromagnet, disturbance was forcibly added to control the signals of electromagnet No. 5 placed at the center of the steel plate. Figure 7 shows an example of the time history (voltage measured) and power spectrum density of disturbance added to the control signals. Here, as is apparent from the results described later, we have confirmed by both experiment and numerical simulation that odd-number modes of third or higher mode vibrations are rarely excited due to the damping characteristics of the control system. Therefore, in this study, we mainly discuss the 1st elastic mode vibration, which most frequently appears.

Figure 8 shows experimental results. In Figs. 8, examples of the results of the time history and power spectrum density of the displacement at the center of the steel plates are shown: (a) and (b) represent the cases in which optimal control and sliding mode control are applied to the 1DOF model, respectively, and (c) and (d) represent the cases in which optimal control and sliding mode control are applied to the continuous model, respectively. The zero on the vertical axis of the time history represents the position of equilibrium levitation. .

First, we examine the differences between the control methods when the 1DOF model is used. When optimal control is applied (Fig. 8(a)), elastic vibrations are excited on the steel plate due to the enforced disturbance applied to the control signal. A peak of the power spectrum density is observed at 4.75 Hz, which is the natural frequency of the 1st elastic mode vibration of the steel plate used in this experiment. In contrast, when sliding mode control is applied (Fig. 8(b)), because it is possible to cancel the disturbance contained in the circuit current, which is difficult to remove with the conventional linear control theory, the elastic vibration of the steel plate is suppressed. However, in sliding mode control, switching of the control input at an infinite speed is assumed; therefore, the theoretical control performance cannot be completely achieved even when the matching condition is satisfied. Consequently, although slight, vibration of the 1st elastic mode remains and a peak is observed in the spectrum.

Next, we examine differences in the modeling of thin steel plates. In Figs. 8(c) and 8(d), because a continuous model that takes into consideration the 1st elastic mode vibration is used, no peak is observed in the power spectrum density, as compared with the results in Figs. 8(a) and 8(b). In particular, when sliding mode control is applied to the continuous model (Fig. 8(d)), the best control performance is obtained among the four cases, because cancellation of the disturbance contained in the control signals, and the suppression of the 1st elastic mode vibration are possible.

Table 1 summarizes the standard deviation of the time history shown in Figs. 8(a)-8(c). By setting the standard deviation when optimal control is applied to the 1DOF model as the standard value (100%) ((a) in Table 1), the standard deviation in each case is expressed as a percentage relative to th standard value, compared with the cases when optimal control is applied ((a) and (c) in Table 1), indicating the superiority of sliding mode control, regardless of the steel-plate model used. In particular, even when the 1DOF

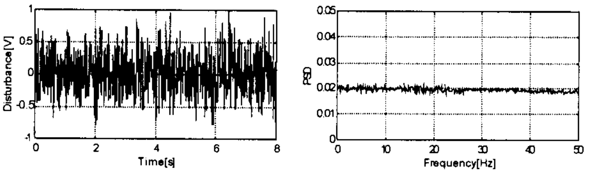
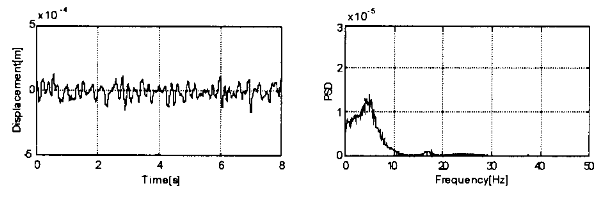
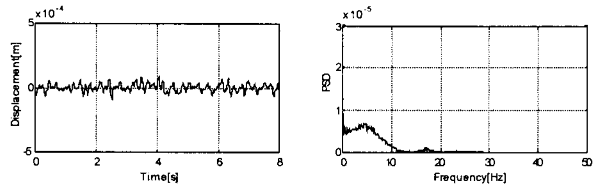


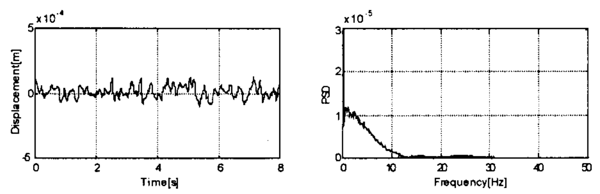
Fig.7 Time history and power spectrum of disturbance into the control input.



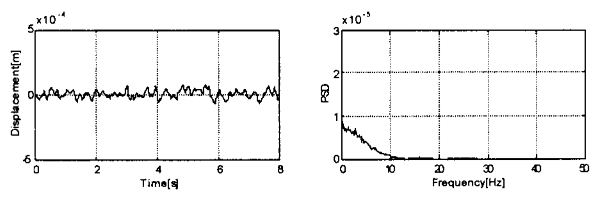
(a) Optimal control using 1 DOF model



(b) Sliding mode control using 1 DOF model



(c) Optimal control using continuous model



(d) Sliding mode control using continuous model

Fig.8 Experimental results of time histories and power spectrums of displacement at center of the steel plate under random excitation from disturbance into the control input.

Table 1 Relative decreasing ratio of the standard deviation on time history (In case that the disturbance was applied for the control input).

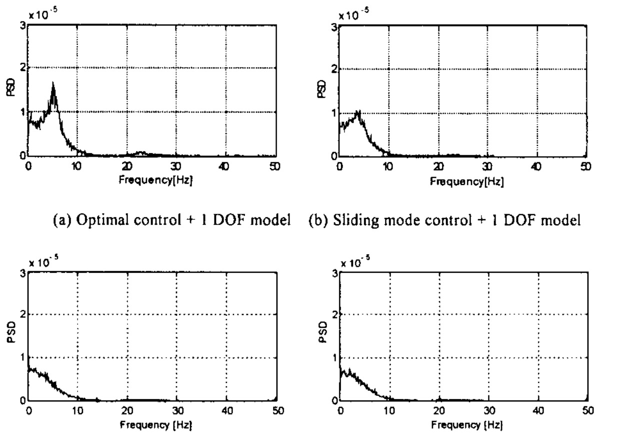
	Modeling and control theory	Standard deviation	Relative decreasing ratio
(a)	1DOF model + Optimal control	5.15×10^{-5} m	100%
(b)	1DOF model + Sliding mode control	3.23×10^{-5} m	63%
(c)	Continuous model + Optimal control	4.25×10^{-5} m	83%
(d)	Continuous model + Sliding mode control	2.85×10^{-5} m	55%

model, which is easy to model, is used, the vibration amplitude can be sufficiently suppressed using sliding mode control ((b) in Table 1).e standard value. The standard deviations for the cases when sliding mode control is applied ((b) and (d) in Table 1) are reduced to around 60% of the standard value, compared with the cases when optimal control is applied ((a) and (c) in Table 1), indicating the superiority of sliding mode control, regardless of the steel-plate model used. In particular, even when the 1DOF model, which is easy to model, is used, the vibration amplitude can be sufficiently suppressed using sliding mode control ((b) in Table 1).

6.4 Performance of suppression of forced elastic vibration generated by external force

We here examine cases in which a steel plate during conveyance receives unexpected external force from sources other than the support electromagnets, which gives rise to elastic vibration. Similar to the arrangement described in Sec. 6.2, an electromagnet for disturbance was placed at the antinode position of the 1st elastic mode vibration, and an external force, as shown in Fig. 6, was applied to the steel plate levitated by electromagnetic attraction force. Thereby, a state in which elastic vibration is easily excited was created. The magnitude of the external force is selected such that the standard deviation of displacement (5.15×10^{-5} m in Table 1) in the case of applying optimal control to the 1DOF model, as described in Sec. 6.3, becomes almost the same as the standard deviation when applying optimal control within approximately 10% error. When disturbance is added to a steel plate from sources other than the support electromagnet, the disturbance basically does not satisfy the matching condition of sliding mode control. Nonami et al. [5] showed that when the 1st delay of the coil current of an electromagnet can be neglected by applying sufficient feedback of the current, the matching condition holds even when the disturbance is absent from the same channel as the control input. However, in this study, the time constant of the electromagnets is 12.6 ms, which is relatively large because it corresponds to 6% of the period of the 1st mode vibration, i.e., 0.21 s. Moreover, due to various restrictions of the system, current feedback cannot be applied to the extent that the 1st delay can be ignored. As a result, it is considered that the complete matching condition does not hold.

In Fig. 9, experimental results are shown (only the power spectrum density of displacement); (a)-(d) represent the same cases as those in Sec. 6.3. In Fig. 9, when the 1DOF model was used (Figs. 9(a) and 9(b)), a peak of the power spectrum density due to spillover of the 1st elastic mode vibration is observed with both control methods. However, in the case of sliding mode control (Fig. 9(b)) which is robust against modeling errors, superiority against lower order spillover can be confirmed. In contrast, when the continuous model is used (Figs. 9(c) and 9(d)), sufficient vibration-suppression performance against 1st elastic mode vibration was observed regardless of the control method applied. However, different from the cases of the 1DOF model, there is almost no difference in the results between Fig. 9(c) and Fig. 9(d). This means that, when the vibration mode of the object to be controlled is taken into consideration in the controller, and therefore spillover



(a) Optimal control + 1 DOF model (b) Sliding mode control + 1 DOF model

(c) Optimal control + continuous model (d) Sliding mode control + continuous model

Fig.9 Experimental results of power spectrums of displacement at center of the steel plate under random excitation from the electromagnet for disturbance.

Table 2 Standard deviation and the relative decreasing ratio on the time histories in the case of the disturbance from the electromagnet.

	Modeling and control theory	Standard deviation	Relative decreasing ratio
(a)	1DOF model + Optimal control	5.27×10^{-5} m	100%
(b)	1DOF model + Sliding mode control	4.48×10^{-5} m	85%
(c)	Continuous model + Optimal control	2.79×10^{-5} m	53%
(d)	Continuous model + Sliding mode control	2.69×10^{-5} m	51%

does not generate modeling errors, the superiority of sliding mode control is not clearly apparent.

Table 2 summarizes the standard deviation of the time history of displacement, similar to Table 1 in Sec. 6.3. When the continuous model is used ((c) and (d) in Table 2), the standard deviation is reduced to approximately 50% of the standard ((a) in Table 2) for both control methods. From this, we can demonstrate that when an external force not satisfying the matching condition is applied to forcibly excite elastic vibration, effective suppression of the elastic vibration can be ensured by the use of the continuous model.

7. Conclusions

Assuming actual processes wherein various disturbances are applied to an electromagnetically levitated thin steel plate, we evaluated the performance of a system including modeling of the thin steel plate, by applying sliding mode control for vibration suppression. The following results were obtained.

(1) For an electromagnetic levitation system of thin steel plates, the application of sliding mode control to the 1DOF model, which is fairly easily modeled, revealed the possibility of effective suppression of disturbance. In particular, sliding mode control was confirmed to be highly robust to uncertainties included in the circuit current of electromagnets. Since the use of a simple model can reduce the development cost of control systems, a result with industrial effectiveness was achieved.

(2) We demonstrated that ensuring superior control performance against disturbance that does not completely satisfy a matching condition is possible by the application of sliding mode control to the continuous model of a thin steel plate, although numerical manipulation is fairly complex. The results of this study are applicable to various disturbances that are expected to occur during actual electromagnetic levitation conveyance processes, thereby enabling practical effective control systems.

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