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Active Vibration Control of a Thin Steel Sheet

The commercial rolling process used in the steel industry to manufacture thin steel sheets tends to cause plate vibrations that lower the quality of the surface finish. This article introduces a noncontact method of active vibration control for reducing the flexural vibrations of a thin steel sheet. The proposed electromagnetic method of control has been implemented in a simple experimental setup where the signal from a motion sensor regulates the attractive force of the magnets that produce a damping force on the steel sheet. © 1995 John Wiley & Sons, Inc.

INTRODUCTION

Thin steel sheet is widely used in automobile structures, electrical appliances, and so on. The steel industry uses a rolling process to manufacture thin steel sheet. Recent market demand for thinner sheets has unveiled the problem of bending vibrations in the rolling process. These plate vibrations deteriorate surface finish quality and should be controlled without introducing surface contacts. Noncontact vibration control has been studied by Oshinoya and Shimogo (1989) and Zhang et al. (1990).

This study introduces a noncontact method of active vibration control using electromagnets. A similar technique is used with magnetic bearings to support a rotating shaft without physical contact. Some examples of magnetic bearing research are reported by Schweitzer and Ulbrich (1980) and Matsumura et al. (1983). A magnetic vibration damper that reduces bending vibrations

in a flexible rotor is described by Nagai et al. (1986) and Allaire et al. (1988). Vibration control of elastic systems is also discussed by Ulsoy (1984).

The active control technique described is applied to suppress vibrations in the traveling steel sheet. Plate vibrations in the steel sheet are detected by an eddy-current type displacement sensor. The measured sensor signal is fed into a controller to produce the actuating current for the magnets. The attractive magnetic force reduces the vibrations in the steel sheet.

In a real rolling process, a large number of magnetic dampers are required. This does not make the solution economically unacceptable because the high-output plant itself is expensive. Nevertheless, it is preferable to reduce the number of required magnetic dampers as much as possible. This consideration led the authors to try modal control to design the control system. It was found, however, that the complexity of the

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modes of vibration makes this approach infeasible. Then, two types of collocated PD (position and derivative) controllers were tested, one an analog PD controller and the other a digital one. A transputer was used to realize high-speed digital control. A simple test apparatus was built to evaluate the usefulness of the proposed technique.

DRIVING METHOD USING VIBRATION MODE SHAPES

If it is possible to predict the modes of vibration accurately, the arrangement method of driving points can be applied. A thin steel plate has a series of resonances and the higher modes are likely to become unstable when feedback control is introduced. An actuator located on a nodal line does not excite the corresponding mode. These nonexcited modes can be selected so that a wide frequency difference exists between the lower and higher frequency modes. The lower frequency modes are intended to be controlled, while the higher frequency modes are not.

Experimental Setup

In the manufacturing process, the steel plate travels at a speed of about 5 m/s and is thinned by rollers to a thickness of about 0.5 mm and a width of 1–2 m. The span of the rollers is usually 10 m. After passing between the rollers, the plate tends to vibrate. The purpose of the active control system is to reduce this vibration.

For experimental convenience, the plate is not made to travel. It is clamped in the x -direction

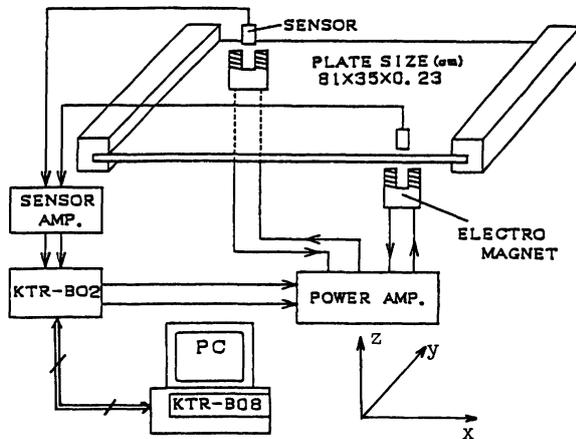


FIGURE 1 The experimental setup.

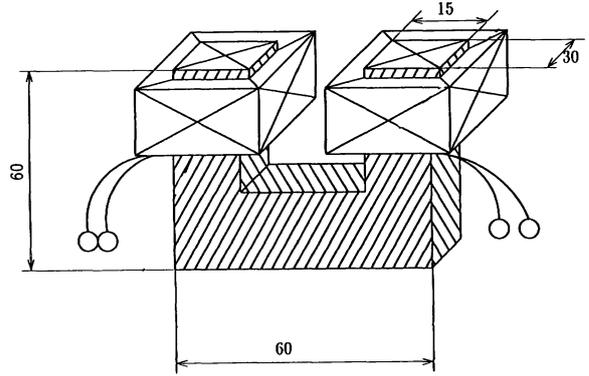


FIGURE 2 The electromagnet.

and free in the y -direction. The plate used in the experiment was 81×35 cm, with a thickness of 2.3 and 0.5 mm. Two gap sensors were arranged on the upper side of the sheet, and two magnetic actuators were installed under the sheet. These are single-operation magnets with a steady-state current of 0.5 A, which means that the attractive force is effective only in the downward direction. A schematic diagram of the experimental setup is shown in Fig. 1. The electromagnet used is shown in Fig. 2. The number of coil turns is 300, the maximum current is 2 A, and the average air gap is 5 mm.

Equation of Motion of System

By considering the deviation from steady-state conditions, the linearized equation for the driving circuit of each magnet can be written as

$$L \frac{di_n(t)}{dt} + Ri_n(t) = e_n(t) \quad (1)$$

$$f_n(t) = Ki_n(t) \quad (2)$$

where e = driving voltage of magnet, [V]; f = force, [N]; i = driving current of magnet, [A]; K = force constant of magnet, [N/A]; L = inductance of magnet, [H]; R = resistance of driving circuit, [Ω]; and n = magnet number (in this case, $n = 1, 2$).

The linearized equation for the vibrations of a thin plate is given by

$$\begin{aligned} \rho h \frac{\partial^2 w(x, y, t)}{\partial t^2} + D \nabla^4 w(x, y, t) - Th \frac{\partial^2 w(x, y, t)}{\partial x^2} \\ = \sum_{n=1}^2 f_n(t) \delta(x - x_n) \delta(y - y_n) \end{aligned} \quad (3)$$

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

$$D = \frac{Eh}{12(1 - \mu^2)}$$

where E = Young's modulus, [N/m²]; f = force, [N]; h = plate thickness, [m]; T = plate tension, [N/m²]; x, y, z = Cartesian coordinates; w = plate displacement, [m]; δ = Dirac's delta function; ρ = plate density, [kg/m³]; and μ = Poisson's ratio.

Modes of Vibration

It is difficult to control a thin plate because it has a series of resonances. One solution is to put actuators on a certain nodal line, the mode for which is not excited by the actuator. The separation between the frequencies of the lower modes, which are to be controlled, and the higher modes, which are uncontrollable, should be made as wide as possible. If the homogeneous part of Eq. (3) has the modal eigenfunctions $\phi_i(x, y)$, then Eq. (3) has the following solution

$$w(x, y, t) = \sum_{i=1}^{\infty} \phi_i(x, y) e^{j\omega_i t}. \quad (4)$$

Assume that $\phi_i(x, y)$ is expanded with the eigenfunctions $X_n(x)$ and $Y_m(y)$ of the beam vibrations. Then

$$\phi_i(x, y) = X_n(x) Y_m(y) \quad (5)$$

where the plate is clamped at $x = 0, a$, and free at $y = 0, b$. The functions $X_n(x)$ and $Y_m(y)$ are expressed as follows:

$$\begin{aligned} X_n(x) &= (\cos \alpha_n a - \cosh \alpha_n a)(\cos \alpha_n x - \cosh \alpha_n x) \\ &\quad - (\sin \alpha_n a - \sinh \alpha_n a)(\sin \alpha_n x - \sinh \alpha_n x) \end{aligned} \quad (6)$$

$$\alpha_1 a = 4.730, \quad \alpha_2 a = 7.853, \quad \alpha_3 a = 10.996, \dots$$

$$\begin{aligned} Y_m(y) &= (\cos \beta_m b - \cosh \beta_m b)(\cos \beta_m y - \cosh \beta_m y) \\ &\quad - (\sin \beta_m b - \sinh \beta_m b)(\sin \beta_m y - \sinh \beta_m y) \end{aligned} \quad (7)$$

$$\beta_1 b = 4.730, \quad \beta_2 b = 7.853, \quad \beta_3 b = 10.996, \dots$$

$$\begin{aligned} \omega_i &= \lambda_i \sqrt{\frac{D}{\rho h}} \lambda_i^2 - g^2, \quad \lambda_i^2 = \alpha_n^2 + \beta_m^2, \\ g^2 &= \frac{h(\rho \mu^2 - T)}{D}. \end{aligned} \quad (8)$$

This solution has the straight nodal lines shown in Fig. 3.

Measured Modes of Vibration

If the node shapes can be predicted accurately, the driving point method can be applied to increase the frequency separation between the lower (controlled) modes and the higher (uncontrolled) modes.

A modal test is conducted by exciting the plate on which white sesame is placed. The sinusoidal excitation force is the attractive force provided by one of the magnets. At the resonant frequency, the sesame is aligned with the nodal lines. The pictures of the 2.3 mm thick plate are shown in Fig. 4. The nodal curves are neither clear nor straight. For the 0.5 mm thick plate, the modes, which are not shown in any of the figures, are much more complicated. The reasons for this are believed to be:

1. the thinness of the plate;
2. the nonuniformity of the mass distribution;
3. the adverse effects of the air surrounding the plate on the modes of vibration.

We decided that modal control is not applicable and decided to adopt a simple PD control design and applied it to a collocated sensor and actuator arrangement.

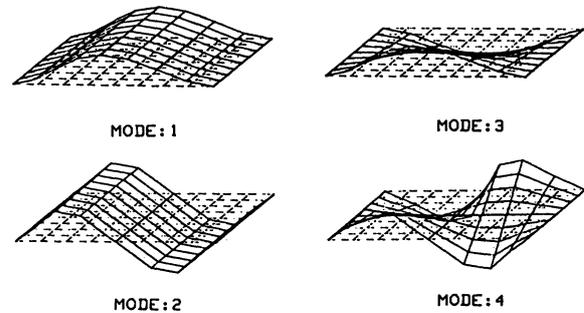


FIGURE 3 The theoretical vibration modes of the plate.



(a) 2nd (2,1) Mode, 30.5 Hz



(b) 3rd (1,2) Mode, 40 Hz



(c) 4th (2,2) Mode, 43.5 Hz

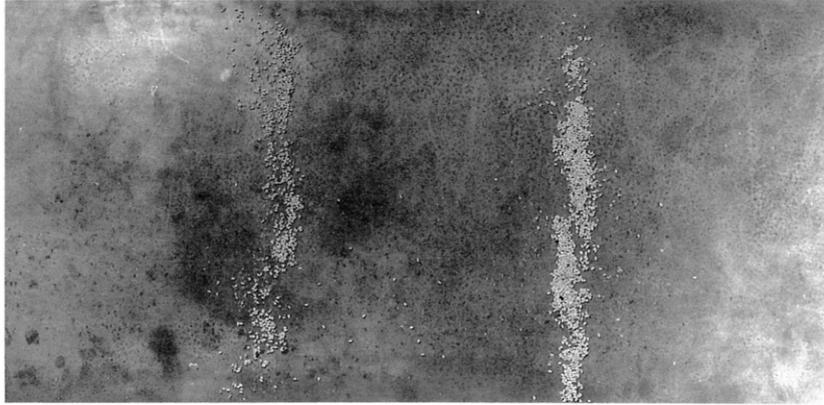
FIGURE 4 The experimental mode shapes of the clamped-free plate.

VIBRATION CONTROL

Individual PD control using two electromagnets was applied to control the plate vibration. The locations are $x_1 = 18$ cm, $y_1 = 5$ cm, $x_2 = 63$ cm, $y_2 = 30$ cm.

Control System

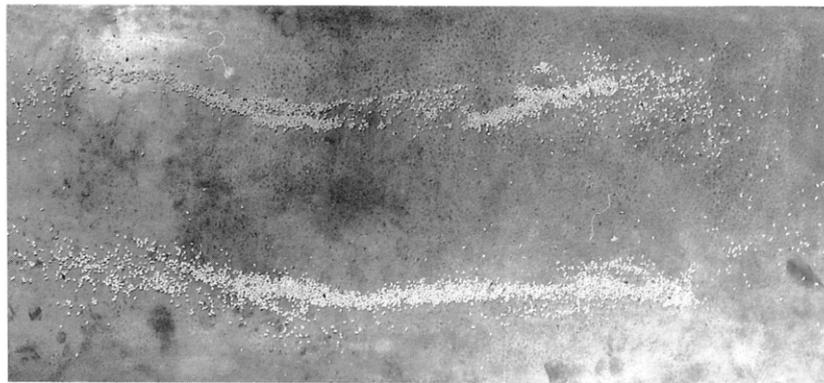
Two control systems were applied and tested. One was an analog PD controller with transfer function



(d) 5th (3,1) Mode, 58 Hz



(e) 6th (3,2) Mode, 76.5 Hz



(f) 7th (1,3) Mode, 133 Hz

$$G_c(s) = K_P + \frac{K_D s}{1 + T_D s} \quad (9)$$

where K_P , K_D and T_D are the proportional gain, derivative gain, and derivative time constant, respectively. This is realized with an RC circuit.

Another controller was a digital PD controller with transfer function

$$G_c(z) = K_P + \frac{K_D(z - 1)}{T_D(z - e^{-\tau/T_D})} \quad (10)$$

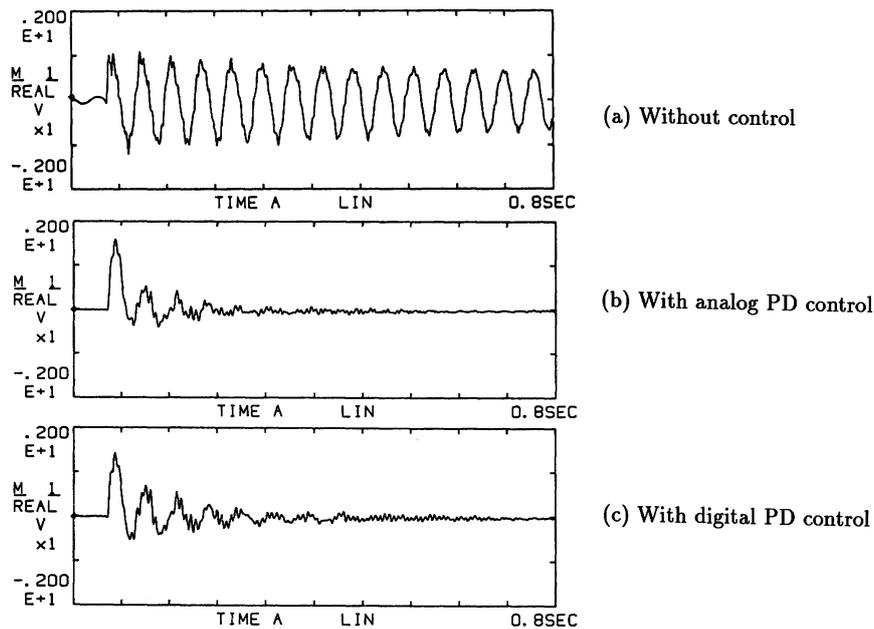


FIGURE 5 The impulse response of the 2.3 mm plate.

where τ is the sampling interval ($\tau = 0.08$ ms). This is realized on a transputer (KTR-B02, KTR-B02) with a parallel C program.

Experimental Impulse Responses

The impulse response tests were conducted by dropping a plastic ball onto the plate and record-

ing the decaying responses. The responses of the 2.3 and 0.5 mm-thick plates are shown in Figs. 5 and 6, respectively. In these figures, (a) indicates that no feedback control was applied; (b) indicates that analog PD control was used; (c) indicates that digital PD control was used. The vibrations were reduced well by both types of PD control. The analog controller gave better results

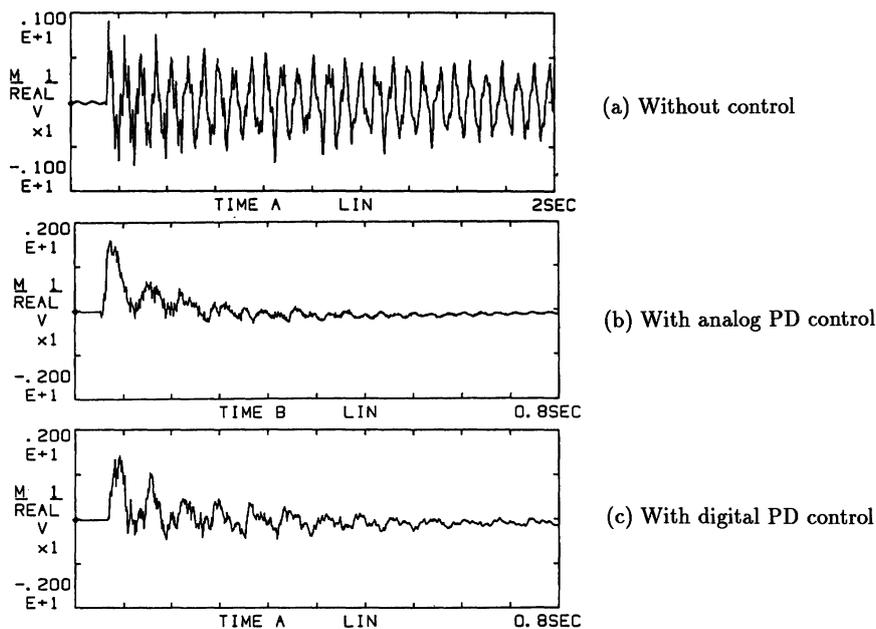


FIGURE 6 The impulse response of the 0.5 mm plate.

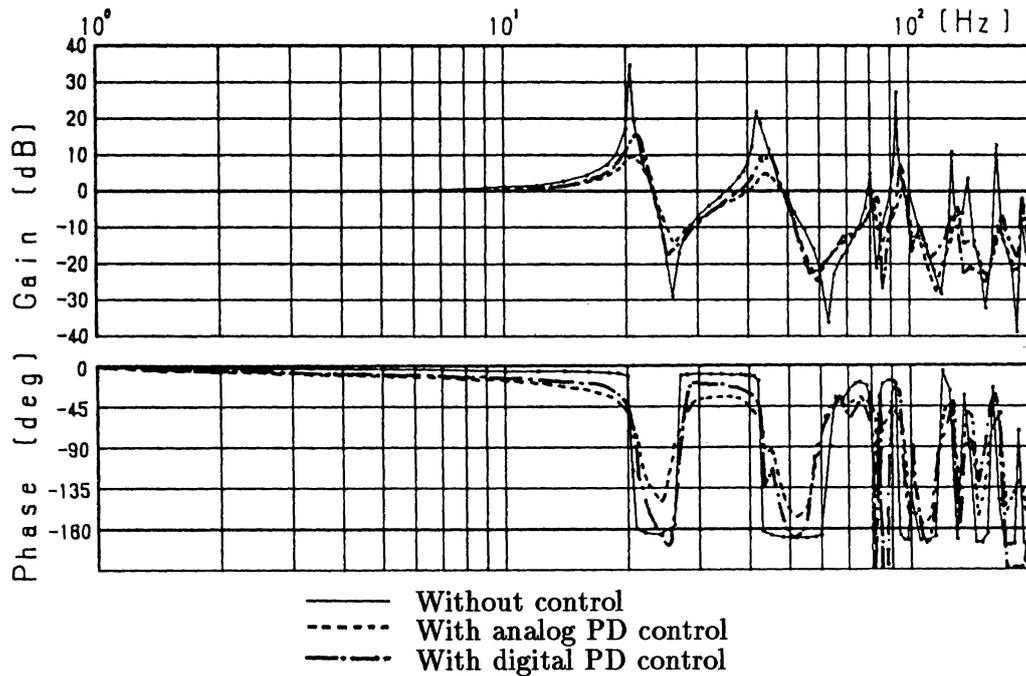


FIGURE 7 The frequency response of the 2.3 mm plate.

than the digital controller. However, higher vibration modes were difficult to suppress. This may be attributed to the complexity of the mode shapes for the thin plate and the tendency of the magnetic flux to saturate.

Experimental Frequency Response

The frequency response test was also conducted by adding a sinusoidal signal to the magnetic power amplifier and recording the response. In

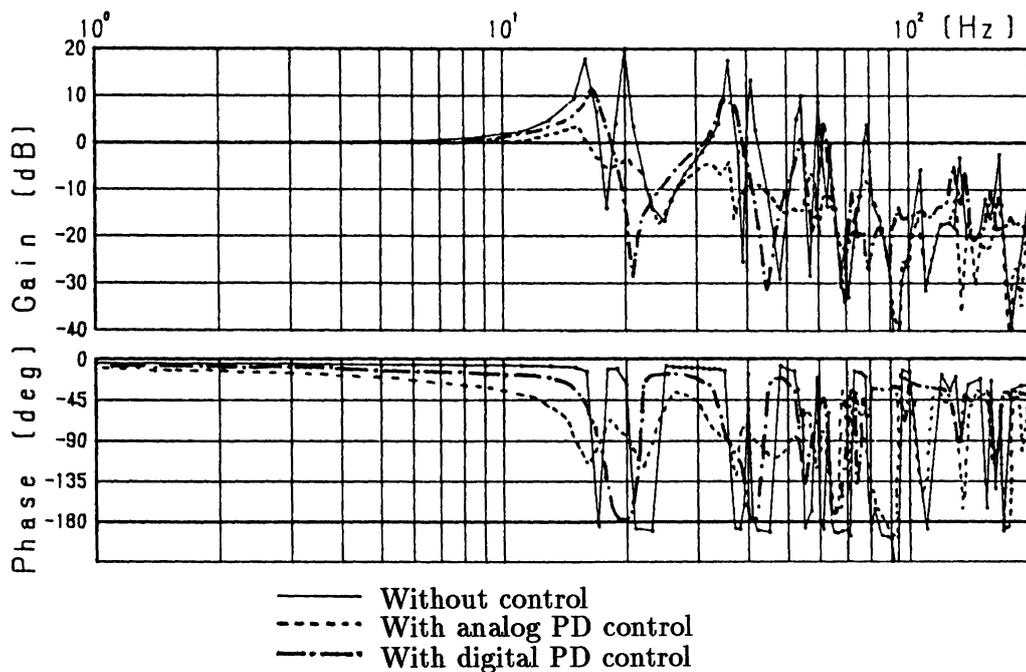


FIGURE 8 The frequency response of the 0.5 mm plate.

this case, an electromagnet was used as a forcing actuator as well as an active damping device. The frequency responses of the 2.3 and 0.5 mm plates are shown in Figs. 7 and 8, respectively. The peaks between 10–200 Hz are well reduced by the PD feedbacks. However, the higher frequency modes were very difficult to control and sometimes they became unstable.

CONCLUSIONS

A noncontact method of active vibration control for reducing the bending vibrations of a travelling steel sheet during the rolling process was developed and tested. An attempt was made to use modal control in designing the control system. It was found, however, that the complexity of the modes of vibration makes modal control impractical. Analog and digital PD controllers with a collocated arrangement of sensors and actuators proved effective in reducing the vibrations of the travelling plate. Further work is underway to improve the results and to test the method in a real rolling process.

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