The Frequency Response of Vibration Control with a Linear Actuator and a Permanent Magnet

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This paper presents the frequency response of a vibration control system. The system prototype comprises a linear actuator, a controller, sensors and permanent magnets. The reduction of vibrations and deformations of thin steel sheets was accomplished by controlling an air-gap between the permanent magnets and the steel sheets. The feedback control system was designed using the Linear Quadratic Regulator (LQR) method. The design of the controller and the frequency responses of the proposed vibration suppression mechanism have been performed through both numerical analyses and experiments. A sinusoidal signal was employed as a continuous disturbance to the system prototype in which different conditions of disturbance were applied by varying the signal frequency from 3 to 60 rad/sec. The frequency responses shown in the Bode diagrams show that the magnitude of the vibration body displacement in the case without feedback control is higher than the case with feedback control. In conclusion, feedback control has been applied successfully, and the proposed vibration suppression mechanism is effective for this vibration control system.

Keywords: vibration control, frequency response, permanent magnet, linear actuator, linear quadratic regulator.
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1. Introduction

In the process of plating, coating and rolling steel sheets, vibration during conveyance often leads to problems with deformation, peeling and nonuniform products because the sheets are very flexible. As a result, a vibration suppressor with mechanical contacts is not suitable as a countermeasure. Objects are easily damaged due to their material makeup when they have just been rolled, coated or plated. Therefore, a noncontact suppression mechanism is more suitable for minimizing deformation of the steel sheets. Noncontact vibration control methods that use the attractive force of electromagnets have already been proposed [1,4]. 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 the air-gap length. If the vibration amplitude of the object is large, minimizing deformation by means of electromagnets is ineffective for control of the object. In contrast, a control method using permanent magnets located at the desired iron sheet passage has also been proposed [5]. However, this method uses passive control and cannot be successfully suppressed under large disturbances.

A noncontact suspension system using permanent magnets and linear actuators has been proposed [6]. The key element of the design is the force control mechanism. A linear actuator drives a permanent magnet and varies the air-gap between the magnet and the object.

Variation of the size of the air-gap changes the attractive force. Here the vibration control system uses this type of force control mechanism for vibration suppression because the control range is almost the same as the actuator stroke length. In this system, vibration control range is expected to be correspondingly wide.

In this paper, the vibration suppression mechanism and the frequency response of the proposed vibration control systems were studied. The structure of the vibration suppression system is introduced and the aim of the system is shown. The principle of the control mechanism is explained and experimental examinations are conducted to demonstrate the frequency response and feasibility of the control method.

2. Vibration Suppression Mechanism

A schematic diagram of a vibration suppression mechanism for a plating or coating process is shown in Fig.1. The steel sheet is fed from the right side of the figure and is directed upward by a roller. While the steel sheet is being fed into the solution bath, plating or coating is conducted. After the plating process is completed, the steel is seasoned or cooled in a vertical feed. During the seasoning process, the steel is especially sensitive to deformation. Consequently vibration control in the seasoning process is very important.

In this paper, a vibration suppression mechanism, as shown in Fig. 1 is proposed. The mechanism consists of two permanent magnets, two linear actuators, sensors that measure the steel sheet displacement and a controller. The sensors and controller are omitted in the figure. The intention of the system is to reduce vibration caused by the roller feed mechanism, air wipers, and high-
permanent magnet can suppress vibration and deformation of the object.

3. Experimental Device

Figs. 1 and 2 represent the vibration suppression mechanism and the method of controlling the vibration. This paper represents a new design based on a similar operating mechanism, and a block diagram of the system is shown in Fig. 3. An experimental system to examine the performance of the proposed vibration control mechanism was manufactured. A photograph of the system is shown in Fig. 4. The system consists of a vibration body, two permanent magnets installed on a slider, a voice coil motor (VCM) and sensors. The vibration body is structured as a parallel spring made by phosphor bronze with steel plates on both ends facing the permanent magnets. Two permanent magnets are installed on the slider driven by the VCM. Two sensors are installed for measuring the displacements of the vibration body and the slider. A laser sensor is used for the motion of the vibration body and an eddy current sensor is used for the motion of magnets, which is the same as the VCM movement.

The control system block diagram is shown in Fig. 3. As seen in the figure, the controller uses a digital signal processing (DSP). It calculates the generating force of the VCM from two sensor signals and the output is the current signal for the filter. The output of the filter is the current signal for the amplifier. The amplifier changes the signal to the current for the VCM and the VCM changes the currents to the force proportionally. For DSP programming, MATLAB with SIMULINK and Pass/C67 was used.

In this paper, the system was modeled to analyze and adopt the linear control theory to synthesize the control system and the frequency response. For modeling, the specifications should be determined. The relationship between the attractive force of the permanent magnet, the force of the vibration body and the resultant force from displacement of a permanent
magnet and a steel plate on the vibration body set to 20 mm is shown in Fig. 5. As shown in the figure, the relationship between the attractive force of the permanent magnet and the displacement is nonlinear.

The disturbance force was set to a sinusoidal function by a dc motor drive, as shown by the dashed line in Fig. 4. The relationship between the voltage of the dc motor and the disturbance frequency is shown in Fig. 6.

4. System Modeling

In the model, the motion of the vibration body, as supported by a parallel spring is assumed to be translational. An illustration of the model is shown in Fig. 7.

The symbols used in the model are as follows: $z_i$ and $z_p$ are the displacements of the vibration body and the permanent magnet, $d_l$ and $d_r$ are the air gap length between the left and right permanent magnets and the vibration body, $d_0$ is the air-gap length when the vibration body is centered between the magnets, $f_{m1}$ and $f_{m2}$ are the attractive forces of the left and right permanent magnets, $f_m$ is the resultant force of the magnets, $f_{vcm}$ is the actuator generating force, $f_d$ is the disturbance force, $c_i$ and $c_p$ are the damping coefficients of the vibration body and the permanent magnet, $k_i$ is the parallel spring constant, $k_p$ is the permanent magnet constant, $m_i$ is the equivalent mass of the vibration body, and $m_p$ is the mass of the permanent magnet together with the voice coil motor (VCM) and the slider. The specification values of the prototype are indicated in Table I

The resultant force is assumed to be inversely proportional to the square of the gap length.

$$f_m = \frac{k}{(d_l - z_i + z_p)^2} = \frac{k}{(d_r + z_i - z_p)^2},$$

where $k$ is constant and the motion equation of the vibration body is

$$m_i \ddot{z}_i = -k_i z_i - c_i \dot{z}_i + f_{m1} + f_d.$$  

The motion equation of the slider is

$$m_p \ddot{z}_p = -k_p z_p - c_p \dot{z}_p + f_{vcm} - f_m.$$  

The inputs of the system are defined as the forces of the actuator (VCM) $f_{vcm}$ and the disturbance $f_d$. The outputs are the displacement of the vibration body $z_i$ and
the slider \( z_p \). The model can be represented by Eqs. (1), (2) and (3).

By linearization of the attractive force of the magnets, the resultant force can be represented by

\[
\tilde{f}_m = k_n(z_i - z_p),
\]

where \( k_n \) is a constant value and Eqs. (2) and (3) become

\[
m_i \ddot{z}_i = -k_p z_i - c_i \dot{z}_i + k_n (z_i - z_p) + f_j,
\]

\[
m_p \ddot{z}_p = -k_p z_p - c_p \dot{z}_p + f_m - k_n (z_i - z_p),
\]

The system can be represented by the block diagram shown in Fig. 8. There are PD controllers in the loops of the vibration body and the magnets. The feedback gains are calculated by means of the LQR control theory. Using the LQR method, a state space model can be derived from Eqs. (4) – (6) as

\[
x' = Ax + Bu,
\]

\[
y = Cx,
\]

where \( u = f_m \), \( y = [z_i, z_p]^T \), and the state vector \( x = [z_i, \dot{z}_i, z_p, \dot{z}_p]^T \).

We examined the two feedback system LQR1 and LQR2. The feedback system LQR1 uses the weighting matrix \( Q_1 \) of

\[
\begin{bmatrix}
1 \times 10^4 & 0 & 0 & 0 \\
0 & 1 & 0 & 0 \\
0 & 0 & 1 \times 10^4 & 0 \\
0 & 0 & 0 & 1
\end{bmatrix}
\]

LQR2 uses the matrix \( Q_2 \) of

\[
\begin{bmatrix}
1 \times 10^6 & 0 & 0 & 0 \\
0 & 1 & 0 & 0 \\
0 & 0 & 1 \times 10^6 & 0 \\
0 & 0 & 0 & 1
\end{bmatrix}
\]

Table 1 Parameters of the vibration control system.

<table>
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<tr>
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<th>unit</th>
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<tbody>
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<td>( c_i )</td>
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<td>Ns/m</td>
</tr>
<tr>
<td>( c_p )</td>
<td>0</td>
<td>Ns/m</td>
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<td>N/m</td>
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<td>( k_n )</td>
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<tr>
<td>( m_i )</td>
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</tr>
<tr>
<td>( m_p )</td>
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</tr>
<tr>
<td>( k_m )</td>
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<td>N/m</td>
</tr>
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</table>

Fig. 9. Displacement signals of the vibration body with the disturbance frequency set to 3 rad/sec and the following system conditions: 1) without feedback control 2) with feedback control (LQR1) and 3) with feedback control (LQR2).

Fig. 10. Displacement signals of the vibration body with the disturbance frequency set to 36 rad/sec and the following system conditions: 1) without feedback control 2) with feedback control (LQR1) and 3) with feedback control (LQR2).

5. Simulation Results

Simulation results were obtained with a nonlinear force acting on the vibration body. The frequency response of the vibration suppression was recorded at frequency disturbance of 3, 36, 57 and 300 rad/sec. The conditions of the simulations were as follows: 1) without feedback control and 2) with feedback control. In the case with feedback control, the controller was designed using the LQR method with the weighting matrices for calculating the feedback gain set to \( Q_1 \) for LQR1 and \( Q_2 \) for LQR2.

According to the simulation results for frequency disturbances of 3, 36, 57 and 300 rad/sec as shown in
Fig. 11. Displacement signals of the vibration body with the disturbance frequency set to 57 rad/sec and the following system conditions: 1) without feedback control 2) with feedback control (LQR1) and 3) with feedback control (LQR2)

Fig. 12. Displacement signal of the vibration body with the disturbance frequency set to 300 rad/sec and the following system conditions: 1) without feedback control 2) with feedback control (LQR1) and 3) with feedback control (LQR2)

Figs. 9, 10, 11 and 12 the weighting matrices for LQR1 and LQR2 were set to Q₁ and Q₂. In the case in which the weighting matrix of the LQR method was set to Q₁, it should be noted that, at a low frequency, the amplitude of the displacement signal from the vibration body in the system with feedback control was higher than that without feedback control. At a high frequency, the difference between the amplitude in the systems with feedback control and without feedback control was small.

In the case in which the weighting matrix of the LQR method was set to Q₂, the performance of the feedback control is shown in Figs. 9, 10, 11 and 12. From these results, it can be observed that the amplitude of the displacement signal from the vibration body in the system with feedback control is smaller than that without feedback control.

Fig. 13 shows the Bode diagram of the vibration control system when the conditions were as follows: 1) without feedback control and 2) with feedback control when the weighting matrices are set to LQR1 and LQR2.

6. Experimental Results

Vibration control experiments were also performed using the experimental prototype in Fig. 4 under both conditions. To study the frequency response, trials were conducted under 2 conditions: 1) without feedback control and 2) with feedback control. The frequency of the disturbance was set to 3, 36 and 57 rad/sec as shown in Figs. 14, 15 and 16. From the results, the feedback control was designed using the LQR method, with the weighting matrices for calculating the optimal gain set to Q₂ and the Block-diagram of the systems shown in Fig. 8.

In Figs. 14, 15 and 16, which show the experimental results for disturbance frequencies of 3, 36 and 57 rad/sec, the line indicates displacement of the vibration body in the case without feedback control and with feedback control. As a result, in the case without feedback control, the amplitude of the displacement signal from the vibration body was found to be higher than that with feedback control. However, the feedback control designed using the LQR method did not completely suppress the disturbance displacement.

From these results, it was observed that the amplitudes of the displacement signal from the vibration body in the system with feedback control were lower than that without feedback control.

7. Conclusions

A frequency response-based feedback control scheme for a vibration suppression system was proposed in this paper. A linear actuator and a permanent magnet are the key elements of the control scheme design. The feedback control was designed using the LQR method. A model was set up for the feasibility analysis, and a system prototype was manufactured for experimental confirmation. The experimental results from the analysis
Fig. 14. Displacement signals of the vibration body with the disturbance frequency set to 3 rad/sec and the following system conditions: 1) without feedback control and 2) with feedback control.

Fig. 15. Displacement signals of the vibration body with the disturbance frequency set to 36 rad/sec and the following system conditions: 1) without feedback control and 2) with feedback control.

Fig. 16. Displacement signals of the vibration body with the disturbance frequency set to 57 rad/sec and the following system conditions: 1) without feedback control and 2) with feedback control.

of the prototype performance demonstrated the efficiency of the feedback control design.

Further research is required to improve the controller effectiveness in the experimental design by applying optimal LQR weighting matrices.

References


