Paper:

Filter Design of Adjusting Common Phase for Vibration Suppression Control of Multi-Degree-of Freedom System

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In flexible structures such as a stacker crane, vibration is generated by the high acceleration of the traveling axis in the load transportation slider. Vibration suppression is necessary for productivity improvement. We utilized the acceleration feedback control for vibration suppression. The flexible structure has several natural vibration modes of varying phase, and vibration suppression is difficult by the simple feedback control. The second mode is excited by the simple feedback control for the vibration suppression of the first mode when the phase between the first mode and the second mode are different, e.g., a flexible structure. This paper proposes a filter design by adjusting the common phase for the feedback control. The filter using the operational amplifier reverses the phase of the second mode and such that it is the same as the first mode. The effect of vibration suppression is verified by the experiment.

Keywords: mechatronics, flexible structure, vibration suppression, filter design

1. Introduction

Stacker cranes are used for baggage transportation in many factories or warehouses [1, 2]. The necessary floor space for the setting is small, and the stacker crane has the advantage of a high affinity in environments such as factories. High speed and high precision of the stacker crane are important for productivity improvement. The mast of the stacker crane is 10 m high, and it is a flexible structure. The residual vibration occurring in the mast when the traveling axis stops is a problem. When the residual vibration occurs, the positioning precision becomes worse, and the productivity decreases.

A suppression method of the residual vibration includes a method to strengthen the crane rigidity and a brake utilization method. By the rigidity strengthening method, the weight of the crane increases, and the durability and cost become a problem. Using the method with brakes, the consumption cost becomes a problem. Therefore, it is effective to suppress the residual vibration by a control without improving the crane mechanism.

From the vibration suppression by the control, a method to improve a motion reference is suggested. The vibration can be suppressed using several methods. The first is a method of forcibly swinging the mast by changing the running speed by performing acceleration/deceleration when the operation stops [3]. The second is a method of setting the speed increase/decrease at a time that is an integral multiple of two times or more of the natural vibration period during deceleration [4]. The third is a method to reduce the vibration by causing the object to collide [5]. In addition, an input shaping method that suppresses the vibration of a flexible robot arm has been proposed [6, 7]. In the method of installing a dynamic vibration reducer on a tower structure, the actuator and the sensor arrangement are considered in view of the vibration mode [8–11]. Although a method of attaching the dynamic vibration reducer at a predetermined position is effective, if vibration can be suppressed without attaching the device, subsequently, it is an effective method.

In the vibration suppression method by improving the motion reference aiming for high speed, methods of final-state control [12], and minimum jerk position control [13] have been proposed; further, a two-degree-of-freedom control has been established as a feedforward reference for obtaining a high-speed response. As these references require calculations using the cost function, the calculation is complicated.

Methods of reference shaping by adding a bang-bang controlled motion in addition to the motor speed exist according to the operation pattern of the stacker crane [14, 15]. In reference shaping, an acceleration/deceleration speed reference for vibration suppression can be provided according to the first natural frequency of the mast part. Incidentally, the reference is set through the low-pass filter such that the excitation is not by the second natural
frequency. When the state of the conveyance mass and the mast height changes, it is necessary to change the reference accordingly, which poses a problem in practical use.

In this research, we aim to suppress the residual vibration of the mast part by the feedback control of the acceleration of the flexible structure. The feedback control for measuring the acceleration of the machine end of the two-inertia system has yielded an effective vibration suppression effect [16–19].

We herein propose a feedback control of three inertia systems using a filter that creates the same vibration phases [20]. The target structure has three vibration modes with different phases, and vibration cannot be suppressed by the simple acceleration feedback control. Therefore, we propose to insert a filter to adjust the phase delay of each mode in the feedback loop, thereby suppressing the vibration by a simple acceleration feedback control by adjusting the phase of the vibration suppression. We herein propose a filter that adjusts the phases of all vibration modes and suppresses vibration by the feedback control of the acceleration. The vibration suppression effect is verified experimentally.

2. Controlled Object

The flexible structure used in this study is shown in Fig. 1.

The mass is fixed at the tip of the two cantilever beams. An acceleration sensor is attached to the tip. When the AC servomotor is driven, the ball screw rotates, and the structure translates. Using an FFT (Fast Fourier Transform) analyzer, the frequency response of the flexible structure is measured. The input voltage to the AC servomotor of the ball screw is the input signal, and the output voltage of the acceleration sensor attached to the tip of the mast part is the output signal. White noise is provided as the input voltage and the frequency response is measured. The measured frequency response is shown in Fig. 2.

The vertical axis of the graph describes the output acceleration [m/s²] for the reference voltage of 1 V in dB notation. Here, 1 V represents the rated torque. The flexible structure has three natural vibration modes. The three natural frequencies are 3 Hz, 19.2 Hz, and 51.8 Hz. At the natural frequencies of the frequency response shown in Fig. 2, the numerical value of the phase lag is read; the phases of the first, second, and third modes are $-230^\circ$, $-378^\circ$, and $-664^\circ$, respectively.

Flexible structure modeling for designing the filter is performed. The transfer function of the flexible structure is expressed by Eq. (1).

\[
G_s(s) = \frac{b_0 + b_1s + \cdots + b_6s^6}{a_0 + a_1s + \cdots + a_6s^6} \quad \ldots \ldots (1)
\]

Because it is modeled up to the third vibration mode, it is expressed as a transfer function of the sixth order, and the coefficient is estimated by the least-squares method. The transfer function is separated into three vibration modes. The separation result is shown in Eq. (2). The second term on the right side represents the third mode, the third term represents the second mode, and the fourth term represents the first mode.

\[
G_s(s) = \frac{0.44364 - 27762.567 + 13.63092s}{105580.93 + 2.5s + s^2}
+ \frac{12141.759 + 34.73518s - 32.172 - 8.18422s}{14538.57 + 2.74s + s^2}
+ \frac{355.3 + 0.93s + s^2}{363092} \quad \ldots \ldots (2)
\]

The estimated frequency response is shown in Fig. 3. The solid line represents the measured frequency response. The dashed line represents the estimated frequency response. The phases of the first, second, and third modes are $-193^\circ$, $-433^\circ$, and $-622^\circ$, respectively. Both patterns coincide. In the vicinity of the natural frequency, the amplitude and the phase are estimated.

The relationship of the phase is the same as in the measured frequency response, and it can be modeled as in Eq. (2).

The frequency response for each vibration mode is modeled by Eq. (2). When each mode response is calculated from Eq. (2), the phases of the first, second, and third modes are $187^\circ$, $-65^\circ$, and $84^\circ$, respectively. The phases of the first and third modes are positive values and are defined as the same phase mode. The phase of the second mode is a negative value and is defined as the different phase mode.

Fig. 1. Flexible structure.

Fig. 2. Measured frequency response.

Fig. 3. Estimated frequency response.
3. Design of Proposed Filter

In the vibration mode of the flexible structure, the phases of the first and third mode are same, whereas the phase of the second mode is different. Thus, if the same phase mode and the different phase mode coexist in the same system, it is impossible to suppress all vibrations with a simple feedback control system as shown in Fig. 4. When negative feedback is performed, the vibration mode of the same phase can be suppressed, but the vibration mode of the different phases is excited. On the contrary, if positive feedback is performed, the vibration mode of the different phase can be suppressed, but the vibration mode of the same phase is excited.

In the measured frequency response, we found that the gain of the second mode is larger than that of the first mode and third mode. Therefore, when feedback control is performed, the vibration suppression effect is not sufficient in the first mode and third mode. If the feedback gain is increased to increase the effect, the gain of the second mode becomes too large and unstable. These two problems are solved by inserting the common phase filter $G_f(s)$ in the feedback loop. The feedback control system with the filter insertion is shown in Fig. 5, where $K$ represents the feedback gain.

At this time, the filter is designed such that the open loop transfer function $G_f(s)G_f(s)$ satisfies the following two conditions:

1. The phases of all vibration modes are the same.
2. The loop gains of all vibration modes are equal.

The design objective is to adjust the phase of the second mode to the same phase as the phases of the first and third modes by inverting the phase of the second mode. We herein describe the design method to achieve the design objective.

The structure of the filter $G_f(s)$ to be designed is shown in Fig. 6. First, to separate the signals of each vibration mode from the output signal, the bandpass filters are designed to pass only the natural frequencies. At this time, the bandpass filters corresponding to the respective vibration modes are set to $G_{fi}(s)$ ($i=1,2,3$).

Next, these signals are weighted and added together. $a_i$ is a coefficient for determining the weight of each vibration mode. The phase of each vibration mode of $G_f(s)G_f(s)$ is adjusted by setting the coefficients $a_i$ of the mode in which the phase is to be inverted to negative and other coefficients $a_i$ are set to positive. Further, by changing the magnitude of the coefficient $a_i$, the gain of each vibration mode of $G_f(s)G_f(s)$ becomes equal.

The overall filter circuit is shown in Fig. 7. The signal of the acceleration sensor is passed through the voltage follower and subsequently separated by each natural frequency by the bandpass filters. In the second mode frequency band, the phase is delayed by the inverting amplifier. $R_1$, $R_2$, and $R_3$ are the variable resistors, and the loop gain of each mode is adjusted. Finally, the signals are summed by the adder circuit.

The circuit diagram of the bandpass filter is shown in Fig. 8. For the three modes, because it is easy to set the center frequency of the bandpass filter, the multiple-feedback-type is used.
The transfer function of the bandpass filter is shown in Eq. (3).
\[
\frac{V_{\text{out}}}{V_{\text{in}}} = \frac{-1}{s^2 + \left(\frac{1}{R_3C_2} + \frac{1}{R_3C_1}\right) + \frac{1}{R_1C_1C_2}\left(\frac{1}{R_1} + \frac{1}{R_2}\right)}
\]  

The center frequency of the filter is shown in Eq. (4).
\[
f = \frac{1}{2\pi} \sqrt{\frac{1}{R_3C_1C_2}\left(\frac{1}{R_1} + \frac{1}{R_2}\right)}
\]

The feedback gain at this frequency is shown in Eq. (5).
\[
K = \frac{R_3}{R_1} \cdot \frac{-C_2}{C_1 + C_2}
\]

For each filter, the values of \( R_1, R_2, R_3 \) and \( C_1, C_2 \) are determined such that 3 Hz, 19.2 Hz, 51.8 Hz would be the center frequencies. The resistances and the capacitors of the circuit are shown in Table 1.

These values are substituted into Eq. (3), and the transfer function of the bandpass filter is obtained.
\[
G_{f1}(s) = \frac{-38.0s}{s^2 + 39.2s + 371.4}
\]
\[
G_{f2}(s) = \frac{-262.6s}{s^2 + 247.6s + 15638.2}
\]

The feedback gain at this frequency is shown in Eq. (5).
\[
K = \frac{R_3}{R_1} \cdot \frac{-C_2}{C_1 + C_2}
\]

For each filter, the values of \( R_1, R_2, R_3 \) and \( C_1, C_2 \) are determined such that 3 Hz, 19.2 Hz, 51.8 Hz would be the center frequencies. The resistances and the capacitors of the circuit are shown in Table 1.

These values are substituted into Eq. (3), and the transfer function of the bandpass filter is obtained.
\[
G_{f3}(s) = \frac{-644.7s}{s^2 + 683.3s + 110420}
\]

The transfer function of the whole filter is shown in Eq. (9).
\[
G_f(s) = -\left\{\frac{R_f}{R_1}G_{f1}(s) - \frac{R_f}{R_2}G_{f2}(s) + \frac{R_f}{R_3}G_{f3}(s)\right\}
\]

In the open-loop transfer function \( G_s(s)G_f(s) \), the value of Eq. (10) is set such that the gains of the respective modes are equal.
\[
\frac{R_f}{R_1} = 1.3, \quad \frac{R_f}{R_2} = 0.5, \quad \frac{R_f}{R_3} = 0.53
\]

The frequency response of \( G_f(s) \) is shown in Fig. 9. Because the loop gains of the respective modes are equalized, the gain near the second frequency is low. In the vicinity of the frequency of the second mode, the phase is inverted.

The frequency response of the open-loop transfer function is shown in Fig. 10. The gains of the three modes in \( G_s(s)G_f(s) \) are 20 dB.

The phase of the second mode changes by \(-197^\circ\), i.e., from \(-378^\circ\) to \(-575^\circ\).

The loop gain of the open loop is set to the same amplitude from the first mode to the third mode, and the phases of the first mode and the second mode can be equalized. The proposed filter can thus be designed.
A rectangular signal of width 12.5 ms is used as the impulse signal. The feedback gain is set to a common $K = 0.5$. The impulse responses of the four systems are shown in Figs. 12–15.

To evaluate the vibration suppression effect, the absolute value of the response is integrated. The results of integration are shown in Table 2.

When using the filter, the amplitude is reduced to 42% owing to the vibration suppression. With the feedback control without the filter, the vibration suppression effect is only 9% or 2%.

Next, the frequency of the impulse response is analyzed. The vibration suppression effect is confirmed for each vibration mode. The spectral amplitudes in the four systems are shown in Figs. 16–19.

Figure 17 shows that the amplitude peak of the second mode and third mode are reduced. With the filter, we can suppress the second mode without affecting the first mode.

In Fig. 18, although the peak of the second mode amplitude decreased, the peak of the first mode amplitude increased. Although the amplitude of the second mode is decreased, the amplitude is increased by exciting the amplitude of the first mode. In Fig. 19, the peak of the first mode decreases, but the peak of the second mode remains high. Although the amplitude of the first mode is decreased, the amplitude of the second mode is excited.

5. Conclusions

In a flexible structure imitating a stacker crane, we proposed a filter that suppresses vibrations in a multi-degree-of-freedom system with three vibration modes of different phases. The following conclusions were obtained:

1) From the measurement result of the frequency response, the transfer function of the flexible structure was identified as the sixth-order equation by the least-squares method. The phases of the first and third modes are positive values and are defined as the same phase mode. The phase of the second mode is a negative value and is defined as the different phase mode.
2) The filter that causes the vibration phases to adjust the common phase was proposed, the amplitude of the open loop transfer function was aligned, and the phases of the first mode and second mode were aligned in the same phase.
3) The feedback control system using the common phase filter was constructed.
4) The experimental system was constructed, and responses were measured by an impulse signal provided as the torque reference. Consequently, we found that the feedback control using the proposed filter is effective in the flexible structure.


**Table 2.** Effect of vibration suppression.

<table>
<thead>
<tr>
<th>System</th>
<th>Reduced rate of amplitude</th>
<th>Reference figure</th>
</tr>
</thead>
<tbody>
<tr>
<td>System without feedback control</td>
<td>100%</td>
<td>Fig.12</td>
</tr>
<tr>
<td>Positive feedback system with proposed filter</td>
<td>41.8%</td>
<td>Fig.13</td>
</tr>
<tr>
<td>Negative feedback system without filter</td>
<td>91.2%</td>
<td>Fig.14</td>
</tr>
<tr>
<td>Positive feedback system without filter</td>
<td>98.2%</td>
<td>Fig.15</td>
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</table>

**References:**


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