Relative Vibration Suppression in a Positioning Machine Using Acceleration Feedback Control

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A method to suppress the residual vibration of relative displacement is proposed. In factory automation, the positioning machines are precisely controlled by servo motors; however, their reaction force will unavoidably vibrate the machine stand on which the motors are mounted. A common problem is that the resulting residual vibrations cause disturbances that prevent high speed and accurate positioning, due to the inherent flexible structures in the machine stand, thereby unexpectedly extending the settling time in some industrial applications. This paper presents machine stand acceleration feedback (MSAFB) control, which detects the acceleration signal of the machine stand using an accelerometer to improve the robustness against disturbances from the environment and uncertainty in the system. This feedback strategy is superior to input shaping strategy, which is based on feedforward control. Simulations and experiments show the effectiveness of the proposed MSAFB controller, which was designed specifically for mechanical plants with rocking mode vibration.

Keywords: servo motor, acceleration feedback, vibration control, rocking mode, machine stand, disturbance localization

1. Introduction

Servo motors are widely used in machines that must perform accurate and repeated movements, such as semiconductor manufacturing equipment, chip shooters, industrial robots, and food processing machines (3). We now stand poised at the next generation of machine development, so high demand exists for positioning to speed-up and to improve the positioning accuracy in order to reduce the cycle time of the manufacturing process, which is called takt time. The settling time of the machines has to be shortened during the positioning without degrading the accuracy.

In general system configurations, servo amplifiers provide electrical current to the servo motors, which must be controlled to follow positioning commands. Command profiles are programmed by programmable logic controllers (PLCs), motion controllers, or pulse drivers. Installed rotary encoders or linear scales detect angular or linear positions to constitute position and velocity feedback loops to improve disturbance suppression ability.

In general machine configurations, a mover of the motor is connected to positioning objects, while a stator is rigidly fixed on a machine stand that is used as a base platform. However, when the action force is generated between the mover and the stator, its reaction force causes vibrations of the machine stand at low frequency, e.g., a rocking mode vibration (2,3). This phenomenon mostly stems from the inherent flexibility and low stiffness of the columns or leveling bolts that support the machine in keeping it standing flat against the floor.

One well-established strategy to suppress the resulting residual vibrations of the relative position between the machine stand and the mover is input shaping control (ISC) (4–6). This useful strategy is effective when applied in suppressing vibrations at the tip of a connected mechanism. The ISC can reduce residual vibration in an entire machine and can mostly shorten the takt time. However, two issues remain with the ISC:

[1] The settling time is sometimes extended, especially in short stroke positioning because the delay in the commands becomes relatively large.

[2] Robustness against disturbances from the environment and uncertainty in the system is not guaranteed because the ISC is designed with a feedforward-based approach.

In this paper, we propose machine stand acceleration feedback (MSAFB) to overcome the aforementioned issues. As is discussed in previous studies (7–9), MSAFB can suppress the residual vibration of relative displacement between the mover and the stator, which is usually measured by position encoders. Nevertheless, in practical industrial applications, because the positioning objects come equipped with adsorption mechanisms, e.g., robot hands and adsorption nozzles, the vibrations of relative displacement between the tip of the positioning objects and the machine stand need to be suppressed. The latter relative displacement is not precisely measurable without additional high-cost displacement sensors. Therefore, we use the MSAFB control to keep the latter relative displacement constant in a steady state and to improve
the robustness. The acceleration feedback control including MSAFB is known as a representative vibration suppression strategy in a variety of fields \(^{(10)-(13)}\). However, few applications are currently used in factory automation \(^{(14)-(16)}\). This paper presents an MSAFB controller designed specifically for a rocking mode vibration system. We evaluated the effectiveness of the proposed MSAFB control via simulations and experiments.

The rest of this paper is organized as follows. In Section 2, we show a mechanical plant of the rocking mode vibration system and the control structure with the proposed part. Section 3 describes a simulation of the positioning waveforms to determine the effects of suppressing the relative vibrations and reducing the settling time. Section 4 shows the experimental results. The conclusions are given in Section 5.

2. Control Structure

A model of the mechanical plant, a block diagram of the control system, and the design method of MSAFB controller for vibration suppression of the relative displacement are shown and described.

2.1 Plant Machine

The overview of the positioning machine is shown in Fig. 1. In Fig. 1(a), a linear motor has been horizontally utilized for simplicity of analysis. The stator of the motor is mounted on the top of the machine stand while the mover is connected to the tip of a positioning object with a robot hand on the opposite side. The robot hand aims to execute a repeated pick-and-place motion, for example, placing a work on a conveyer belt underneath the machine stand. The work is an object, e.g., a machine part or an electronic component, which will be picked up by the robot hand and be moved to right and left in Fig. 1(a).

When the driving force is given from the motor to the mover, its reaction force will cause rocking mode vibrations in the machine stand due to mutual interference. As is shown in Fig. 1(b), when we drive the motor, the machine stand is vibrated with its swaying motion. Because the vibration amplitude is different between the top and bottom in the machine stand owing to the inherent flexibility of the columns and the leveling bolts, the installed linear encoder on the top of the stand cannot correctly measure the relative displacement between the robot hand and the work. Hence, the conventional MSAFB has to be modified for the rocking mode vibration system so as not to interfere with the semi-closed position and velocity feedback loops that are generally working on positioning and disturbance attenuation.

Hereafter, we define the relative displacement between the stator and the mover as the motor-side displacement, and also define the relative displacement between the robot hand and the work as machine-side displacement. In Fig. 1(a), the motor-side displacement can be measured by the linear encoder, but the machine-side displacement cannot be measured without using an additional displacement sensor.

2.2 Plant Modeling

Figure 2 shows a schematic model of the pick-and-place machine with the rocking mode vibration. In this figure, the flexibility of the mover and the robot hand, namely their rotational modes, are not considered, so suppressing the machine stand vibration is the main focus. When we assume the plant as a simple horizontal two degrees-of-freedom (2DOF) vibration system \(^{(17)}\), the equation of motion at any time \(t\) is derived as follows:

\[
M_m \ddot{x}_m(t) = f_{dr} - D_m (\dot{x}_m - \dot{x}_{st}) + d, \quad \cdots \quad (1)
\]

\[
M_{st} \ddot{x}_{st}(t) = f_r + D_m (\dot{x}_m - \dot{x}_{st}) - D_{st} \dot{x}_{st} - K_{st} x_{st} + f_{ext}, \quad \cdots \quad (2)
\]

where \(x_m\) is the displacement of the mover, \(x_{st}\) is the displacement of the top of the machine stand, \(M_m\) and \(M_{st}\) are the mass of the mover and the machine stand, respectively, \(f_{dr}\) is the driving force generated by the motor, \(D_m\) is the friction coefficient, \(d\) is the disturbance to the mover, \(f_r\) is the reaction force that is equal to \(-f_{dr}\), \(D_{st}\) is the viscous damping coefficient of the machine stand, \(K_{st}\) is the stiffness of the machine stand, \(f_{ext}\) is the external disturbance from the environment to the machine stand. By using the above symbols, the motor-side displacement \(x_{fb}\) and the machine-side displacement \(x_e\) can be expressed as follows:

\[
x_{fb} = x_m - x_{st}, \quad \cdots \quad (3)
\]

\[
x_e = x_m - x_w, \quad \cdots \quad (4)
\]

In the rocking mode vibration, we assume the displacement of the work \(x_w\) as

\[
x_w(t) = k_{ro} x_{ro}(t), \quad \cdots \quad (5)
\]

where \(k_{ro}\) is a ratio of vibration amplitude between the motor-side and the machine-side displacements. This formulation is
because the natural frequency and damping coefficient correspond at the top and bottom of the machine stand. So that means the phases of the vibrations are the same, but the amplitudes are different in the rocking mode vibration system.

2.3 Control System  In the positioning, residual vibrations will occur at both the motor-side and the machine-side due to low stiffness of the \( K_{st} \). In particular, the ISC strategy is beneficial for simultaneously suppressing the vibrations of both displacements. However, because suppressing only the vibration at the machine-side displacement is sufficient as long as the takt time is reduced in some practical situations, we utilize the MSABF to reduce the setting time without suppressing the vibrations of the base platform.

Figure 3 shows the block diagram of the mechanical plant and the control system. The rocking mode is modeled in the mechanical plant, and the framework of the 2DOF control is utilized. The descriptions of the new symbols are as follows: \( x^\ast \) is the position command, \( F_{ipf}(s) \) is the transfer function of the low pass filter, \( C_{ISC}(s) \) is the transfer function of the ISC unit, \( k_p \) is the proportional gain of the position controller, \( C_{acc}(s) \) is the transfer function of the velocity controller, \( C_{acc}(s) \) is the transfer function of the MSABF controller, \( x_c \) is the machine-side displacement, \( x_m \) is the motor-side displacement, and \( a_{st} \) is the machine stand acceleration.

In Fig. 3, the mechanical plant encircled by the dashed line is modeled with reference to Eqs. (1) to (5). The rest of the area of Fig. 3 shows the control structure that consists mainly of the low pass filter, the ISC unit, the position controller, the velocity controller, and the MSABF controller.

The velocity controller is a proportional and integral (PI) controller defined as

\[
C_{v}(s) = k_v \left( 1 + \frac{1}{T_c s} \right) \tag{6}
\]

where \( k_v \) is the proportional gain and \( T_c \) is a time constant.

The \( F_{ipf}(s) \) and \( C_{ISC}(s) \) work as the feedforward controller. The order of \( F_{ipf}(s) \) is designed to make a proper transfer function of the feedforward controller. The transfer functions from position command \( x^\ast \) to the position, velocity, and torque references \( (P_{ref}, v_{ref}, \text{and } \tau_{ref}) \) are defined as follows:

\[
\begin{bmatrix}
P_{ref} & V_{ref} & \tau_{ref} \\
\end{bmatrix} = C_{ISC}(s)F_{ipf}(s)X^\ast, \quad \cdots \quad \cdots \quad (7)
\]

\[
C_{ISC}(s) = \begin{bmatrix} F_{z}(s) & F_{z}(s) & F_{p}(s)M_m s^2 \end{bmatrix} \quad \cdots \quad (8)
\]

where \( F_{z}(s) \) and \( F_{p}(s) \) are transfer functions with the same function as notch filters, and where \( \omega_1, \omega_2, \omega_3, \text{ and } \omega_4 \) are tuning parameters that determine command responsiveness. To suppress the residual vibrations, \( \omega_2 \) and \( \omega_4 \) should be set to anti-resonance and resonance frequencies of the machine stand vibration characteristics, respectively. \( \zeta_4 \) and \( \zeta_2 \) should be set to the damping coefficients of the anti-resonance and resonance.

2.4 Design of the MSABF Controller  The MSABF controller \( C_{acc}(s) \) was designed on the basis of a disturbance localization algorithm under the recognition that the reaction force \( f_r \) is a kind of disturbance \( f_{ext} \).

The transfer function from \( f_{ext} \) to \( x_e \) is derived from Fig. 3 as

\[
\begin{align*}
X_e &= \frac{G_{d}(s)}{G_{d}(s)G_{d}(s) + [C_{acc}(s) - M_m] M_m s^2} \cdot \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdOTS
\end{align*}
\]

where \( F_{z}(s) \) and \( F_{p}(s) \) are transfer functions with the same function as notch filters, and where \( \omega_1, \omega_2, \omega_3, \text{ and } \omega_4 \) are tuning parameters that determine command responsiveness. To suppress the residual vibrations, \( \omega_2 \) and \( \omega_4 \) should be set to anti-resonance and resonance frequencies of the machine stand vibration characteristics, respectively. \( \zeta_4 \) and \( \zeta_2 \) should be set to the damping coefficients of the anti-resonance and resonance.

The right of Eq. (9) is for instance implemented in Fig. 4, where \( F_{ipf}(s) \) is the transfer function of a bandpass filter that works on noise and offset elimination of the detected
machine stand acceleration $a_{st}$. Owing to the effect of integral compensator in $C_v(s)$, the order of $F_{bpf}(s)$ should be set over 2, and the integrators in the MSAFB controller have to be implemented using pseudo-integrators. Because the conventional position and velocity controllers are communalized with the MSAFB controller, the implementation showed in Fig. 4 helps to tune the feedback parameters $k_p$, $k_i$, and $T_v$ on the ground.

In Fig. 4, the transfer function from $a_{st}$ to $f_{dr}$ is derived as

$$
\frac{F_{dr}}{A_{st}} = \frac{k_1 + \frac{k_3 + C_v(s)}{s} + k_p C_v(s)}{s^2} k_2 F_{bpf}(s),
$$

where $k_1$, $k_2$, and $k_3$ are the tuning parameters of the proposed MSAFB controller. Comparing Eq. (10) with Eq. (9), the optimal values of $k_1$, $k_2$, and $k_3$ are given by

$$
[k_1 \ k_2 \ k_3]^T = \left[ M_m k_{na} \ k_{na} - 1 \ D_m \right]^T.
$$

The MSAFB controller consists of four loops. One is a proportional feedback loop having $k_1$, another is the second-order integral feedback loop having $k_2$, and the others are first-order integral feedback loops having $k_3$ and $k_3$. The conventional MSAFB controller has only the proportional feedback loop, and then the motion of the mover is controlled to follow the vibration of the top of the machine stand. The conventional strategy is basically working to keep the motor-side displacement constant in a steady state by setting $k_1 = M_m$ and $k_2 = 0$, which means $k_{na} = 1$ in Eq. (9). However, the mover has to be controlled in practice to follow the vibration of the bottom of the machine stand to keep the machine-side displacement constant. In our controller, the integral loops are essentially working on trajectory formation to eliminate the interference between the semi-closed feedback loops and the conventional MSAFB loop. The proposed MSAFB aims to suppress the vibration of the machine-side displacement. Because the objectives of the conventional and the proposed MSAFB controllers are different, we only show the effects of the proposed MSAFB controller in simulations and experiments below.

3. Simulations

The effects of suppressing the relative vibrations and of reducing the settling time using our MSAFB controller were simulated to test the working principle.

3.1 Positioning Conditions

The block diagram shown in Fig. 3 was programmed in MATLAB/Simulink. Because we used a ball-screw driven system and a rotary servo motor in the experimental evaluation described in Section 4, the translational states in Fig. 3 were appropriately transformed to rotational states with the pitch of the ball-screw at 20 mm/rev.

In the simulations, we set the velocity loop response, namely the open-loop gain crossover frequency, to 28 Hz. The cutoff frequency of $F_{bpf}(s)$ was 13 Hz. The plant parameters were selected so that the natural frequency of the machine stand became 12.6 Hz, the motor inertia was $3.2 \times 10^{-4}$ kg·m², and the inertia ratio between the motor and the load was set to 5.8.

We executed positioning simulations for three cases by switching these control configurations:

- **Case 1**: 2DOF control (without vibration suppression method).
- **Case 2**: 2DOF control with ISC.
- **Case 3**: 2DOF control with MSAFB.

In Case 1, the inverted mechanical plant modeled by $C_{ISC}(s)$ is regarded as a rigid system by setting $F_z(s) = F_x(s) = 1$. In Case 2, the ISC is applied by setting the tuning parameters as follows:

$$
[\omega_z \ \omega_p \ \zeta_p] = [2 \pi \times 11.7 \ \ 2 \pi \times 12.6 \ 0]^T.
$$

In Case 3, MSAFB was applied by setting the tuning parameters $k_1$, $k_2$, and $k_3$ as

$$
[k_1 \ k_2 \ k_3]^T = [0 \ -1 \ 0]^T.
$$

Hence, we set $k_{na} = 0$ and $D_m = 0$ in the simulations. The bandwidth of the bandpass filter $F_{bpf}(s)$ was set from 1.26 to 126 Hz, centering on 12.6 Hz in a log scale.

The command profile is summarized in Table 1. The position command $x^*$ was formulated using an S-shaped curve with a triangle-shaped profile of the velocity command. The max speed was 3000 rpm, the acceleration and deceleration time was 40 ms, and the positioning stroke was 20 mm at the translational machine-side displacement.

### Table 1. Command profiles

<table>
<thead>
<tr>
<th>Position profile</th>
<th>S-shaped curve</th>
</tr>
</thead>
<tbody>
<tr>
<td>Velocity profile</td>
<td>Triangle pattern</td>
</tr>
<tr>
<td>Max speed</td>
<td>3000 [rpm]</td>
</tr>
<tr>
<td>Acceleration &amp; deceleration time</td>
<td>40 [ms]</td>
</tr>
<tr>
<td>Acceleration/deceleration</td>
<td>5.1 [G]</td>
</tr>
<tr>
<td>Position stroke</td>
<td>20 [mm]</td>
</tr>
</tbody>
</table>

3.2 Simulated Position Error Waveforms

Figure 5 shows the simulated position error signals. The position error signals regarding the motor-side and the machine-side displacements are given in Fig. 3 by $(x^* - x_{fb})$ and $(x^* - x_c)$, respectively. Figure 5 also shows the used velocity reference to calculate settling time $T_s$. As illustrated, $T_s$ is a time frame where the position error signal regarding the machine-side displacement goes into position threshold 0.5 mm after the deceleration.

With 2DOF control (see Fig. 5 top), the residual vibration of the motor-side displacement was almost completely
suppressed by the position and velocity feedback controller, while that of the machine-side displacement clearly rose up around 12 Hz. By setting \( \omega_z, \omega_p, \zeta_z, \) and \( \zeta_p \) shown in Eq. (12), when the ISC was utilized in addition to the 2DOF control, both of the residual vibrations were suppressed, and the vibrations were not excited in the entire machine (see Fig. 5 middle). However, the settling time takes 43.1 ms in Case 2. When our MSAFB controller was utilized with the 2DOF control setting \( F_z(s) = F_p(s) = 1 \), the residual vibration at the machine-side displacement was almost completely suppressed, while that at the motor-side displacement became activated (see Fig. 5 bottom). This is because the positioning object was actively moved to follow the vibration motion of the bottom of the machine stand. The settling time was reduced to 31.9 ms in Case 3. The settling time was estimated from the simulations to be reduced by around 25% by the MSAFB.

4. Experimental Results

The experimental results are shown herein, along with the test of the robustness against plant variation.

#### 4.1 Experimental Setup and Conditions

Figure 6 shows a picture of the experimental setup. The system configuration is detailed in Fig. 7. The bottom of the machine stand was designed as a surface plate on which the top of the machine stand was suspended by leaf springs to limit the fundamental rocking mode vibration to about 12 Hz. The ball screw driven system controlled by the rotary servo motor (HF-KP13, Mitsubishi Electric Corp.) was mounted on the top of the machine stand. The pitch of the ball-screw was 20 mm/rev. The motor inertia was \( 3.2 \times 10^{-6} \) kg·m², and mass of the mover, which did not include a ball screw nut, was 0.93 kg. The motor was driven by a rapid prototype control system (dSPACE DS1005) on a trial basis.

Three sensors were installed in the experimental system: one was the rotary encoder for detecting the angular position of the motor, another was the accelerometer (Bandwidth from 0.8 to 3000 Hz) located at the right-hand side of the machine stand to measure the machine stand vibration horizontally, and the other was a laser displacement sensor mounted on the bottom of the machine stand. The last one was utilized to evaluate the residual vibration of the machine-side displacement, and it was not used for implementing feedback control.

In the experiments, the implemented values of the tuning parameters in the ISC and the MSAFB were the same as Eqs. (12) and (13). To improve the offset and noise elimination ability, the bandwidth of the bandpass filter was set narrow from 2.2 to 71.6 Hz, centering on 12.6 Hz in the log scale. The command profiles used in experiments were the same as those in the simulations. Their details have already been shown in Table 1.

Figure 8 shows the frequency response from the motor torque to the motor angular velocity. Both resonance and anti-resonance were detected around 12 Hz due to flexibility of the leaf springs for the machine stand vibration. The inertia ratio between the motor and the load was approximately 5.8.
4.2 Positioning Results Along with the simulated results, the measured position error signals are shown in Fig. 9. The waveforms in Fig. 9 correspond almost exactly to those in Fig. 5. However, because the measurement range of the used laser displacement sensor is 10 mm, the transient state of the machine-side displacement was not measured. The residual vibration amplitude $V_{p-p}$ was specifically 2.28 mm in Case 1, but $V_{p-p}$ was reduced to 0.14 and 0.21 mm in Cases 2 and 3, respectively. The residual vibration was suppressed to less than 10% by using ISC or MSAFB. Moreover, the setting time $T_s$ was 44.8 ms in Case 2, but the $T_s$ was reduced to 32.9 ms in Case 3. Therefore, the setting time was reduced by over 25% by the MSAFB. These results demonstrate that our MSAFB controller is superior to the ISC strategy for reducing the settling time.

4.3 Torque Waveforms Figure 10 shows the measured torque signals depicted by $f_{dr}$ in Fig. 3. In the vertical axis, the typical torque output was 0.32 Nm as set 100% for reference. Because the velocity references used in Fig. 10 have the same profiles as those in Fig. 9, we show the velocity reference at the top only.

The top of Fig. 10 shows that the torque signal residually vibrated around 12 Hz after the deceleration. Thus, the semi-closed feedback loops were working to keep the position error of the motor-side displacement zero. The middle of Fig. 10 shows that the use of the ISC method enabled torque peaks to appear during the positioning. These peaks increased as the cut-off frequency of $F_{lpf}(s)$ was set higher to reduce the settling time. The ISC strategy has problems with torque saturation, so reducing the settling time is difficult. The bottom of Fig. 10 shows that the shape of the torque signal in Case 3 almost correspond exactly to that in Case 1, but the torque signal actively vibrated due to the feedback torque of the MSAFB after the deceleration because the reference trajectory was compensated.

4.4 Machine Stand Acceleration Waveforms Figure 11 shows the measured acceleration signals of the machine stand depicted by $a_{st}$ in Fig. 3. The velocity references used in Fig. 11 have the same profiles as those in Figs. 9 and 10.

In addition, Fig. 11 shows the machine stand vibration occurred in Cases 1 and 3, but the vibration was almost completely suppressed by the ISC after the deceleration in Case 2.

4.5 Evaluation Tests of Robustness To determine the robustness against variation in the mechanical characteristics, we carried out positioning tests by shifting the identified value of the natural frequency of the rocking mode vibration.

Figure 12 shows the measured position error signals in Cases 2 and 3. Although the true value of the natural frequency, basically the resonance and the anti-resonance, is measurable from Fig. 8, $\omega_r$ and $\omega_p$ of the ISC and the central frequency of the bandpass filter $F_{bpf}(s)$ were shifted to 15 Hz in this evaluation. In detail, we set $\omega_r = \omega_p = 2\pi \times 15$ rad/s, and the bandwidth of the bandpass filter was set from 2.23 to 101 Hz centering on 15 Hz in the log scale. In Fig. 12, the $V_{p-p}$ was reduced from 1.0 mm in Case 2 to 0.4 mm in Case 3. Compared to the ISC method, the amplitude of the residual vibration was reduced by 60%. Therefore, the MSAFB was superior to the ISC in terms of robustness against the
variation in the mechanical characteristics, even when the bandwidth of $F_{bpf}(s)$ was not ideally expanded.

5. Conclusions

In this paper, an MSAFB controller was designed to suppress the vibration of the relative displacement in the rocking mode vibration system. The results can be summarized as follows:

1) The positioning machine and its base platform were structurally modeled. A vibration ratio $k_{vo}$ was used for the description of the rocking mode vibration system because the vibration amplitudes were different between the top and bottom of the base platform.

2) The MSAFB controller was designed on the basis of the disturbance localization algorithm. The MSAFB consists of the proportional feedback loop, the first-order integral feedback loop, and the second-order integral feedback loop.

3) The positioning waveforms were simulated. The settling time was estimated to be reduced by around 25% compared to the ISC method.

4) The effectiveness of the proposed MSAFB was demonstrated experimentally. Compared to the ISC method, the settling time was reduced by over 25% and the robustness against variation in the mechanical characteristics was improved by 60%.

References


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