Abstract
Precise positioning technologies are widely used in industrial machines such as device manufacturing equipment. Many studies have been done on feedback control systems for precise positioning stages that improve accuracy by using sensor signal feedback. However, the sensor output does not always directly indicate the performance of the industrial machines because the final requirement of a stage system is generally to adjust a movable stage to a target structure. Therefore, the relative displacement between the stage and the target structure must be controlled. However, the stage and other structures must be considered as elastic elements, especially for precise positioning devices. Thus, this study aims to achieve precise positioning in relative displacement between flexible structures. The authors previously proposed a relative displacement observer that estimated the relative displacement between the stage and the target structure by using only the stage displacement. In this paper, we propose additional control system for relative displacement by using self resonance cancellation technique. The positioning performance of the proposed system is evaluated in numerical simulation and experimental stage system. Results of numerical simulation and experiments show that the proposed control system improves the relative positioning performance.

Keywords: Positioning control, Relative positioning, Precise positioning stage, Flexible structure, Linear motor

1. Introduction

Manufacturing machinery used in industrial fields is required to have ever higher productivity. The positioning stage, which is a key component of these machines, requires mechanical technologies that have both high accuracy and high speed control. For example, the stage system used in semiconductor manufacturing and inspection equipment requires nanometer-order positioning accuracy. The positioning stage is generally controlled using sensor information such as a linear scale or a laser interferometer. When linear guides are used for this kind of equipment, rolling friction behaves as a nonlinear elastic element in the micro-displacement region (Maeda, et al., 2009). There are many studies on the modeling of nonlinear friction and its compensation (Kikuuwe, et al., 2006). The authors have previously proposed a control method using phase stabilization (Ogawa, et al., 2011). Also, many studies have explored ways to improve the positioning stage performance, such as disturbance observer (DOB) (Yamamoto, et al., 2008), final-state control (FSC) (Yazaki, et al., 2015), and perfect tracking control (PTC) (Sakata, et al., 2007).

In the case of semiconductor manufacturing and inspection equipment, the most important factor is the relative positioning accuracy between an exposure column and a semiconductor wafer. Therefore, accurate control of a precise positioning stage does not always lead to improved performance of the whole manufacturing machine. If the target
structure (e.g., an exposure column) is considered to be a complete rigid body, sensor information, which is measured from the base to the stage, is equal to the relative position, which is measured from the target structure to the stage. However, the target structure needs to be considered as an elastic body (flexible structure) in a precise positioning stage because even slight vibration may affect the performance.

The authors previously proposed a relative displacement observer that estimated the relative displacement between the stage and the target structure by using only the stage displacement (Ogawa, et al., 2016a, 2016b). In this paper, we clarify that the characteristic of relative positioning system constructed with flexible arm is similar to that of two-mass system. Therefore, we propose a new control system that combines the proposed observer and the self resonance cancellation (SRC) technique (Sakata, et al., 2014) in order to improve control performance of relative positioning system. The results of numerical simulation and experiments show that the proposed control system improves the relative positioning performance. The proposed control system including the relative displacement observer is very simple and therefore easy to apply to various industrial systems. And also, the observer is very useful for real systems in which relative displacement cannot be measured directly.

2. Evaluation stage system

2.1 Constitution of stage system

Figure 1 shows the principal evaluation stage system to evaluate the performance of the relative positioning control system. This system has a movable stage driven in one direction by one linear motor and two linear scales. Here, linear scale 1 detects the stage's displacement on the basis of the movement of the base part of the stage. Scale 1 imitates a sensor that is usually used for stage control. The scale head of linear scale 2 is attached to the tip of the flexible arm, so it detects the stage's displacement on the basis of the movement of the flexible arm. Scale 2 imitates the relative displacement that has to be controlled. Here, we define the signal of linear scale 1 as “stage displacement” and the signal of linear scale 2 as “relative displacement”. In this paper, only the stage displacement is used for the feedback signal of the control system, and the relative displacement is used only for evaluation.

The system configuration of the evaluation stage system is shown in Fig. 2. The linear motor of the stage is controlled by using a digital signal processor (DSP) via a servo amplifier. Here, servo amplifier is driven as a current amplifier which controls driving current of the motor based on current signal from DSP. The two position signals are input to the DSP, and then digital control software is executed on the DSP. The properties of the evaluation stage system are listed in Table 1.

![Fig. 1 Structure of stage evaluation system. It comprises a movable stage driven by a linear motor, two linear scales, a flexible arm, and mounting springs. Displacement of the stage is measured by two linear scales.](image-url)
2.2 System identification

The characteristics of the evaluation stage system are identified by using sine sweep measurement. Here, sine wave current signal in various frequencies is input to the servo amplifier which controls driving current. The transfer characteristics without control system from the linear motor driving force to the stage displacement are shown in Fig. 3. Here, both characteristics show resonance characteristics at 17 Hz, which is the resonant point due to the mounting spring. However, the results for relative displacement show gain peaks at 122 Hz and 355 Hz. These resonant modes are ascribed to the flexible arm. When it comes to designing a feedback control system, the characteristics of the stage displacement can be used, but the characteristics of the relative displacement cannot. Therefore, in order to improve the control system performance, an observer should be constructed that estimates the relative displacement by using the stage displacement.

3. Relative displacement control system

3.1 Relative displacement observer

This section describes the proposed relative displacement observer. First, the transfer characteristic of the stage displacement in Fig. 3 is modeled.

In general, the transfer function from driving force to a stage displacement is modeled as a superposition of the rigid mode and resonance modes as follows:
Fig. 3  Transfer characteristics of stage system in experiment. Broken blue line represents characteristics from the driving force to the stage displacement. Solid red line represents characteristics to the relative displacement. Here, the characteristic of relative displacement is used for only evaluation, because it is not usually measured in real system directly.

\[ G(s) = \frac{1}{ms^2} + \sum_{i=2}^{N} \frac{a_i}{s^2 + 2\zeta_i \omega_i s + \omega_i^2}, \]  

where \( s \) is the Laplace operator, \( m \) is the moving mass of the stage, \( \omega_i \) is the resonant frequency of the \( i \)-th mode, \( \zeta_i \) is the damping coefficient of the \( i \)-th mode, \( a_i \) is the residue of the \( i \)-th mode, and \( N \) is the number of considered modes.

Each parameter in Eq. (1) can be identified from the transfer characteristic of the stage displacement in Fig. 3. Here, parameters are determined with hand tuning. The identified model of the stage displacement is indicated by the broken blue line in Fig. 4. Here, we assumed \( N = 4 \), so the model is composed of one rigid mode and three resonant modes. The first resonant mode is caused by the mounting springs, and the second and third modes are caused by the flexible arm.

Equation (1) can be rewritten as a state space equation with \( i \)-th modal amplitude \( q_i \):

\[
\begin{align*}
\dot{x} &= Ax + Bu \\
y &= C_s x,
\end{align*}
\]  

where

\[
\begin{align*}
x &= [q_1 \cdots q_N \dot{q}_1 \cdots \dot{q}_N]^T, \\
A &= \begin{bmatrix} 0_{N\times N} & I_N \\ \Omega_N & Z_N \end{bmatrix}, \\
B &= \begin{bmatrix} 0_{N\times 1} \\ K_N \end{bmatrix}, \\
C_s &= [1 \cdots 1 \ 0_{1\times N}], \\
\Omega_N &= \text{diag}[0 \ -\omega_2^2 \ -\omega_3^2 \ \cdots], \\
Z_N &= \text{diag}[0 \ -2\zeta_2\omega_2 \ -2\zeta_3\omega_3 \ \cdots], \\
K_N &= \left[ \frac{1}{m} \ a_2 \ a_3 \ \cdots \right]^T.
\end{align*}
\]

Because the evaluation system is 1 input and 2 outputs system originally, the plants for stage displacement and relative displacement should share the same state variables: the \( A \) matrix and \( B \) matrix. That is, the difference between the two models is only the output matrix. In other words, once the output matrix for the relative displacement, \( C_r \), is determined by using the output matrix for the stage displacement, \( C_s \), we can estimate the relative displacement by using estimated state variables from a conventional state observer.
Fig. 4 Identified frequency responses of the stage displacement and estimated model of the relative displacement. Broken blue line indicates the measured frequency response of the stage displacement. Solid red line indicates the estimated characteristics of the relative displacement.

Furthermore, because the state space equation is shown as Eq. (3), each state variable indicates the modal amplitude of each resonant mode. Then, each coefficient ratio of $C_s$ and $C_r$ is determined by the amplitude ratio of each modal shape. As a result, when $C_r$ is described as Eq. (4), the only unknown parameter is coefficient $\alpha_{ri}$, which is determined by the modal shape for each mode:

$$C_r = [\alpha_{r1} \cdots \alpha_{rN} \quad 0_{1\times N}]$$

Some methods exist that can be used to determine coefficient $\alpha_{ri}$ in Eq. (4). For example, the amplitude ratio of each modal shape in the resonant frequency can be determined by using additional sensors such as accelerometers. In our study, we performed a FEM analysis using ANSYS® (version 14.5, ANSYS Inc.) to determine $\alpha_{ri}$.

The authors previously proposed an analysis method that couples with mechanical-control models and considers elastic deformation (Ogawa, et al., 2013). The conditions of the FEM analysis are listed in Table 2, its mesh model is shown in Fig. 5, and the results are shown in Fig. 6. The four modes of vibration in which the relative displacement is dominant are also shown in Fig. 6. The resonant frequencies and modal shapes are obtained from the FEM analysis. The amplitude ratio of the stage displacement and relative displacement, $\alpha_{ri}$, is determined only by the modal shape in FEM. Here, the resonant frequencies in Fig. 6 differ from the experimental results in Fig. 3. The difference is presumably due to modeling errors stemming from the material properties or constraint conditions.

<table>
<thead>
<tr>
<th>Table 2 Properties of FEM modal analysis.</th>
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<tbody>
<tr>
<td>Mesh type</td>
</tr>
<tr>
<td>Number of nodes</td>
</tr>
<tr>
<td>Number of elements</td>
</tr>
<tr>
<td>Number of modes</td>
</tr>
</tbody>
</table>
Fig. 5  Mesh model in FEM analysis. The FEM model is generated by a CAD model that includes mechanical dimensions, material properties, and boundary conditions of each mechanical element. Here, the four mounting plates are fixed to the ground.

Fig. 6  Results of FEM modal analysis. Resonant frequencies and modal shapes are obtained from the FEM analysis. There are four modes of vibration in which the relative displacement is dominant. These are mainly resonant modes of the flexible arm.

By using the $C$ matrix for the relative displacement $C_r$, we estimate the transfer characteristic of the relative displacement, as indicated by the red line in Fig. 4. Note that this transfer characteristic of relative displacement is not identified from the experimental results in Fig. 3 but is estimated from combination of the identified characteristics of the stage displacement in Fig. 4 and output matrix $C_r$ which is determined from FEM analysis. Also, note that the analytical results cannot be directly used to model the plant due to a modeling error in the resonant frequencies between the experiment and FEM analysis. However, because only the modal amplitude ratio is used for the modeling in the proposed observer, Fig. 4 indicates that the estimation is sufficiently accurate.

3.2 Feedback controller design

The feedback control system using the proposed relative displacement observer follows the process below.

(a) The state observer based on Eq. (2) estimates the state variables by using the stage displacement.

(b) The relative displacement that is used for the position feedback signal is calculated by using the output equation with $C_r$.

(c) The velocity feedback signal is calculated by using the output equation with $C_{src}$.

Here, $C_{src}$ is output matrix for velocity feedback by using self resonance cancellation (SRC) techniques. The output matrix of SRC system is designed as

$$C_{src} = \gamma \times C_s + (1 - \gamma) \times C_r.$$  \hspace{1cm} (5)

Here, $\gamma$ is a design parameter for cancellation of resonance. In this paper, $\gamma$ is designed to improve stability margin of the control system at the resonant mode in 140 Hz, and we determined it as $\gamma = 1.2$ with hand tuning. Bode diagrams of plant model is shown in Fig. 8.

The block diagram of the proposed control system is shown in Fig. 7. It is constructed as a P-PI control system that uses estimated state variables. Here, the observer gain $L$ is designed by the method of linear quadratic regulator by using stage displacement model.
Fig. 7 Block diagram of proposed control system. It comprises a state observer, a relative position estimator, and a feedback control system. The feedback controller is constructed as P controller for position loop and PI controller for velocity loop.

Fig. 8 Bode diagrams of plant model. The black dashed lines is the imaginary plant with SRC. The design parameter $\gamma$ is determined to improve stability margin of the control system at the resonant mode.

The control parameters for each system are designed independently as all systems have same stability in a Nyquist diagram. The designed parameters of controllers are listed in Table 3. Here, because the relative displacement is used for the velocity feedback signal, stability of the velocity loop is decreased due to resonant mode of flexible arm. So, a notch filter is adopted to the system 2. The characteristics of designed control systems are shown in Fig. 9. Here, the system 1 is a traditional control system, and the stage displacement is adopted for position control signal. Therefore, the relative displacement cannot follow the reference trajectory in the system 1. In the systems 2–4, the relative displacement can follow the reference trajectory. However, as shown in Fig. 9, the bandwidth of the system 2 is lower than other systems due to phase delay of the notch filter. The system 3 is also a traditional control system, and it is known that the system has good performance because the resonant peak gain of the stage displacement is smaller than that of the relative displacement. The system 4 is proposed system shown in Fig. 7. The bandwidth of the system 4 is higher than that of system 3 and almost same as that of system 1. It is because the SRC system cancel the resonant mode.

The stage movement conditions in simulation evaluation are listed in Table 4, and the results are shown in Fig. 10. As shown in Fig. 10, in system 1, the position response settles immediately at the target position. However, the response of system 2 is delayed compared to system 1, and amplitude of settling vibration is large. In systems 3 and 4, positioning performances are good, but the settling vibration is slightly small in system 4.
Fig. 9 Characteristics of designed control systems. (a) is the bode diagram of open-loop characteristics. (b) is the Nyquist diagrams of open-loop characteristics. (c) is the gain response of sensitivity functions. All systems are designed as they have same stability in a Nyquist diagram.

Fig. 10 Simulation results of the stage positioning. Broken black line indicates the reference trajectory, the blue dashed lines and red solid lines indicate the relative displacement response of the each system.
4.2 Experimental results of evaluation stage system

In the final experiment, we evaluated the effectiveness of the proposed control system using the stage evaluation system that was shown in Fig. 1. The experimental conditions were the same as the numerical simulation shown in Table 4, and the results are shown in Figs. 11 and 12. Figure 11 indicates the positioning error of the relative displacement sensor and Fig. 12 indicates the positioning response for the target position.

As shown in Figs. 11 and 12, the proposed control method (system 4) reduced the position error signal compared to system 3, and its performance was almost the same as that of system 1, which is the same result as in the numerical simulations. This shows that the proposed control system improved the relative positioning performance of the precise positioning system.

Table 3 Parameters of controllers

<table>
<thead>
<tr>
<th></th>
<th>P-Kp</th>
<th>V-Kp</th>
<th>V-Ki</th>
<th>Notch filter</th>
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<td>System 1</td>
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<tr>
<td>System 2</td>
<td>100</td>
<td>18</td>
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</tr>
<tr>
<td>System 3</td>
<td>150</td>
<td>20</td>
<td>3770</td>
<td>No</td>
</tr>
<tr>
<td>System 4</td>
<td>150</td>
<td>25</td>
<td>6283</td>
<td>No</td>
</tr>
</tbody>
</table>

(a) Systems 1 and 2

(b) Systems 3 and 4

Fig. 11 Experimental position error of the relative displacement sensor. The systems 1-4 are the same as in Fig. 9. Vertical axis indicates the position error that is calculated as difference between position trajectory and relative displacement signal.

Table 4 Stage movement parameters

<table>
<thead>
<tr>
<th>Move distance</th>
<th>30 mm</th>
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<tbody>
<tr>
<td>Velocity pattern</td>
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<tr>
<td>Acceleration</td>
<td>2 m/s²</td>
</tr>
<tr>
<td>Maximum velocity</td>
<td>0.2 m/s</td>
</tr>
</tbody>
</table>

(a) Systems 1 and 2

(b) Systems 3 and 4

Fig. 12 Experimental positioning responses for target position. Vertical axis is the stage position measured by the relative displacement sensor. The target position is zero. Systems 1 through 4 are the same as in Fig. 11.
5. Conclusion

To improve the positioning performance of mechanical systems that include a precise positioning stage, we proposed a relative displacement observer and feedback control system by using SRC techniques. The proposed system estimates the relative positioning accuracy between the stage and the positioning target structure. We evaluated the performance of the proposed control system by numerical simulation and experiments. The results showed that the proposed system improved the relative positioning performance of the system. The proposed observer is very simple, so it should be easy to apply to industrial equipment.

References