High-precision positioning considering vibration and disturbance suppression in piezo-driven stage systems

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Abstract
This paper presents a controller design approach to compensate for the disturbances and achieve robust vibration suppression against variations in the resonant frequency in a piezo-driven stage system. Disturbances such as nonlinearities due to hysteresis and creep phenomena are inherent in piezoelectric actuators, resulting in a low control performance of positioning or low tracking accuracy. Although various types of modeling and model-based approaches have been proposed to compensate for the nonlinearities, these approaches have limitations such as substantial modeling complexity, time-consuming modeling process, and high computational cost required for their evaluation and implementation. Moreover, resonant vibration modes in the mechanism lead to residual vibrations in the positioning or may even destabilize the system. In particular, the vibrational dynamics tend to have a low stability margin in piezo-driven systems because of sharp resonant peaks associated with the low structural damping. In this study, a state observer is employed to estimate the disturbance and vibration mode signals, wherein the observer is designed to suppress the disturbance and the vibration by using the estimated signals. Considering the hysteresis as an input disturbance to the linear plant renders complex modeling processes unnecessary. To compensate for the mechanical vibrations, pole-assignment method is used to achieve the gain-peak reductions and robustness against variations in the resonant frequency. The effectiveness of the proposed approach was verified by conducting experiments on a piezo-driven stage system.

Keywords: Piezo-actuators, High-precision positioning, Vibration suppression, Disturbance suppression, State feedback controller

1. Introduction

High-precision positioning is one of the key technologies in various mechatronic systems such as storage devices (Atsumi, 2016), industrial machines (Butler, 2011, Khoshdarregi et al., 2014), and manipulators (Brogårdh, 2007). As one of the actuator devices in mechatronic systems, piezoelectric actuators have been widely employed in nanopositioning fields, including nanopositioning stage systems, nanorobotic manipulators, scanning tunneling microscopes (STMs), and atomic force microscopes (AFMs), because of advantages such as high resolution, rapid response, and large force generation (Devasia et al., 2007, Fleming and Leang, 2014). However, uncertainties exist in piezo-driven systems, such as nonlinearities due to hysteresis and creep phenomena, which deteriorate the positioning and/or tracking accuracy. In addition, resonant vibrations in the mechanism lead to vibratory responses or system instability. Hence, to achieve a higher and precise control performance, it is necessary to compensate for nonlinear characteristics as well as resonant vibrations.

Hysteresis is one of the typical nonlinear characteristics in the piezoelectric actuators, which manifests between the applied voltage and the output displacement. The maximum error due to the hysteresis can be approximately 10–15 % for the maximum stroke in an open-loop control. In the open-loop control, various types of modeling and inverse model-based feedforward approaches to compensate for the hysteresis have been proposed such as the Bouc–Wen model-based compensation (Rakotondrabe, 2011), the Preisach model-based compensation (Ge and Jouaneh, 1996, Song, et al., 2005), and the Prandtl–Ishlinskii model-based compensation (Janaideh et al., 2011). However, the limitations of these approaches...
are the substantial modeling complexity and time-consuming modeling process because of the difficulties in understanding the hysteresis behavior and deriving complex mathematical models for the same. In addition, the performance of inverse model-based compensation methods depends on the accuracy of the modeling; hence, the control performance is unsatisfactory because of modeling error and/or parameter variations. The creep phenomenon, which is another nonlinear component, requires complex mathematical modeling and parameter tuning, wherein the problems associated with the hysteresis compensation exists in the open-loop control.

Feedback control systems that employ high-resolution sensors are indispensable to overcome the problems of the open-loop control (Leang and Devasia, 2007), wherein the high-gain feedback controller helps in suppressing the nonlinearities and other disturbances. However, the feedback gain cannot be adequately increased because the general positioning mechanisms that employ piezoelectric actuators inherently include mechanical vibration modes. In particular, the vibrational dynamics tend to have a low stability margin because of the sharp resonant peaks associated with the low structural damping. In practice, notch filters are used to suppress the resonant peaks (Okazaki, 1990, Seki et al., 2014). However, because the notch filter can only reduce the gain peak of the corresponding resonant frequency in a small range, a robust stability against the variations in the resonant frequency cannot be ensured. In addition, the phase delay in the notch filter decreases the phase margin. To overcome the control problem of a system that includes piezoelectric actuators, advanced control techniques have been developed to improve the control accuracy and bandwidth such as the adaptive methods (Zhong and Yao, 2008), the robust-control approach (Salapaka et al, 2002), the sliding-mode control (Abidi and Savanovic, 2007), and the repetitive control (Shan and Leang, 2013). However, these control techniques cannot be implemented because of high computational cost or complex algorithms and parameter tuning depending on the system configuration. Therefore, these approaches should be appropriately selected considering the control purpose, plant characteristics, computational performance of the controller, etc. From a practical viewpoint, to meet the changing target specifications and system configurations, many practical control approaches need to be developed and prepared for industrial applications, and the effectiveness of such systems must be verified.

This paper presents a controller design approach to compensate for the disturbances and achieve robust vibration suppression against the variations in the resonant frequency in a piezo-driven stage system. A disturbance observer is introduced to cancel the disturbance, wherein the hysteresis is considered as an input disturbance for the linear characteristic of the nominal condition. Hence, in the controller design, the complicated mathematical modeling of the hysteresis is not required owing to the disturbance observer. In addition, a state observer is designed to attenuate the gain peaks of the vibration modes, wherein poles are assigned to reduce the sensitivity near the resonant frequencies. The position and originality of this article is to show a simple and practical control design approach considering compensation of nonlinearities without complex model and vibration suppression against the variations simultaneously in the piezo-driven systems. The effectiveness of the proposed design approach is verified by conducting experiments on a piezo-driven stage system. In the experiments, several motion patterns and transient responses are evaluated from a practical viewpoint.

2. System configuration and plant model

2.1. System configuration of the target piezo-driven stage system

Figure 1 shows an overview of a target piezo-driven stage (P-622.1, Physik Instrumente GmbH & Co. KG.). The stage (25mm×30mm) is constrained to move only in one direction with the help of frictionless elastic hinges. The maximum stroke of stage is 250 μm, which is obtained using the integrated displacement amplifier mechanisms. Figure 2 shows the system configuration as the experimental setup. The stage displacement $y$ is detected using a linear encoder.
2.2. Linear dynamics modeling

The solid blue lines in Fig. 3 indicate the frequency characteristic of the stage displacement \( y \) for the control input \( u \). The characteristic is measured using a dynamic signal analyzer (35670A, Agilent Technologies, Inc.) by employing swept-sine excitations. The figure shows that the mechanism includes two mechanical vibration modes due to the elastic deformation of the hinges in the high-frequency range. Hence, the linear dynamics can be formulated as a plant mathematical model \( P(s) \), which comprises two vibration modes.

\[
P(s) = \frac{y}{u} = K_a K_g \left( \sum_{i=1}^{2} \frac{k_{mi}}{s^2 + 2\zeta_{mi}\omega_{mi}s + \omega_{mi}^2} \right),
\]

where \( K_a \) is the gain of the power amplifier, \( K_g \) is the linear plant gain, \( \omega_{mi} \) is the natural angular frequency of the \( i \)th vibration mode, \( \zeta_{mi} \) is the damping coefficient of the \( i \)th vibration mode, and \( k_{mi} \) is the modal constant of the \( i \)th vibration mode. As the cutoff frequency of the power amplifier is above 10 kHz, the characteristic is considered as a gain component. The broken red lines in Fig. 3 indicate the frequency characteristic of the mathematical model \( P(s) \). Table 1 lists the model parameters identified via curve fitting.

2.3. Simple feedback control system

A simple feedback control system shown in Fig. 4 is generally designed for the plant \( P(s) \). In the figure, \( C_n(s) \) and \( r \) indicate the feedback compensator and the displacement reference, respectively. Figure 3 shows that each vibration mode has sharp peaks. The gain peak of the 1st resonant frequency is approximately 30 dB higher than the DC gain of the plant. Hence, in the feedback controller design, damping should be sufficient to ensure the stability of the system and extend the servo bandwidth. As a simple approach to compensate for the effects of the vibration modes using a feedback compensator, gain compensation using a notch filter is generally applied in the industrial products. Here, \( C_n(s) \) is designed

![Fig. 2 System configuration of the target system. The piezoelectric actuator is driven based on the control input obtained from a power amplifier. The displacement signal is transferred to a controller through an interface board.](image)

![Fig. 3 Frequency characteristics of the stage displacement \( y \) for the control input \( u \). The plant characteristic includes two vibration modes.](image)

![Fig. 4 Feedback control system. The feedback compensator \( C_n(s) \) is generally designed considering robust stability, vibration suppression, and positioning performance.](image)

### Table 1 Parameters of the plant model identified via curve fitting based on the measurement result.

| Parameter  | Value       |
|------------|-------------|
| \( K_a \)  | 10.0        |
| \( K_g \)  | 2.0 \times 10^6 |
| \( \omega_{m1} \) [rad/s] | 2π \times 360 |
| \( \zeta_{m1} \) | 0.012 |
| \( k_{m1} \) | 4.0 |
| \( \omega_{m2} \) [rad/s] | 2π \times 600 |
| \( \zeta_{m2} \) | 0.008 |
| \( k_{m2} \) | 3.4 |

(MercuryII6830, GSI Group Inc.) attached to the stage; the resolution of the encoder is set to 5 nm per pulse. The pulse signals are transferred to a controller (DS1103, dSPACE GmbH) through an interface board with a sampling period of 20 \( \mu \)s. The stacked piezoelectric actuator is driven based on the control input \( u \) obtained from a power amplifier (HSA 4014, NF Corporation). The gain of the power amplifier is set to 10.
as the following proportional-integral compensator and notch filters.

\[ C_n(s) = \left( K_{PN} + \frac{K_{IN}}{s} \right) \prod_{i=1}^{2} \left( \frac{s^2 + 2\zeta_{ni}\omega_{ni} + \omega_{ni}^2}{s^2 + 2\zeta_{ndi}\omega_{ndi} + \omega_{ndi}^2} \right) \]  

(2)

Two notch filters are designed to reduce the gain peaks of each vibration mode. The integral compensation is designed to improve the steady state performance. Table 2 lists the designed parameters of \( C_n(s) \). These parameters are tuned by considering the stability of the system and settling performance without residual vibration. Figure 5 shows the Nyquist diagrams. Figure 6 shows the closed-loop characteristics of the feedback control system with the notch filter under the resonant-frequency variations.

![Fig. 5 Nyquist diagrams of the feedback control system with the notch filter under resonant-frequency variations.](image)

fig. 5

![Fig. 6 Closed-loop characteristics of the feedback control system with the notch filter under the resonant-frequency variations.](image)

fig. 6

![Fig. 7 Sensitivity gain of the feedback control system with the notch filter.](image)

fig. 7

These figures show that the suppression performance against the variations in the resonant frequency is insufficient. From the plant characteristic shown in Fig. 3 and the sensitivity characteristic shown in Fig. 7, the disturbance in the control system excites these vibration modes with a gain compensation. Figure 8 shows the positioning waveforms for the step inching motion: the upper figure shows the step response for the reference; the lower figure shows the error waveform between the reference and the response. The creep phenomenon can be compensated by feedback control with integral compensation. On the other hand, this result confirms that overshoot and residual vibration occur for each motion because of the hysteresis characteristic and the slight variation in the resonant frequency.
3. Feedback controller design considering vibration and disturbance suppression

3.1. Design of state observer considering disturbance

Based on the mathematical model expressed in Eq. (1), the state space representation can be expressed as follows.

\[ \dot{x} = A_P x + B_P u, \]
\[ y = C_P x, \]

where

\[ x = \begin{bmatrix} x_1 & \dot{x}_1 & x_2 & \dot{x}_2 \end{bmatrix}^T, \]
\[ A_P = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\omega_{m1}^2 & -2\zeta m1\omega_{m1} & 0 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & -\omega_{m2}^2 & -2\zeta m2\omega_{m2} \end{bmatrix}, \]
\[ B_P = \begin{bmatrix} 0 & K_a K_g k_{m1} & 0 & K_a K_g k_{m2} \end{bmatrix}^T, \]
\[ C_P = \begin{bmatrix} 1 & 0 & 1 & 0 \end{bmatrix}. \]

\( x_1 \) is the output displacement of the 1st vibration mode, and \( x_2 \) is the output displacement of the 2nd vibration mode. To consider the compensation of the hysteresis characteristic as well as the vibration suppression, the hysteresis characteristic is regarded as the input disturbance for the linear dynamics. Hence, an augmented plant that includes the disturbance \( d \) as a state variable needs to be defined, which is given as follows.

\[ \dot{x}_o = A_o x_o + B_o u, \]
\[ y = C_o x_o, \]
where
\[
x_o = \begin{bmatrix} x & d & y \end{bmatrix}^T, \quad A_o = \begin{bmatrix} A_P & B_P & 0 \\ 0 & 0 & 0 \end{bmatrix}, \quad B_o = \begin{bmatrix} 0 \\ B_P \\ 0 \end{bmatrix}^T, \quad C_o = \begin{bmatrix} C_P & 0 \end{bmatrix},
\]
Based on Eq. (6), the state observer is designed as follows.
\[
\dot{x}_o = A_o\dot{x}_o + B_ow + LC_o(x_o - \dot{x}_o),
\]
where \(\dot{x}\) denotes the estimation value, and \(L\) is the gain of the observer. Here, by considering convergence speed of the estimated variables and the sensor noise, \(L\) is designed as follows.
\[
L = [0.0014, 6.2111, -0.0011, -2.5371, 0.0001]
\]
Figure 9 shows a block diagram of the control system with the state observer considering the disturbance. Here, \(P(s)\) is the plant, \(u\) is the control input, \(y\) is the stage displacement, \(\dot{y}\) is the estimated displacement, \(\dot{e}\) is the estimation error, \(F\) is the state-feedback gain \(F = [f_{p1}, f_{o1}, f_{p2}, f_{o2}]\). The characteristic equation of the plant \((y/u)\), shown in Fig. 9, is given as follows.
\[
G_p(s) = s^4 + D_1s^3 + D_2s^2 + D_3s + D_4 = 0
\]
Eq. (8) shows that poles can be arbitrarily assigned using the state-feedback gain \(F\), where the gain is designed using a pole placement method considering attenuation of the vibration peaks and reduction in the sensitivity gain around the resonant frequencies. Here, the state-feedback gain is designed as \(F = [1 \times 10^{-4}, 8 \times 10^{-5}, 0.25, 8 \times 10^{-5}]\).

Figures 10, 11, and 12 show the sensitivity gain with the state feedback, the pole assignments of the characteristic of \(u\) for \(y\), and the frequency characteristics of \(u\) for \(y\), respectively. In Fig. 12, the blue solid lines and the red broken lines indicate the characteristics without the state feedback and with the state feedback, respectively. These figures show that the vibration poles approach the real axis in the complex plane, while sufficiently suppressing the sensitivity gain and the gain peaks of the vibration modes by applying the designed state feedback.

### 3.2. Design of the feedback control system

Figure 13 shows the block diagram of the position control system. The position feedback compensator \(C_o(s)\) is designed for the augmented plant including state feedback. The damping effect against each vibration mode is governed by the state feedback, while the integral compensation should be applied to cancel the steady state error for the step.
reference in the feedback compensator. In this study, the following proportional-integral compensator is designed as $C_o(s)$.

$$C_o(s) = K_{oi} + \frac{K_{op}}{s} \tag{9}$$

The integral gain $K_{oi}$ and the proportional gain $K_{op}$ are determined considering the stability of the system and the servo performance. These parameters are set as $K_{oi} = 35$ and $K_{op} = 0.005$. Figure 14 shows the Nyquist diagram. Figure 15 shows the frequency characteristics of the stage displacement for reference in Fig. 13. In these figures, the blue solid lines indicate the control performances for the plant characteristic under the nominal condition without the resonant-frequency variations shown in Fig. 3. The red broken lines and the green chained lines indicate the conditions for frequency variations of $+20\%$ and $-20\%$, respectively. These results show that the vibration suppression performance against the variations in the resonant frequency is satisfactory.

![ Nyquist diagram of the feedback control system with the state observer under resonant-frequency variations.](image)

![ Frequency characteristics of the stage displacement $y$ for the reference $r$ with the state observer under the resonant-frequency variations.](image)

(a) experimental results for the input with different amplitudes (frequency : 1 Hz)

(b) experimental results for the input with different frequencies (amplitude : 2.5 V)

![ Lissajous waveforms of the displacement for the sinusoidal-input voltage in the experiments.](image)
Seki, Noda and Iwasaki, Journal of Advanced Mechanical Design, Systems, and Manufacturing, Vol.11, No.3 (2017)

4. Experimental verifications

4.1. Hysteresis compensation performance

To verify the estimation and compensation performances of the controller for the hysteresis characteristic, the results of the open-loop control and the feedback control using the estimated disturbance signal \( \hat{d} \) are compared. In the experiment, the sinusoidal signals with different amplitudes and frequencies (amplitude: 0.5, 1.5, 2.5, 3.5, and 4.5 V, frequency: 0.1, 1, and 10 Hz) are applied as the control input \( u \). Figure 16 shows the Lissajous curve of the displacement for the control input, where (a) shows the experimental results for the input with different amplitudes, (b) shows the experimental results for the input with different frequencies, left figures of (a) and (b) indicate the experimental results for the open-loop control, and right figures of (a) and (b) indicate the experimental results considering the estimated disturbance signal \( \hat{d} \). This result verifies that the hysteresis characteristic can be compensated using the estimated disturbance signal without accurately modeling the hysteresis.

4.2. Positioning performance

The positioning performance is evaluated using the designed control system, shown in Fig. 13, wherein the resonant-frequency variation is simulated by placing a load on the stage to verify the effectiveness of the control performance. The red broken lines in Fig. 17 show the plant characteristic with a load of 22 g acting on the stage, while the blue solid lines indicate the nominal plant characteristic, shown in Fig. 3. This result show that the 1st resonant frequency changes to 290 Hz (−70 Hz) after placing the load. On the other hand, the influence of nonlinearity to frequency characteristics depends on the amplitude. Green and cyan lines in Fig. 17 show the measurement results of frequency characteristics without load using different amplitudes. The amplitudes are set as 50 and 150 mV, considering over-stroke at resonant frequencies. Although the gains at low frequency have tiny variation due to the effect of hysteresis in the characteristics, these variations can be suppressed as shown the experimental results in Fig. 16. On the other hand, the resonant frequencies slightly change for different amplitudes. However, the amount of variations is smaller than loading of mass on the table.

Figure 18 shows the response waveforms for the step reference with an amplitude of 50 \( \mu \)m under the nominal condition.
Conclusions

This paper presents a controller design approach considering the compensation for disturbances and a robust vibration suppression against the resonant-frequency variations in the piezo-driven stage systems. A disturbance observer was designed to cancel the disturbance for the case where the hysteresis was considered as the input disturbance for the linear characteristic of the nominal plant. The results show that complicated mathematical models are not required for the hysteresis when using the disturbance observer in the controller design. Moreover, a state observer was designed to attenuate the gain peaks of the vibration modes, wherein the pole assignment was considered to reduce the sensitivity near the resonant frequencies. Here, a robust vibration and disturbance suppression performance against the resonant-frequency variations can be achieved using the feedback control system along with the state observer.

Figure 20 shows the experimental results in the inching motion for an amplitude interval of 50 µm. Figure 21 shows the experimental results in the step motion with different amplitudes. In the figures, (a) indicates the displacement references, responses, and error waveforms between the reference and the response; (b) indicates the magnified error waveforms for the control system with the notch filters; (c) indicates the magnified error waveforms for the control system with the state observer. To verify the difference in each response, the magnified error waveforms are overlapped with the step-reference input time of 0 s in Figs. 20(b) and 20(c) and Figs. 21(b) and 21(c). These figures confirm that the dispersion and overshoot due to the hysteresis can be suppressed using the control system along with the state observer. In addition, the residual vibration that occurs after positioning can be suppressed using the control system along with the state observer.

Figure 22 shows the response waveforms for the impulse disturbance after positioning, where the disturbance is inputted at 0.5 s after positioning for 50 µm. This result shows that, as the sensitivity gain at the resonant frequencies reduces by employing the proposed control system, as shown in Fig. 10, the disturbance can be suppressed sufficiently.

5. Conclusions

This paper presents a controller design approach considering the compensation for disturbances and a robust vibration suppression against the resonant-frequency variations in the piezo-driven stage systems. A disturbance observer was designed to cancel the disturbance for the case where the hysteresis was considered as the input disturbance for the linear characteristic of the nominal plant. The results show that complicated mathematical models are not required for the hysteresis when using the disturbance observer in the controller design. Moreover, a state observer was designed to attenuate the gain peaks of the vibration modes, wherein the pole assignment was considered to reduce the sensitivity near the resonant frequencies. Here, a robust vibration and disturbance suppression performance against the resonant-
frequency variations is achieved. The effectiveness of the proposed approach was verified by conducting experiments using a piezo-driven stage system. The amount of the frequency variation in this paper is considered sufficient to keep the performance against small mass change, modeling error, and the characteristic change due to aging or temperature variation. On the other hand, when the resonant frequency greatly changes due to installation of the large mass, the feedback gains should be tuned for the plant characteristic with large mass. In the future works, the adaptive algorithm should be introduced to improve the robustness against large frequency variations.

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