Practical Ultraprecision Positioning of a Ball Screw Mechanism

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This paper describes the problem of ultraprecision positioning with a ball screw mechanism in the microdynamic range, along with its solution. We compared the characteristics of two ball screw mechanisms with different table masses. The experimental results showed that the vibration resulting from the low stiffness of the ball screw degraded the positioning performance in the microdynamic range for the heavyweight mechanism. The proposed nominal characteristic trajectory following (NCTF) controller was designed for ultraprecision positioning of the ball screw mechanism. The basic NCTF control system achieved ultraprecision positioning performance with the lightweight mechanism, but not with the heavyweight mechanism. A conditional notch filter was added to the NCTF controller to overcome this problem. Despite the differences in payload and friction, both mechanisms then showed similar positioning performance, demonstrating the high robustness and effectiveness of the improved NCTF controller with the conditional notch filter. The experimental results demonstrated that the improved NCTF control system with the conditional notch filter achieved ultraprecision positioning with a positioning accuracy of better than 10 nm, independent of the reference step input height.

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NOMENCLATURE

d_{micro} = difference between the peak and the final displacements in an open-loop step response
K_I = integral gain of the PI compensator
K_P = proportional gain of the PI compensator
K_{Pu} = proportional gain in practical stability limit
t_r = period during the application of u_r to the mechanism
u_r = step input height
x_f = final displacement of an open-loop step response
α = factor indicating a damping characteristic
ω_n = controller parameter in rad/s
ω_{notch} = natural frequency of the conditional notch filter
ζ = dimensionless controller parameter
ζ_{notch} = damping ratio of the conditional notch filter
ζ_{prac} = dimensionless controller parameter in practical stability limit

1. Introduction

Precision positioning systems are important in industries in which several kinds of precision positioning mechanisms are used. The performance of precision positioning systems is greatly influenced by the characteristics of the mechanism employed. While the use of linear motors for high-speed positioning is on the increase, ball screw mechanisms are still the most widely used precision positioning mechanism.[1]

Precision positioning mechanisms include mechanical elements such as linear guides, a power transmission, support units for the power transmission, and an actuator. These mechanical elements often have friction characteristics, which degrade the positioning system performance by causing steady-state errors, tracking errors, and limit cycles, as well as generally slowing down the mechanism. A solution to these friction problems is very important for ultraprecision positioning of mechanisms with mechanical contacts. Noncontact mechanisms including linear motors and pressurized fluid bearings are effective for solving the problems and used for ultraprecision positioning.[2,3] However the use of their mechanisms is often unpractical.

The behavior of mechanisms with friction characteristics depends on the working range of the mechanism. Moreover, mechanisms exhibit microdynamic characteristics such as springlike behavior[4] or nonlinear spring behavior[5] in a narrow working range. These behaviors have been observed in ball or slide bearings,[6-8] DC servomotors,[9,10] and ball screws.[11] This behavior works to mask the dead zone caused by Coulomb friction and is thus often considered to be useful for ultraprecision point to point (PTP) positioning of mechanisms with a friction characteristic.[12] Experimental nanometric positioning of mechanisms with microdynamic characteristics has been reported.[6,13]

This behavior also indicates the reduction of the friction damping effect in a small working range. Positioning mechanisms often have several vibration factors, and the influence of these factors on positioning performance depends on the friction characteristics.

For high-performance precision positioning, several types of

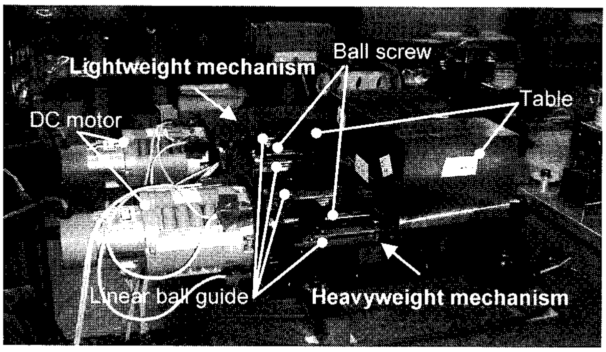


Fig. 1 Experimental ball screw mechanisms

controller have been applied to mechanisms with microdynamic characteristics, and their performances have been examined. Switching-type controllers made up of several sub-controllers, and controllers with disturbance observers are frequently used for high-precision positioning over a wide working range.[10,14,15] Sliding mode controllers also have been used for precision positioning systems.[16] These controller designs require accurate model parameters and sufficient knowledge of control theory, which present barriers to their practical use. For this reason, classical controllers are still widely used in industrial applications due to their simple structure and ease of design.

One of the authors has developed a simple and practical control method for PTP positioning.[17,18] The controller has a simple structure and its design is simple in that it does not require an exact dynamic model or its parameters. The control method has been used on positioning mechanisms with friction characteristics and was shown to be quite useful.

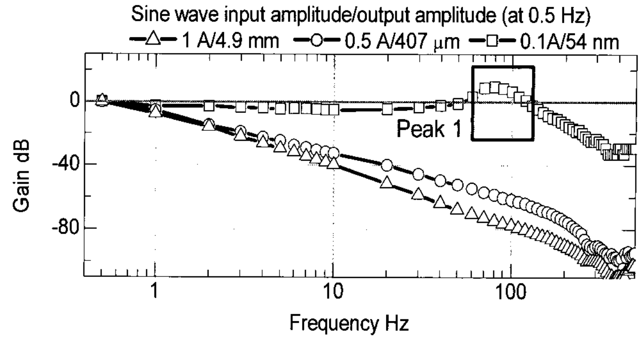
This paper describes the problem of ultraprecision positioning with a conventional ball screw mechanism that exists in a range of microdynamics, and its practical solution. In Section 2, we explain experimental ball screw mechanisms and examine their dynamic characteristics. In practical application, the movable mass of the precision positioning mechanism often changes. In this paper, we examine two ball screw mechanisms with different movable masses. In Section 3, we introduce the nominal characteristic trajectory following (NCTF) control method as a practical solution, and examine its positioning performance. We also describe the problems in using a heavyweight mechanism in ultraprecision positioning. We propose a practical solution to the problem and experimentally confirm its usefulness in Section 4. Finally in Section 5, we present our conclusions.

2. Dynamic Characteristics of Ball Screw Mechanisms

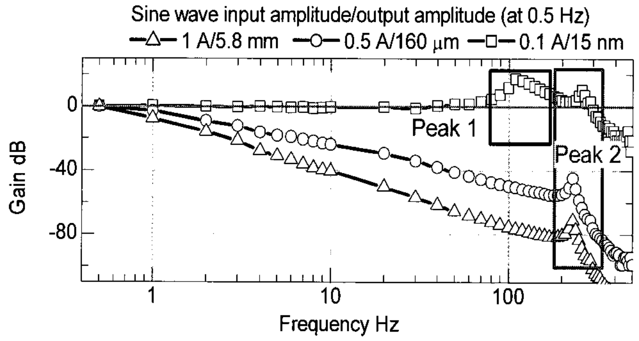
2.1 Experimental Mechanisms

We used two ball screw mechanisms in this study as shown in Fig. 1. The ball screw mechanisms have similar structures, and differ only in their table masses. The mechanism with the heavy additional mass is referred to as the heavyweight mechanism, while the other is referred to as the lightweight mechanism. The value of the additional mass fixed on the table of the heavyweight mecha nism is similar to that of the mass of the lightweight mechanism.

Each mechanism comprises a table, two linear ball guides, a ball screw, a coupling, and a DC motor. Friction in the mechanisms comes from the DC motor, the preloaded double-nut, the linear ball guides, and the ball bearings supporting the screw shaft. The stiffness of the coupling is very high and the deformation of the coupling is negligible. The influence of the stiffness between the screw and the nut depends on the mass of the table. The motor is powered by pulse-width modulated amplifiers limited to 45 V and 6 A. The controller sampling frequency is 5 kHz and the feedback position is determined by a laser position sensor (Agilent 10897B) with a resolution of 1.24 nm. The lead of the ball screw is 2 mm/rev.



(a) Lightweight mechanism



(b) Heavyweight mechanism

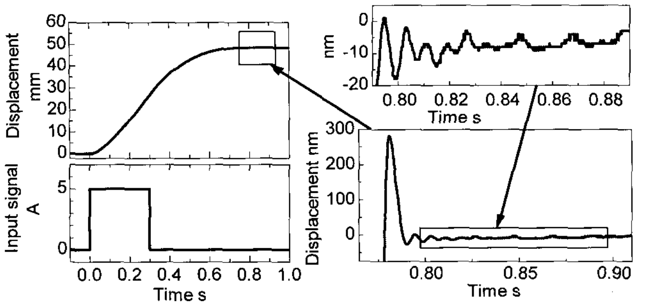
Fig. 2 Open-loop frequency characteristics of ball screw mechanisms

2.2 Dynamic Characteristics

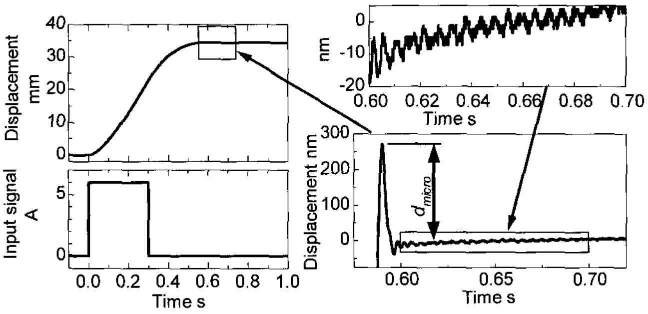
Figures 2(a) and 2(b) show the magnitude of the open-loop frequency responses of the lightweight and heavyweight mechanisms, respectively. In both mechanisms, three sine wave input amplitudes were used: 1 A, 0.5 A, and 0.1 A. For each sine wave amplitude, the frequency of the input signal ranged from 0.5 Hz to 500 Hz. The output gains were normalized to 0 dB at 0.5 Hz. As the results show, both mechanisms presented a resonant peak close to 100 Hz, referred to as Peak 1. Peak 1 was observed only with a sine wave input amplitude of 0.1 A. With this input, the amplitude of the displacement was 54 nm for the lightweight mechanism and 15 nm for the heavyweight mechanism. The resonant peak observed with the small sine wave input amplitude was due to the nonlinear springlike microdynamic behavior. This behavior is dominant in the microdynamic characteristic of the mechanism. The dynamic model is often expressed as a second-order system having the same resonant peak.[6,13,15]

In Fig. 2(b), the frequency response of the heavyweight mechanism shows a second resonant peak close to 220 Hz, referred to as Peak 2. Unlike Peak 1, which was observed only in the microdynamic range, Peak 2 was present for all three input amplitudes, and was due to the low stiffness of the ball screw. Peak 2 was significant across a narrower working range. In the microdynamics range where the Peak 1 was observed, Peak 2 also greatly influenced the mechanism characteristics. This indicates that low stiffness of the power transmission makes the control system with the mechanism sensitive to the change of the movable mass. Friction generally increases the damping ability of positioning systems, but the springlike behavior did not provide sufficient damping ability for ultraprecision positioning.

The open-loop responses of the ball screw mechanisms to rectangular inputs are shown in Fig. 3. Figure 3(a) shows the response of the lightweight mechanism. An input amplitude of 5 A and a duration of 0.3 s produced a final displacement of 48.5 mm. The magnified view shows the behavior of the table immediately before stopping (the origin is shifted to the final displacement). The displacement peaked at 0.781 s, and was a result of the springlike



(a) Lightweight mechanism



(b) Heavyweight mechanism

Fig. 3 Open-loop step responses of the two ball screw mechanisms

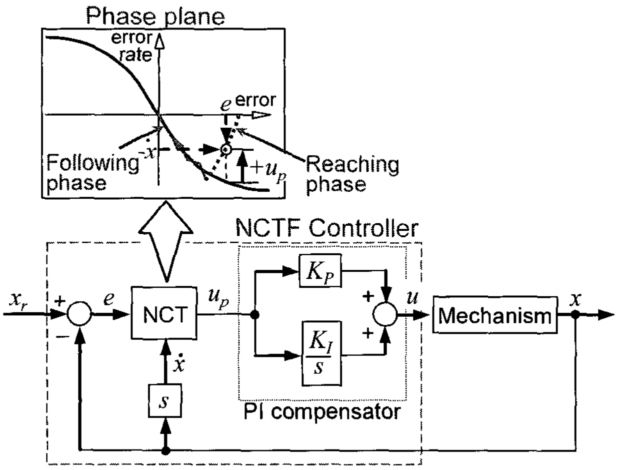


Fig. 4 Basic NCTF control system

behavior. Thus the range in which the springlike behavior was observed is equal to or wider than the difference between the peak displacement and the final displacement. The vibration observed after the peak displacement also resulted from the springlike behavior. The frequency was about 120 Hz and the vibration attenuated quickly.

Figure 3(b) shows the response of the heavyweight mechanism. An input amplitude of 6 A and a duration of 0.3 s produced a final displacement of 34.5 mm. The displacement peaked at 0.59 s. As in Fig. 3(a), the peak displacement was the result of springlike behavior. However, the residual vibration after the peak displacement was the result of low stiffness of the ball screw; its frequency was about 223 Hz. As shown in Fig. 3(b), the vibration continued for a long time in the microdynamic range. For ultraprecision positioning and low sensitivity to the change in movable mass, the control system requires a high damping ability at the frequency that results from the low ball screw stiffness.

3. Basic NCTF Control Method for PTP Positioning

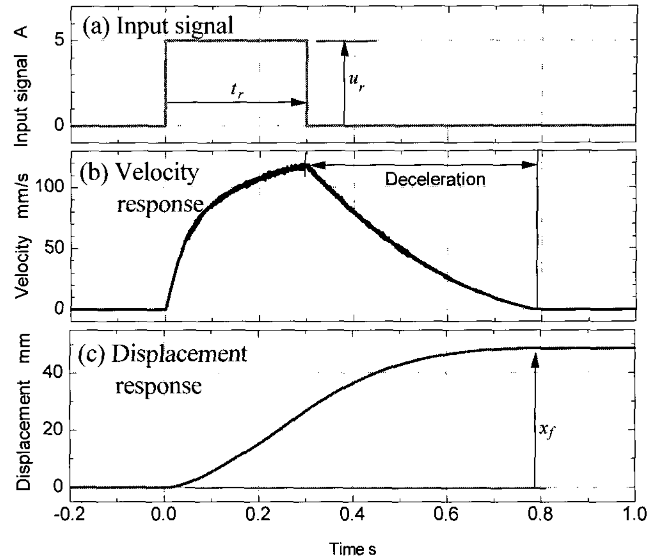


Fig. 5 Open-loop response of the lightweight mechanism for constructing the NCT

We used the NCTF control method on the ball screw mechanisms for ultraprecision positioning because the NCTF controller is easy to design and has a simple structure. In this section, we introduce the concept and the positioning performance of the basic NCTF control method.

3.1 Concept

Figure 4 shows the structure of the NCTF control system for PTP positioning. The controller consists of a nominal characteristic trajectory (NCT) and a proportional-integral (PI) compensator. The NCT represents the deceleration motion of the table in positioning, which significantly influences the positioning performance. The role of the PI compensator is to make the mechanism motion follow the NCT, finishing at the origin of the phase plane. The output of the NCT is a signal u_p, which is the difference between the actual error rate of the mechanism $(-\dot{x})$ and the error rate of the NCT. On the phase plane, the table motion is divided into a reaching phase and a following phase. During the reaching phase, the compensator controls the table motion to achieve the NCT. The next step is the following phase, in which the PI compensator causes the mechanism motion to follow the NCT, leading it back to the origin of the phase plane.

As described in Section 1, controllers should be easy to design and understand in practical use. Some model-based controllers such as the sliding mode controller or the seeking mode controller in hard disk drives[19] have a similar structure and include the reference element of the deceleration motion. However, those controller designs require exact dynamic model information and a knowledge of control theory, which makes them difficult to use in practice. To avoid this problem, the NCT and the PI compensator parameters are determined in the following manner:

1) The NCT representing the deceleration motion is constructed on the phase plane using open-loop displacement and velocity responses of the actual mechanism during the deceleration. This means that the construction does not require the exact dynamic information of the mechanism, and is much simpler than the model-based controllers described above. The mechanism characteristics, including nonlinear effects caused by friction and saturation elements, are reflected in the NCT. Thus, the actual mechanism has the ability to follow the constructed NCT.

2) The practical stability limit of the actual mechanism can be found by driving the mechanism with only the NCT and

proportional elements connected. The PI compensator is determined using the practical stability limit, the simple step responses, and the NCT information.

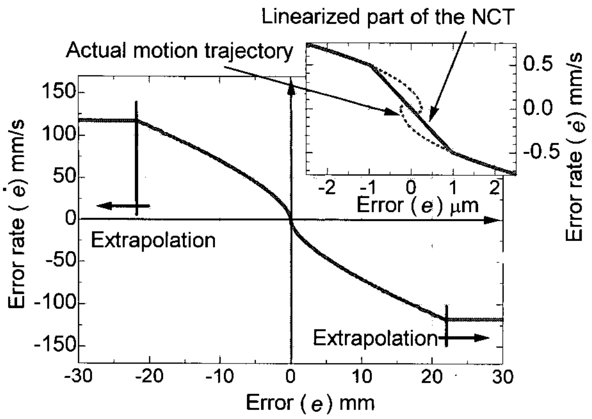


Fig. 6 NCT constructed from the open-loop response

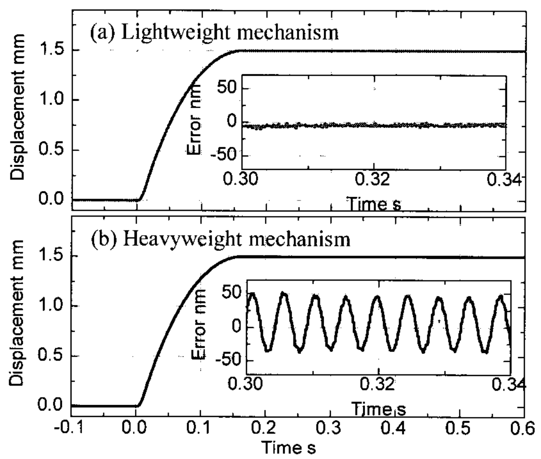


Fig. 7 Step responses of the NCTF control systems designed based on the procedure in Subsection 3.2

A conditional freeze integrator[20] is used to reduce the overshoot caused by the integrator windup. The NCTF controllers have been designed to drive mechanisms such as a rotary motion mechanism or a linear motor with a linear ball guide directly. The simulated and experimental results showed that the NCTF control systems had better performance than the conventional proportional-integral-derivative control systems and the proportional-derivative control system with a disturbance observer.[20]

3.2 Design Procedure

The NCTF control system was designed according to the following procedure[21]:

1) NCT construction from an open-loop experiment

Figure 5 shows the open-loop response of the ball screw mechanism. The NCT was constructed from this open-loop response during the deceleration of the mechanism. Figure 6 shows the NCT as constructed. The horizontal scale represents the measured displacement during the deceleration (the final displacement x_f is shifted to zero). The vertical scale indicates the corresponding velocity.

When the step input is applied to mechanisms showing the springlike behavior, the overshoot is observed and causes a whorl-like trajectory near the origin of the phase plane.[18] Since this whorl-like trajectory has negative effects in positioning, it should be eliminated. To do this, the NCT was linearized with a straight line close to the origin. The whorl-like actual motion trajectory and its linearized trajectory are shown in the magnified view of Fig. 6.

2) Determination of the stability limit of the control system (practical stability limit)

The practical stability limit is determined by driving the mechanism with only the NCT and the proportional element. The value of the proportional gain is increased until continuous oscillations are generated, indicating the onset of instability. The practical stability limit ζ_{prac} is given as

$$\zeta_{prac} = K_{Pu}\left(\frac{K\alpha}{2\omega_n}\right) \qquad K = \frac{x_f}{u_r t_r}. \tag{1}$$

The parameter α represents the gradient of the NCT near the origin on the phase plane. For our experimental ball screw mechanisms, Eq. (1) became

$$\zeta_{prac} = 2.4\left(\frac{32.3\times 505}{2\omega_n}\right) \ . \tag{2}$$

3) Adjustment of the PI compensator gains within the stable operation region

Since the practical stability limit is determined experimentally, it includes the effects of nonlinear friction and the sampling delay of the digital controller. As described in,[17,18] the compensator gains are calculated from

$$K_P = \frac{2\zeta\omega_n}{K\alpha} \qquad K_I = \frac{\omega_n^{\ 2}}{K\alpha} \tag{3}$$

The parameters ζ and ω_n are selected to satisfy the following inequality:

$$\zeta < K_{Pu}\left(\frac{K\alpha}{2\omega_n}\right) \tag{4}$$

Larger values of ω_n increase the positioning accuracy and the robustness of the control system to disturbance forces but tend to decrease the system stability. In this paper, the margin of safety of the design was set to 60%. K_p and K_i were selected to be 0.98 As/mm and 156.78 A/mm, respectively.

3.3 Positioning Characteristic

Figure 7 shows the step responses of the NCTF control systems for the lightweight and the heavyweight mechanisms. The controllers in both control systems were identical. The NCT and the PI compensator parameters were determined from the characteristics of the lightweight mechanism. The experimental results show that the NCTF control system with a lightweight mechanism had a positioning accuracy of better than 10 nm. The heavyweight mechanism macroscopically shows the same waveform as the lightweight mechanism, but the heavyweight mechanism clearly experiences microscopic residual vibrations in a steady state, which degrade the positioning accuracy. In addition, the frequency of the vibrations tended to increase from 160 Hz to 250 Hz.[22]

4. Solution of the Vibration Problem

4.1 Improvement of the NCTF Controller

As described in Section 2, the heavyweight mechanism had two resonant frequencies. One of these was due to the springlike behavior of the mechanism and was also present in the lightweight mechanism. The other resonant frequency was due to the low stiffness of the ball screw. Both resonant frequencies were beyond the bandwidth of the control system. The mechanism has a low response and it is difficult for the control system with the mechanism to have an enough bandwidth to cover the frequencies. The vibration at the first resonant frequency was quickly attenuated, but the vibration at the second

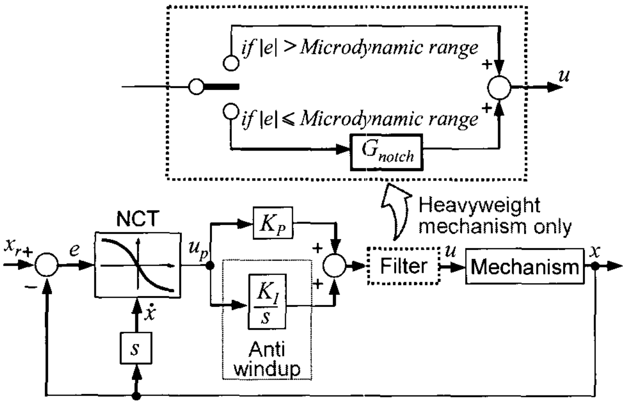
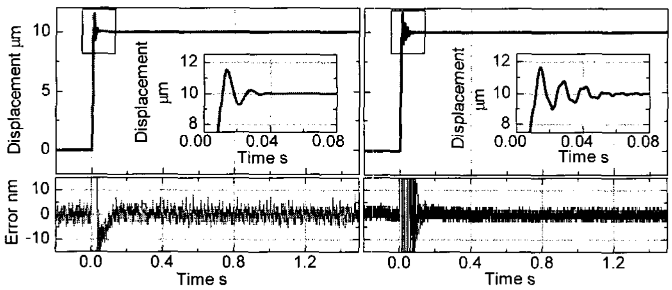
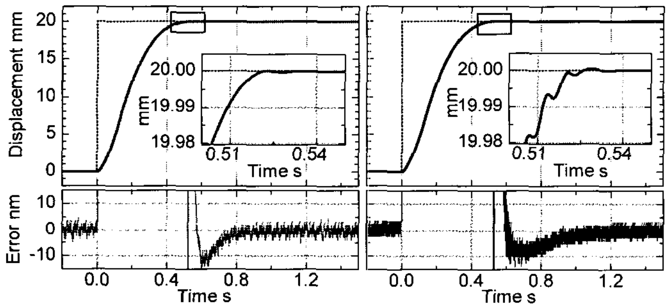


Fig. 8 NCTF control system with the conditional notch filter



(a) Lightweight mechanism (b) Heavyweight mechanism

Fig. 9 Responses of the improved NCTF control systems to a 10 μm step input



(a) Lightweight mechanism (b) Heavyweight mechanism

Fig. 10 Responses of the improved NCTF control systems to a 20 mm step input

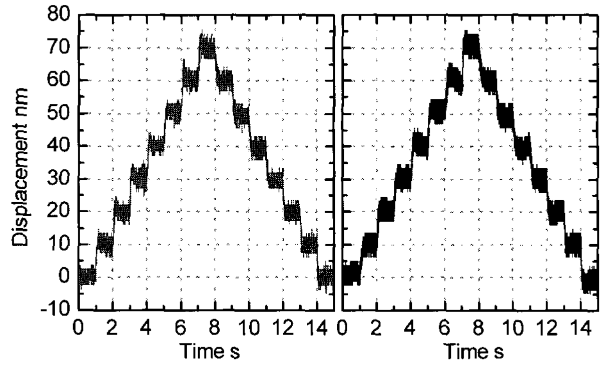
resonant frequency greatly influenced the positioning performance in the microdynamic range. In addition, this second frequency tended to change.

We inserted a notch filter in the NCTF control system to reduce the vibration at the second frequency as shown in Fig. 8. Since the ease of design and understanding is an important characteristic of the NCTF control method, this notch filter needs to be designed easily without requiring the exact model information of the mechanism.

Equation (5) is the transfer function of the notch filter:

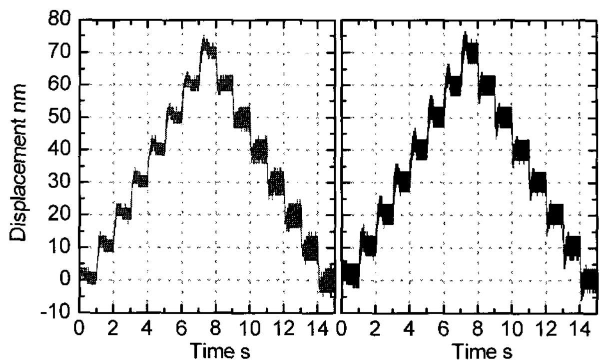
$$G_{notch} = \frac{s^2 + \omega_{notch}^{\ 2}}{s^2 + 2\zeta_{notch}\omega_{notch}s + \omega_{notch}^{\ 2}}. \quad (5)$$

The notch filter has no negative influence on the response in the microdynamic range, but does so in the macrodynamic range. Thus, the active range of the notch filter is limited to the microdynamic range. This filter is referred to as the conditional notch filter. The active range and the natural frequency of the filter are easily determined from the open-loop step response. The difference between the peak and the final displacements d_{micro} shown in Fig. 3(b) is less



(a) Lightweight mechanism (b) Heavyweight mechanism

Fig. 11 Responses of the NCTF control systems to stepwise inputs before warm-up



(a) Lightweight mechanism (b) Heavyweight mechanism

Fig. 12 Responses of the NCTF control systems to stepwise inputs after warm-up

than or equal to the range of the microdynamics. Thus, d_{micro} is used as the active range of the conditional notch filter. The angular frequency of the residual vibration is used as ω_{notch}. Although the frequency of the residual vibration tends to change, a sufficiently large ζ_{notch} can eliminate this. For the heavyweight mechanism, ζ_{notch} was set to 3.

4.2 Effect of Improvement

Figures 9 and 10 show the responses of the NCTF control systems to 10-μm and 20-mm step inputs. These show that both ball screw mechanisms exhibited very similar performances. The notable difference occurred during the transient response. The lightweight mechanism did not show any vibration during the transient motion. The heavyweight mechanism, however, had a vibrating transient motion caused by the additional mass. The vibration was not observed in the steady state because the conditional notch filter was active. The NCTF controller showed robustness to payload variation (4.9 times higher in the heavyweight mechanism) and also robustness to friction variation (2 times larger in the heavyweight mechanism).

Figure 11 shows the responses of the control systems with the lightweight and heavyweight mechanisms to stepwise inputs before warm-up. The first step height was 10 nm. The positioning accuracy of the control system with the heavyweight mechanism was similar to that with the lightweight mechanism. Amplitudes of residual vibrations of the two mechanisms were about 5 nm. Figure 12 shows the responses of the control systems with the lightweight and heavyweight mechanisms to stepwise inputs after warm-up. After the warm-up, the Coulomb friction and viscous friction of the lightweight mechanism were reduced by 13% and 28%, respectively. The two types of friction in the heavyweight mechanism were also reduced by similar percentages. In spite of the changes in friction, a positioning resolution of better than 10 nm and the amplitude of the residual vibrations were maintained, showing that the designed NCTF controller with the conditional notch filter is robust against friction variations.

5. Conclusions

We described the problem of ultraprecision positioning with a ball screw mechanism and introduced a solution. We compared the characteristics of two ball screw mechanisms with different movable masses to clarify the influence of the movable mass change on positioning performance. The experimental results show that the vibration resulting from the low stiffness of the ball screw degraded the positioning performance of the heavyweight mechanism in the microdynamic range.

We selected the NCTF controller for ultraprecision positioning of the ball screw mechanism. It has the advantages of ease of design and simple structure, which are important for practical use. The basic NCTF control system with the lightweight mechanism resulted in ultraprecision positioning performance, but this was not the case with the heavyweight mechanism. To overcome this problem, we improved the NCTF controller by adding the conditional notch filter. We used as simple open-loop experiment to design the NCTF controller and the conditional notch filter, without requiring exact parameter identification. Despite the differences in payload and friction, both mechanisms showed similar performances, demonstrating the high robustness of the NCTF controller and the effectiveness of the conditional notch filter. The experimental results showed that the NCTF control system with the conditional notch filter achieved ultraprecision positioning with a positioning accuracy of better than 10 nm. This positioning performance was achieved independent of the reference step input height.

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