Application of a Disturbance Observer for a Relative Position Control System

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***Abstract*—This paper presents an application of a disturbance observer for a relative position control system. In this system, since the prefixed motion profile is not defined *ex ante*, the acceleration state which is generated based on the profile is also not available. Therefore, feedforward acceleration controls cannot be used, and the position control performance is restricted solely by the bandwidth of the position controller. To enhance the control performance, disturbance observers can be utilized actively. The proposed method considers the position reference just as the result of a disturbance. Therefore, the relative position control can be performed by a disturbance observer as well as a position controller. As a result, the position control performance of the proposed method has been enhanced by up to 30% compared with that of the conventional method. The feasibility of the proposed method has been verified by experimental results using a high-precision linear motion control system as well as by analysis based on Bode plots.**

***Index Terms*—Disturbance observer, high-precision control, linear permanent-magnet synchronous motor, motion control, relative position control.**

I. Introduction

MOTION CONTROL technologies [1], [2] are widely used in industrial applications such as X–Y table machine tools, joint control of industrial robots, and so on. Nowadays, motion control systems have increasingly higher control bandwidth, lower settling time, and better disturbance-rejection performance, all leading to improved productivity. Moreover, based on the enhanced performances, applications are expanding to emerging areas such as the photolithography stage of semiconductor manufacturing processes, haptic motion control for home robots, precision motion control stages with 6 DOF, and surgical robot systems [3]–[7].

There has been much research [1], [8]–[11] on motion control techniques, using several control algorithms and additional sensors. Some papers [1], [8] adopt well-known linear control algorithms for disturbance observers with some modifications. A back-stepping nonlinear controller for motion control systems has been developed [9], and an H^{∞} control technique was used to achieve a robust tracking performance [10]. In [11], the disturbance observer utilized an additional acceleration sensor to increase the control bandwidth.

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Among them, the use of an observer is a highly attractive and effective method to improve the performance of the motion control due to its simple structure. Thus, it has been widely adopted in various application areas. The disturbance observer is a particularly helpful tool in the motion control system since it could reject not only an external disturbance but also an internal model parameter variation. The control performance can be enhanced by rejecting the external disturbance when the disturbance occurs and/or by compensating the internal model parameter variation even when the external disturbance does not occur. There has been significant research into these topics [1], [8], [11]–[14].

Generally, motion control systems have prefixed motion profiles. Therefore, the acceleration state of the motion can be generated based on the prefixed profile. Using the feedforward control with the acceleration state, the position control performance can be significantly enhanced even if the bandwidth of the position controller is not large enough due to some problems such as mechanical resonance.

However, relative position control systems such as surgical robots cannot use the prefixed profile since the position reference is simultaneously generated according to the motion of the target object. For example, surgical robot systems can be operated by a medical doctor. In this case, even though the acceleration state can be obtained through the second-order derivative to enhance the position control performance, practically, it may not be used because of the signal noise from the derivative process. Therefore, the feedforward control cannot be utilized, and the performance of the motion control system is restricted only by the bandwidth of the position controller. Even if the skill of the doctor is outstanding, the performance and productivity of the surgical robot are severely limited solely by the bandwidth of the control system.

This paper presents an active application of a disturbance observer for a relative motion control system. Usually, the main function of the disturbance observer is to reject external disturbances and/or compensate internal model errors. However, the proposed method intentionally exploits the disturbance observer so that the performance of a relative motion control is enhanced while the conventional functions are maintained.

II. Analysis of Conventional Motion Control System

The conventional motion control system simply consists of a position controller and a disturbance observer, as shown

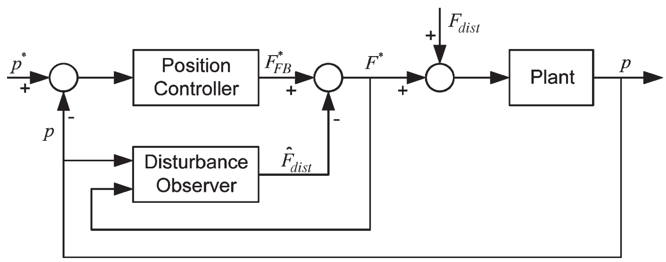


Fig. 1. Block diagram of the conventional motion control system.

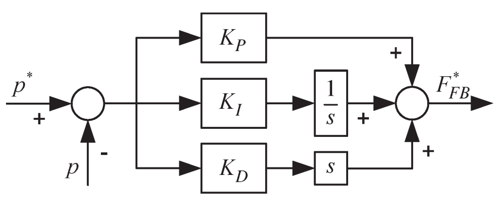


Fig. 2. Block diagram of the position controller.

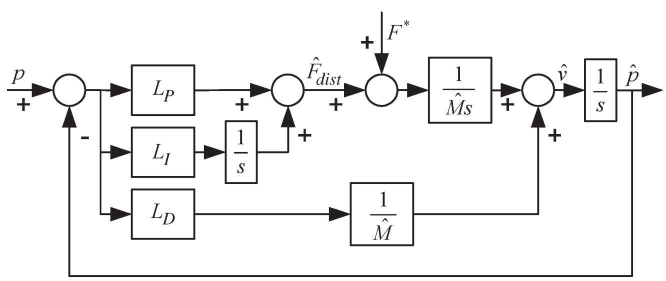


Fig. 3. Block diagram of the disturbance observer.

in Fig. 1 [8]. In addition, the position controller is usually a proportional–integral–derivative (PID) controller, which is a cascaded form of position and speed controllers, as shown in Fig. 2. In order to enhance the control performance, the disturbance observer is adopted to compensate an external disturbance. The block diagram of the conventional disturbance observer [14] is shown in Fig. 3.

The gains of the position controller can be selected by various manners. In this paper, for analysis, the PID gains of the conventional position controller are selected as follows considering a damping factor:

$$\begin{aligned} &K_P = K_{\mathrm{pp}}K_{\mathrm{pv}} + K_{\mathrm{iv}} \quad K_I = K_{\mathrm{pp}}K_{\mathrm{iv}} \quad K_D = K_{\mathrm{pv}} \\ &K_{\mathrm{pv}} = \hat{M}\cdot\omega_{\mathrm{sc}} \quad K_{\mathrm{iv}} = 0.2\cdot K_{\mathrm{pv}}\cdot\omega_{\mathrm{sc}} \quad K_{\mathrm{pp}} = \omega_{\mathrm{sc}}/9 \end{aligned} \tag{1}$$

where $\hat{M}$ is the estimated mass of the plant, K_{pv} and K_{iv} are the proportional and integral gains of the speed controller, K_{pp} is the proportional gain of the position controller, ω_{sc} is the bandwidth of the speed controller, and K_P, K_I, and K_D are the P, I, and D gains of the PID controller, respectively.

The gains of the observer shown in Fig. 3 can also be selected by various alternative methods. By locating the poles of the observer as triple roots, the gains of the conventional observer are selected as

$$L_P = 3\cdot\omega_{\mathrm{ob}}^2\cdot\hat{M} \quad L_I = \omega_{\mathrm{ob}}^3\cdot\hat{M} \quad L_D = 3\cdot\omega_{\mathrm{ob}}\cdot\hat{M} \tag{2}$$

where L_P, L_I, and L_D are the P, I, and D gains of the observer and ω_{ob} is the pole of the observer.

The relative position control system can be described as shown in Fig. 4. The object "A" is the target one, and the object "B" is the controlled one to track the motion of A. Therefore, the position of A, which is p^*, is the position command for B. Fig. 4(a) shows the steady state of the relative motion control. At this instant, p is equal to p^*. For convenience, p^* and p at this instant can be considered as a zero position.

When A moves as shown in Fig. 4(b), p^* becomes d. In the conventional method, the position controller moves B for p to be equal to d. When the relative motion control of the conventional method is performed based on the block diagram in Fig. 1, the control performance can be analyzed as follows.

The plant of this control system is an accurate motion controller actuated by an inverter-driven permanent-magnet linear synchronous motor (LSM). Therefore, the actual plant model consists of a mechanical system, a linear motor, and an inverter system. The mechanical system is comprised of mass, friction, and so on. The linear motor is comprised of electric parameters, mechanical force, and so on. Furthermore, the inverter system drives the linear motor. However, the frequency range which motion control applications deal with is below 100 Hz. Therefore, the inverter system and the linear motor system can be considered to be ideal at that range. In this paper, the whole plant models are not considered. This paper dealt with the only mechanical model with mass quantity for simplicity.

Supposing that $\hat{M}$ is equal to M for simplicity, the transfer functions of the relative position control performance p/p^* can be described as

$$\frac{p}{p^*} = \frac{K_D s^2 + K_P s + K_I}{Ms^3 + K_D s^2 + K_P s + K_I}. \tag{3}$$

As shown in (3), the disturbance observer does not affect the performance of the relative position control if the model of a plant is accurate. However, if there are some errors in the plant model, the disturbance observer can compensate them, and it is able to enhance the position control performance compared with the case where it is not adopted. Even in this case, however, the disturbance observer merely compensates the internal model error and does not affect the performance of the relative position control.

III. Proposed Method for a Relative Position Control System

Fig. 4(b) shows the motion of the target object A. At this instant, A moves and B stays without any movement. However, this motion can be understood in another point of view. At the viewpoint of A, A does not move and B is translated by a certain disturbance, as shown in Fig. 4(c). At this viewpoint, the relative position reference can be thought of as the result of the disturbance. Although an actual disturbance force does not exist, the disturbance observer considers the movement of A as the result of the disturbance and works to compensate the motion from the disturbance. The block diagram of the proposed relative position control can be described as shown

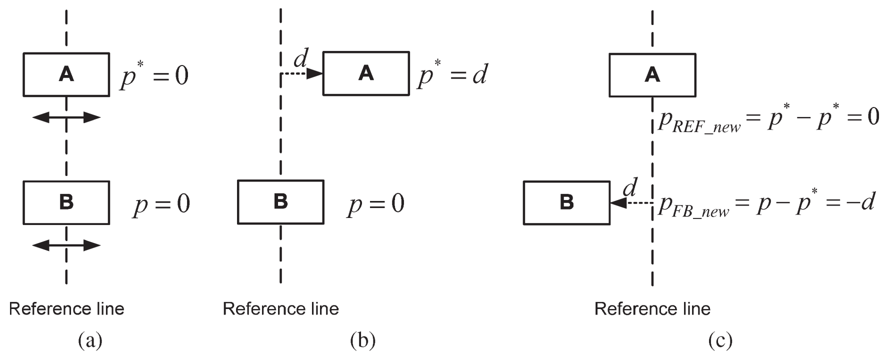


Fig. 4. Relative position control. (a) Steady state of the relative position control. (b) Transient state in the point of view of the conventional method and (c) transient state in the point of view of the proposed method.

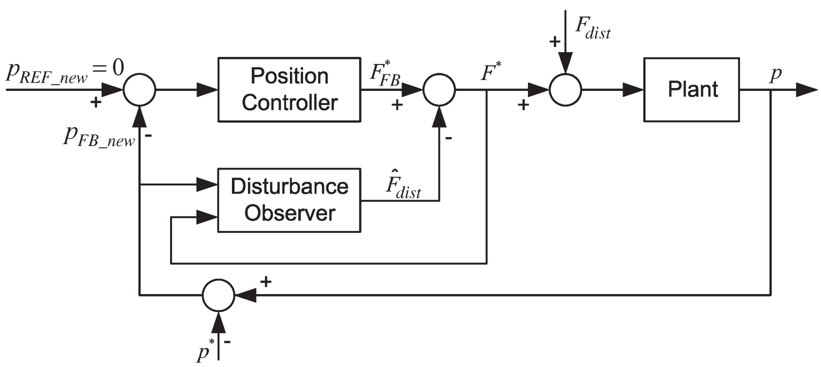


Fig. 5. Block diagram of the proposed method.

in Fig. 5. Therefore, since the problem of relative position control is changed to the problem of disturbance rejection as well as position control itself, the proposed method can take an additional advantage of the disturbance observer for the relative position control while the conventional functions are maintained. As a result, the performance of the relative position control can be enhanced by the proposed method.

The new position reference and position feedback of the proposed method can be described as (4) and (5), respectively. They can be generated in the viewpoint of the target object A

$$p_{\mathrm{REF\_new}} = p^* - p^* = 0 \tag{4}$$

$$p_{\mathrm{FB\_new}} = p - p^* \tag{5}$$

where $p_{\mathrm{REF\_new}}$ and $p_{\mathrm{FB\_new}}$ are the new position reference and feedback of the proposed relative position controller, respectively.

For the analysis of the proposed method, its block diagram can be rearranged. Because of the force feedforward term of the disturbance observer in Fig. 3, F^* is described as (6) from Fig. 1, and the block diagram of the disturbance observer in Fig. 3 can be modified as shown in Fig. 6. The output of the observer controller does not directly feed the plant model of the observer

$$F^* = F^*_{\mathrm{FB}} - \widehat{F}_{\mathrm{dist}}. \tag{6}$$

For a better understanding, the proposed block diagram can be rearranged into the diagram in Fig. 7 which starts with the position command and ends with the plant. It can be seen

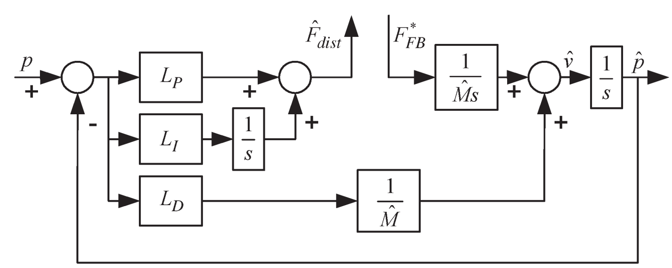


Fig. 6. Block diagram of the disturbance observer.

that the disturbance observer of the proposed system turns into an open-loop force estimator with its own PI controller. This structure provides a better force command which includes some acceleration effects for position regulation. This results in a force command which is better than that of the PID position controller. Using the proposed method, the position control performance can be enhanced while the performance of the disturbance rejection is still kept up.

Supposing that $\hat{M}$ is equal to M, the transfer functions of the proposed relative position control performance p/p^* can be described as

$$\begin{aligned}\frac{p}{p^*} =& \frac{Ms^2(K_Ds^2+K_Ps+K_I+L_Ps+L_I)}{(Ms^3+K_Ds^2+K_Ps+K_I)(Ms^3+L_Ds^2+L_Ps+L_I)} \\ &+ \frac{(K_Ds^2+K_Ps+K_I)(L_Ds^2+L_Ps+L_I)}{(Ms^3+K_Ds^2+K_Ps+K_I)(Ms^3+L_Ds^2+L_Ps+L_I)}.\end{aligned} \tag{7}$$

As described in (7), the position control is performed by the disturbance observer as well as the position controller. Under

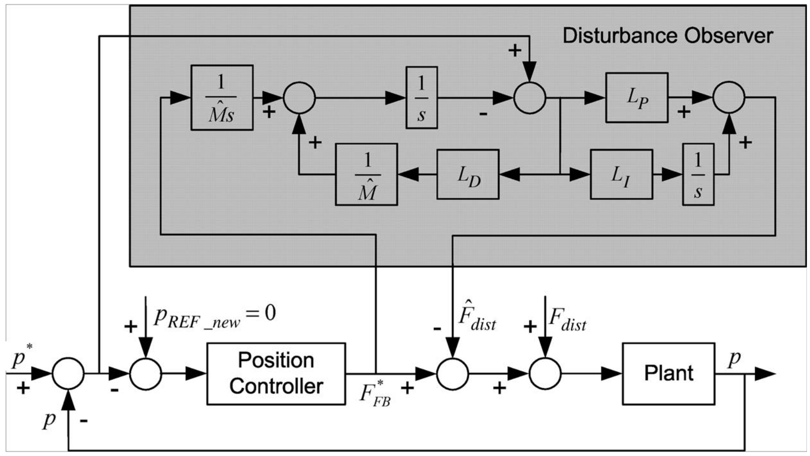
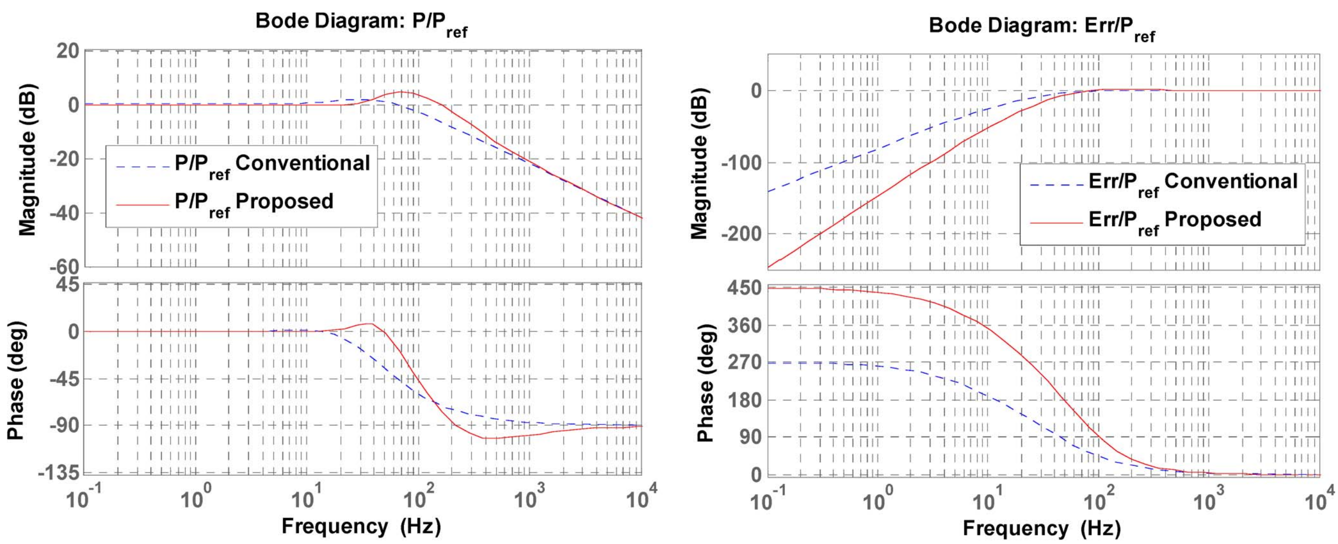


Fig. 7. Rearranged block diagram of the proposed method.



(a) (b)

Fig. 8. Comparison of the position control performance. (a) p/p^* and (b) Err/p^*.

the condition that both $\omega_{\rm sc}$ and $\omega_{\rm ob}$ are set to 80 Hz and that M is equal to 3.2 kg, Fig. 8 shows the comparison of the relative position control performance by Bode diagram between the conventional method and the proposed method using (3) and (7). Fig. 8(a) shows p/p^*, and Fig. 8(b) shows Err/p^*, which means the transfer function of the position error $(Err \equiv p^* - p)$ with respect to the position reference. From Fig. 8, it can be concluded that the performance of the position control with the proposed method can be improved significantly even though the modification is quite simple. Under the condition that the bandwidths of the control system are limited due to mechanical problems such as resonances, this improvement can make a great difference on the performance of the relative position control system.

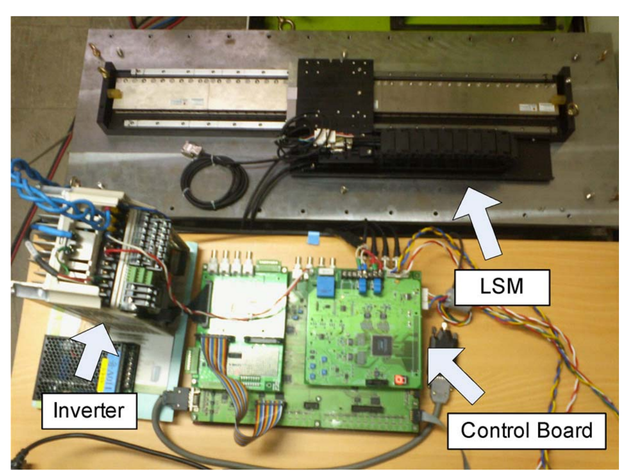


Fig. 9. Experimental setup.

IV. Experimental Results

Experiments using a high-precision motion stage comprising a permanent-magnet LSM and a linear encoder were performed. Fig. 9 shows the experimental setup. The parameters of the LSM under test are listed in Table I. The resolution of the linear encoder is 40 nm, and a commercial inverter has been used. The nominal parameters of the inverter are listed in Table II. The switching frequency was set to 10 kHz, and the current control bandwidth was 2 kHz using a high-bandwidth

TABLE I
LINEAR SYNCHRONOUS MACHINE PARAMETERS

Parameter	Value [Unit]
Rated force	73 [N]
Rated current	1.2 [A_{rms}]
Rated speed	1.5 [m/s]
Maximum force	220 [N]
Maximum current	3.5 [A_{rms}]
Stator resistance	12 [Ω]
Stator inductance	8.6 [mH]
Pole pitch	22.5 [mm]
Mass of mover	3.2 [kg]
Force Constant	47.2 [N/A]

TABLE II
NOMINAL PARAMETERS OF INVERTER

Parameter	Value [Unit]
Rated power	1.2 [kVA]
Rated current	3.2 [A_{rms}]
DC link Voltage	310 [V]

current controller [15]. To implement a high-bandwidth current controller, the control board used a 32-b floating-point DSP, namely, TMS320VC33, and a 12-b 20-MHz sampling analog-to-digital converter, ADS805.

Real mechanical systems have resonance frequencies related with plant structure and material stiffness. However, the exact resonances are not easy to be analyzed. From finite-element-method analyses based on the design data, the resonance frequencies can be obtained but they are not accurate either. If the plant has some mechanical resonance, the controller which has high gains may make the plant unstable or underdamped. In these cases, the gain or bandwidth of the controller should be reduced to make the system stable. For the plant in this paper, both ω_{sc} and ω_{ob} were set to 80 Hz to avoid such a mechanical resonance. The control performance was found out to be underdamped when the bandwidth was higher.

The control results of the conventional PID method strongly depend on the system parameters. Therefore, before the experiments started, the parameters of the system such as the mechanical and electrical parameters were identified as exactly as possible. The identified results were listed in Table I.

To evaluate the performance of the controller, its bandwidth should be considered. If the gains of the controller are tuned without consideration of the bandwidth, the performance of specific conditions may be enhanced. However, the analysis of the controller is not only difficult but also incorrect. The controller might be sensitive to noise and reveal an underdamped response.

In this paper, to compare the control performance impartially between the conventional and proposed algorithms, the gains of the two algorithms are determined with the same method as (1) and (2) based on the bandwidth concept as well as the

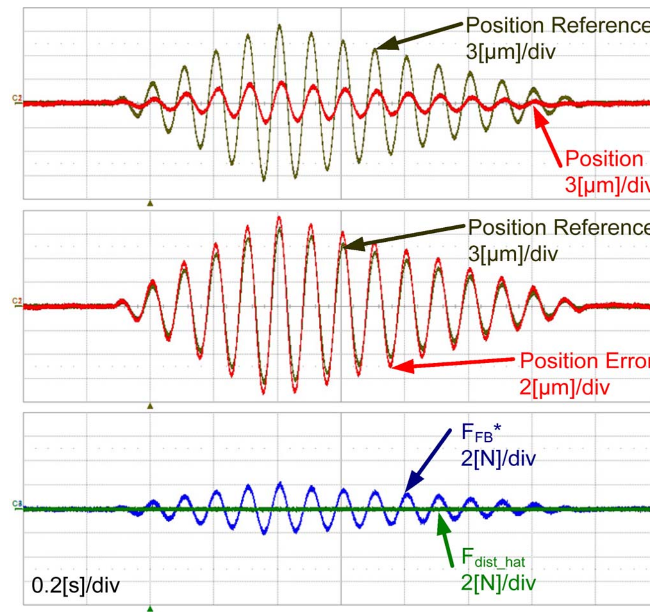


Fig. 10. Relative position control performance of the conventional method without the disturbance observer; the frequency of the position reference is 10 Hz.

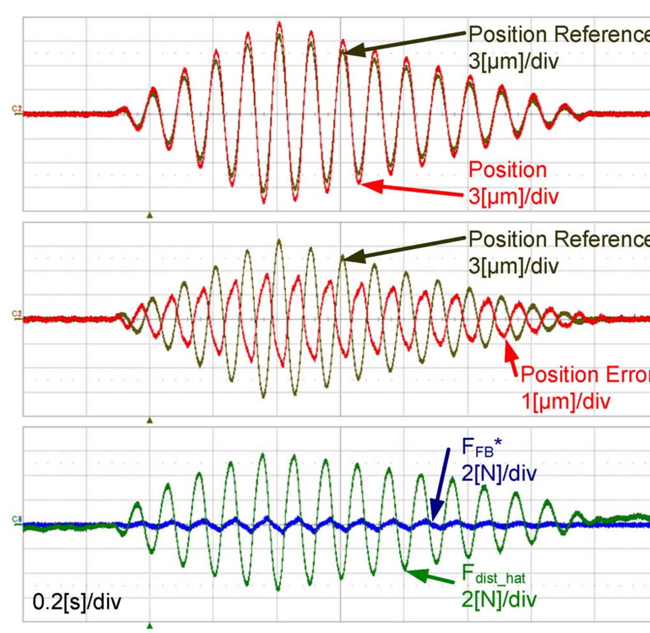


Fig. 11. Relative position control performance of the conventional method with the disturbance observer; the frequency of the position reference is 10 Hz.

carefully identified parameters. These are the common methods which are widely used. The only difference between the two methods is the structure of the overall controller. Therefore, the experimental results can verify the effectiveness of the proposed algorithm.

Figs. 10–12 show the experimental results when the relative position control algorithm was applied. The frequency of the relative position reference was 10 Hz, and the maximum displacement of the relative position reference was 10 μm. The feedforward control based on the acceleration state was not used since the motion profile could not be used in the relative position control system. Each figure shows five values.

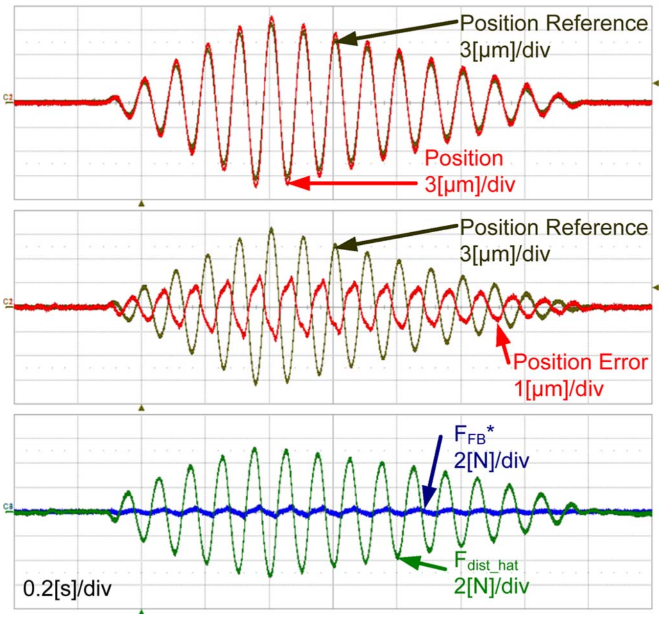


Fig. 12. Relative position control performance of the proposed method; the frequency of the position reference is 10 Hz.

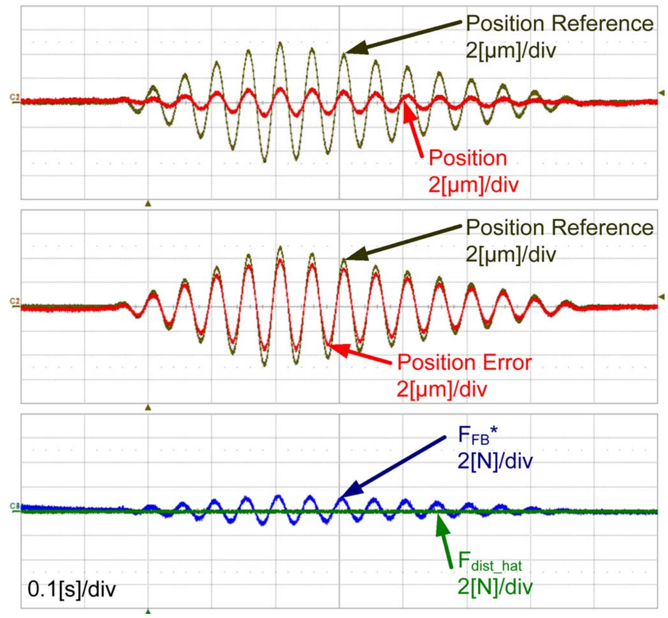


Fig. 13. Relative position control performance of the conventional method without the disturbance observer; the frequency of the position reference is 20 Hz.

From top to bottom, the position reference, the position, the position error, the force reference from the position controller, and the force which the disturbance observer estimated and compensated are shown.

Fig. 10 shows the relative position control performance of the conventional method without the disturbance observer. The position could not track the reference, and the position control performance was not satisfactory. The maximum position error is about 7.5 μm. According to the Bode analysis, as shown in Fig. 8(b), the position error should be reduced by -20 dB, which is one-tenth of the position references. However, the position error from the experimental results was reduced by -15 dB. The difference between the Bode analysis and the experimental result may come from the parameter and plant modeling error such as the mass of the mover and a viscous friction of the plant. Moreover, the performance of a current controller may cause this difference since the assumption of the Bode analysis is that the current controller is an ideal one.

Fig. 11 shows the relative position control performance of the conventional method with the disturbance observer. The position control performance was much enhanced. The position tracked the reference, and the maximum position error was reduced to about 2 μm. As aforementioned, this improved result might be obtained from the effect of the disturbance observer which could compensate internal model errors.

Fig. 12 shows the relative position control performance of the proposed method. The position control performance was an improvement upon the former cases, with the maximum position error of about 1.4 μm. The performance difference between the Bode analysis and the experimental result may be resulted from the reason mentioned earlier. By the proposed method, the relative position control performance was enhanced by up to 30% compared with that of the conventional method with the disturbance observer in Fig. 11.

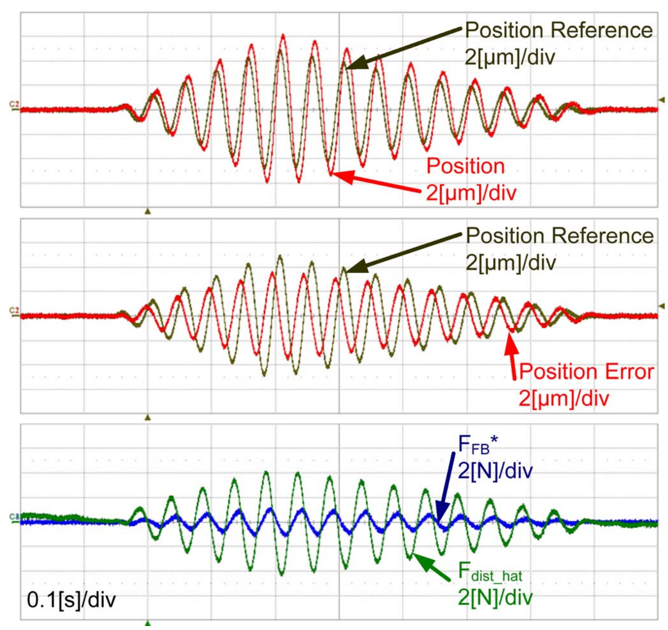


Fig. 14. Relative position control performance of the conventional method with the disturbance observer; the frequency of the position reference is 20 Hz.

Figs. 13–15 show the experimental results under the same condition as that in Figs. 10–12, respectively, except for the different frequency and magnitude of the relative position command. In this test, the frequency was 20 Hz rather than 10 Hz such as that of Figs. 10–12, and the maximum displacement of the relative position reference was 5 μm, in contrast to 10 μm of the previous test.

Fig. 13 shows the relative position control performance of the conventional method without the disturbance observer. The position control performance was still unacceptable.

Fig. 14 shows the relative position control performance of the conventional method with the disturbance observer. In contrast

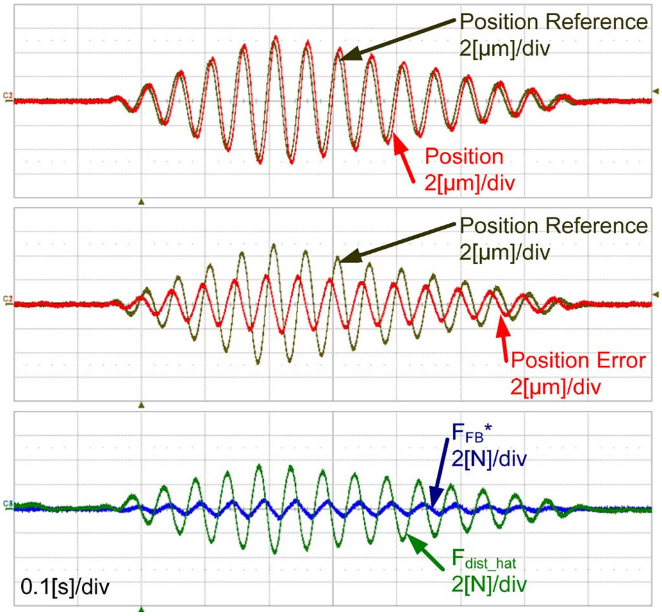


Fig. 15. Relative position control performance of the proposed method; the frequency of the position reference is 20 Hz.

to the previous case shown in Fig. 11, the position control performance was not enhanced. The maximum position error was about 3.6 μm. This is because the bandwidth of the position controller was set to about 9 Hz, and that was insufficient to track the position command whose frequency was 20 Hz in the conventional control structure. In the conventional system, although the disturbance observer can compensate internal model errors, it cannot improve the performance of the relative position control.

Fig. 15 shows the relative position control performance of the proposed method. The position control performance was improved. The maximum position error was about 2.4 μm. Even though the bandwidth of the position controller is not large enough, the performance can be enhanced significantly since the position control is additionally performed by the disturbance observer besides the position controller itself. By the proposed method, the relative position control performance was enhanced by up to 30% compared with that of the case shown in Fig. 14.

Figs. 11, 12, 14, and 15 show the force from the position controller and the force compensated by the disturbance observer at each case. The magnitudes of the forces from the disturbance observer were greater than those from the position controller, and the shapes of the forces from the disturbance observer were less distorted than those from the position controller. Therefore, it could be deduced that the force from the disturbance observer is more dominant and more accurate than that from the position controller. This is because the bandwidth of the disturbance observer is larger than that of the position controller. As a result, the application of a disturbance observer for a relative position control system is significantly effective to enhance the performance.

From the aforementioned experimental results, it can be concluded that the performance of the relative position control can be much improved using the proposed method. In particular, when the bandwidths of the relative position control system are restricted due to mechanical reasons, this improvement can make a significant difference on the performance of the whole system.

V. Conclusion

This paper has presented an application of a disturbance observer for a relative position control system. The proposed method provides a different viewpoint for the relative motion control, in which the position command has been considered as the result of a disturbance. Therefore, the relative position control has been performed by the disturbance observer as well as by the position controller. As a result, with the simple modification, the relative position control performance was enhanced by up to about 30% compared with that of the conventional method with the disturbance observer.

The feasibility of the proposed method has been verified by the experimental results using a high-precision linear motion stage as well as by the analysis based on Bode plots.

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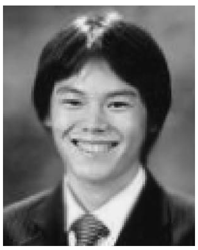

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