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Paper:

Positioning of an X-Y Stage Using the Horizontal Acceleration Signal of the Base Plate

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In this paper, we formulate the effectiveness of the horizontal acceleration feedback for an X-Y stage installed in the reduction stepper by using the 2-degree-of-freedom mechanical model and show that its feedback is equivalent to the zeroing of disturbance. Furthermore, we propose the disturbance observer instead of acceleration feedback, and its superior performance is illustrated by using the numerical simulation.

Keywords: Reduction stepper, X-Y stage, Acceleration feedback, Zeroing, Disturbance observer

1. Introduction

In order to realize higher throughput in submicron lithography, higher velocity X-Y stage must be developed. An X-Y stage using an air bearing guide may be expected to have high speed positioning due to the frictionless structure and pure linearility. However, this stage requires long settling time due to the lack of friction. To realize high velocity positioning, the effectiveness of the horizontal acceleration feedback of the base plate was experimentally demonstrated in the literature.[1] However, an explanation for the theoretical background of its feedback was insufficient.

In this paper, we formulate the effectiveness of acceleration feedback for the first time by using 2-d.o.f. mechanical model and show that an acceleration feedback is equivalent to the zeroing of the disturbance. In order to improve the positioning time, we propose the disturbance observer instead of acceleration feedback, and its superior performance is illustrated by using the numerical simulation. In addition, some experimental results are shown.

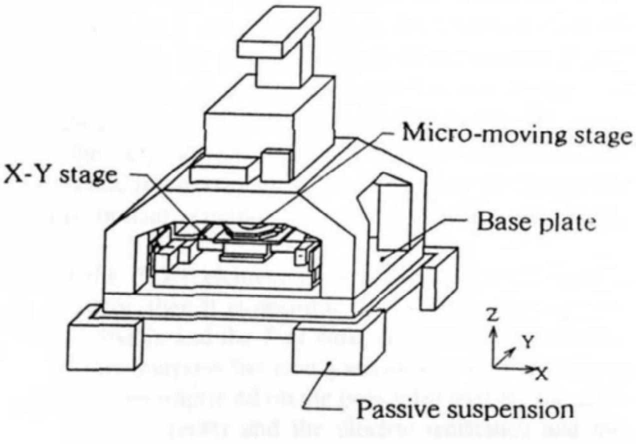


Fig. 1. Reduction stepper system.

2. Modeling of X-Y Stage and Zeroing by Horizontal Acceleration Feedback

Figure 1 shows the reduction stepper system. In this figure, the micro-moving stage for the fine positioning of IC wafer is mounted on the X-Y stage, and this X-Y stage for coarse positioning is mounted on the passive suspension with the goal of attenuating the floor vibration. Recently, an active suspension using the voice coil motor or air spring is introduced to the stepper system. However, we must pay attention to the reaction force due to the step motion of X-Y stage. In the case of leaving the reaction force freely, the settling time of the X-Y stage becomes very long due to the vibration of the base plate.

In this section, we explain the modeling of the X-Y stage system containing the base plate dynamics and illustrate the relationship between the acceleration feedback of the base plate and the zeroing. At first, let us consider the mathematical model of the X-Y stage shown in **Fig.2**. The equation of motion of this model is written by:

$$m_1\ddot{x}_1 + b_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = f \quad \ldots\ldots \quad (1a)$$

$$m_2\ddot{x}_2 + b_2\dot{x}_2 + k_2x_2 + b_1(\dot{x}_2 - \dot{x}_1) + k_1(x_2 - x_1) = -f + f_{ext} \quad \ldots\ldots\ldots \quad (1b)$$

where x_1 is the displacement of the stage; x_2 is the displacement of the base plate; m_1 is the stage mass; m_2 is the base plate mass; k_1 and k_2 are the spring constants of the stage and the base plate, respectively; b_1 and b_2 are the viscous damping of the stage and base plate, respectively; f is the control force; and f_{ext} is the disturbance force.

Next, let us consider the closed position loop shown in

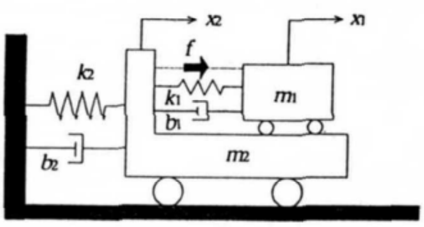


Fig. 2. X-Y stage model.

Fig.3. Because the position sensor such as a laser interferometer, which has a detecting resolution of 5nm, is generally installed on the base plate, the position signal for the control loop becomes (x_1-x_2). **Figure 4** now shows the frequency shaping of response $(x_1-x_2)/x_0$ both with and without acceleration feedback. It is interesting to note from this figure that there is pole/zero cancellation related to the vibration mode of base plate in the measurement of the frequency response from the command x_0 to the position signal (x_1-x_2). Hence, as long as the only frequency response from the command x_0 is considered, we cannot recognize essential qualities of the acceleration feedback. Not only the response from the command x_0 but also the response from the disturbance force f_{ext} must be considered.

Then, we derive the relationship between the position signal (x_1-x_2), the command x_0, and the disturbane force f_{ext} as follows.

(A) In the case of A=0 (without acceleration feedback).

$$(x_1-x_2)=\frac{K_pK_tK_i[(m_1+m_2)s^2+b_2s+k_2]\,(F_vs^2+F_xs+F_i)x_0}{sg''(s)+K_pK_tK_i[(m_1+m_2)s^2+b_2s+k_2](F_vs^2+F_xs+F_i)} \;*$$

$$*\;\frac{-m_1s^3\,(1+sT_d)f_{ext}}{} \quad \ldots\ldots\ldots\ldots (2)$$

(B) In the case of A≠0 (with acceleration feedback).

$$(x_1-x_2)=\frac{K_PK_tK_i(1+sT_a)[(m_1+m_2)s^2+b_2s+k_2](F_vs^2+F_xs+F_i)x_0}{sg'(s)+K_PK_tK_i(1+sT_a)[(m_1+m_2)s^2+b_2s+k_2](F_vs^2+F_xs+F_i)} \;*$$

$$*\;\frac{+\,s[K_tK_iAs^2-m_1s^2(1+sT_a)\,(1+sT_d)]f_{ext}}{}$$

$$\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

$$g''(s)=(1+sT_a)g(s) \quad \ldots\ldots\ldots\ldots (4a)$$

$$g'(s)=(1+sT_a)(1+sT_d)g(s)+AK_tK_im_1s^4 \quad . \ (4b)$$

$$g(s)=m_1m_2s^4+[m_1(b_1+b_2)+m_2b_1]s^3+[m_1(k_1+k_2)$$

$$+\,b_1b_2+m_2k_1]s^2+[b_1k_2+b_2k_1]s+k_1k_2$$

$$\ldots\ldots\ldots\ldots\ldots\ldots (4c)$$

where K_p is the position gain, K_t is the force factor, K_i is the power driver gain, A is the acceleration feedback gain, T_d is the time constant of the power driver, T_a is the time constant of the acceleration feedback, and F_x, F_i, and F_v denote the PID parameters, respectively.

From the second term of (3), we notice that the response from F_{ext} to (x_1-x_2) can be effectively suppressed if an acceleration feedback gain A is optimally selected as follows:

$$A_{opt}=m_1/K_tK_i \quad \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

Namely, the response from f_{ext} to (x_1-x_2) in the case (B) can be suppressed more than those in case (A). In addition, it is numerically clear that both the dominant characteristic poles of (2) and (3) are almost same. Hence, the Separation theorem approximately holds between the PID position control loop only and the PID loop containing the disturbance rejection loop with the acceleration feedback. Then, we can recognize that this effect is equivalent to the zeroing for the disturbance force f_{ext}. In the literature,[1] the presence of A_{opt} as described (5) is not considered at all. The results mentioned above are summarized as follows:

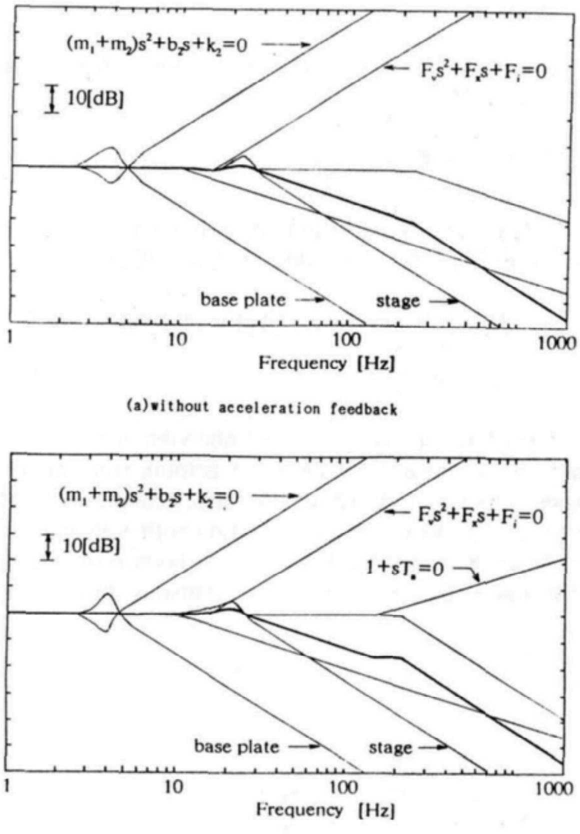


Fig. 4. Frequency shaping of response $(x_1-x_2)/x_0$.

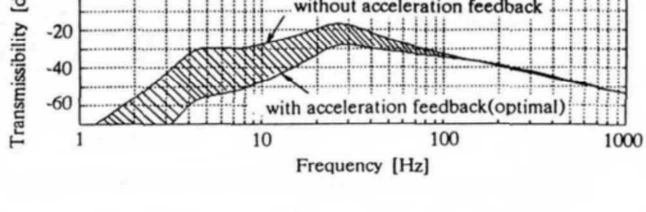


Fig. 5. Transmissibility with and without acceleration feedback.

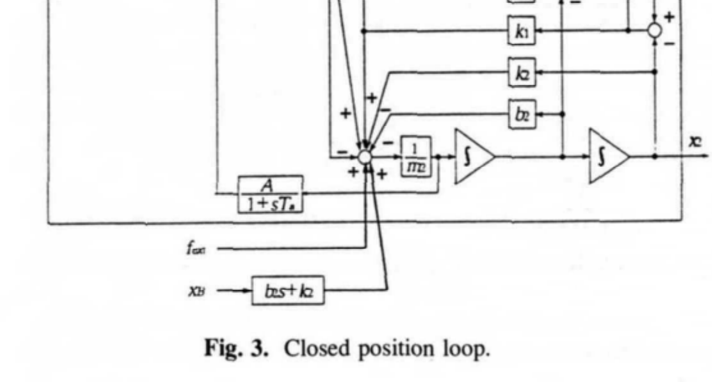


Fig. 3. Closed position loop.

(i) The disturbance rejection becomes optimal by selecting an acceleration feedback gain A_{opt} as (5).

(ii) It is impossible to realize the complete zeroing because the time constants T_a and T_d are finite values. If both T_a and T_d become zero, then the zeroing is perfectly accomplished.

(iii) Thanks to the realization of an approximate Separation theorem, the procedure of parameter tuning in the production field is quite plain to us. At first, the PID parameters for the position loop and the PI parameters for the velocity loop are adequately selected using some means in order to satisfy the specification of control system. Next, the acceleration feedback is appended to the position loop with its PID and PI parameters holding.

(iv) Using the horizontal acceleration signal of the base plate, the response from the floor vibration x_B to (x_1-x_2) can also be suppressed because this equation of motion is rewritten in the following form using (1b).

$$m_2\ddot{x}_2 + b_2(\dot{x}_2 - \dot{x}_B) + k_2(x_2 - x_B) + b_1(\dot{x}_2 - \dot{x}_1) + k_1(x_2 - x_1) = -f + f_{ext} \quad \cdots\cdots (6)$$

Substituting (7) described below into (3), the transfer function form x_B to (x_1-x_2) is obtained.

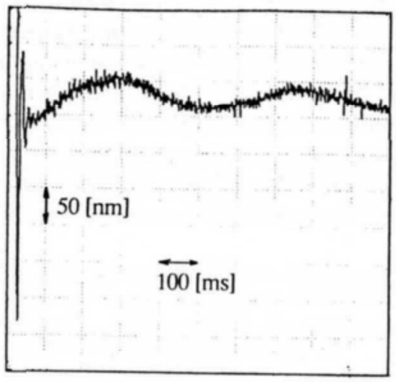
$$f_{ext} \rightarrow (b_2 s + k_2)\, x_B \quad \cdots\cdots (7)$$

In this case, the numerator appearing in the transfer function from f_{ext} to (x_1-x_2) is described as follows

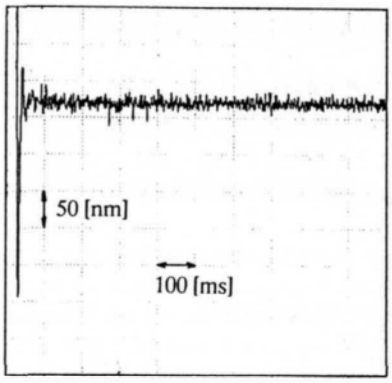
$$s[K_t K_i A s^2 - m_1 s^2 (1 + sT_a)(1 + sT_d)]\,(b_2 s + k_2) \quad \cdots\cdots (8)$$

Then, the zeroing from the floor vibration x_B is essentially the very same effect as the zeroing from the disturbance force f_{ext}. In **Fig.5**, the frequency characteristics from x_B to (x_1-x_2) show the transmissibility both with and without acceleration feedback. Clearly, it is soon proved that the acceleration feedback makes the transmissibility better at the low-frequency.

In order to verify the performance of closed loop with acceleration feedback, we finally show the positioning waveform of X-Y stage in **Fig.6**. The difference between cases (A) and (B) is apparent from this figure. **Figure 6(a)** indicates the error signal of case (A). Obviously, it takes a



(a)Without acceleration feedback



(b)With acceleration feedback

Fig. 6. Experimental step response.

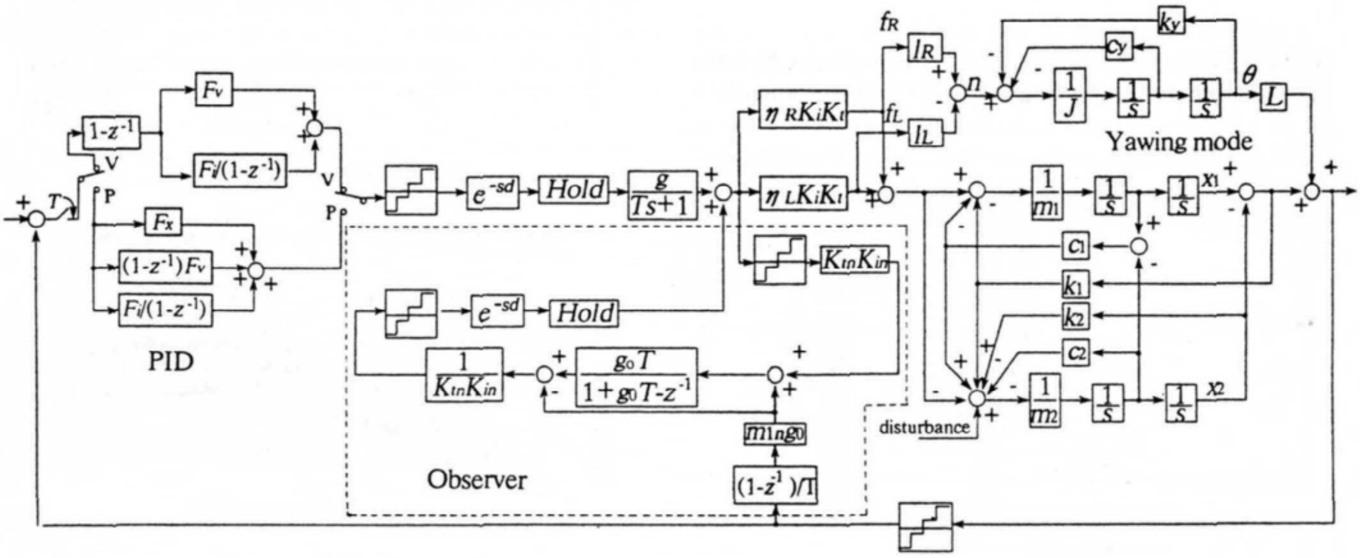


Fig. 7. Block diagram for simulation study.

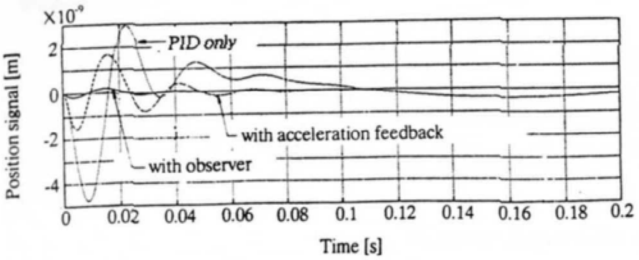


Fig. 8. Calculated step response.

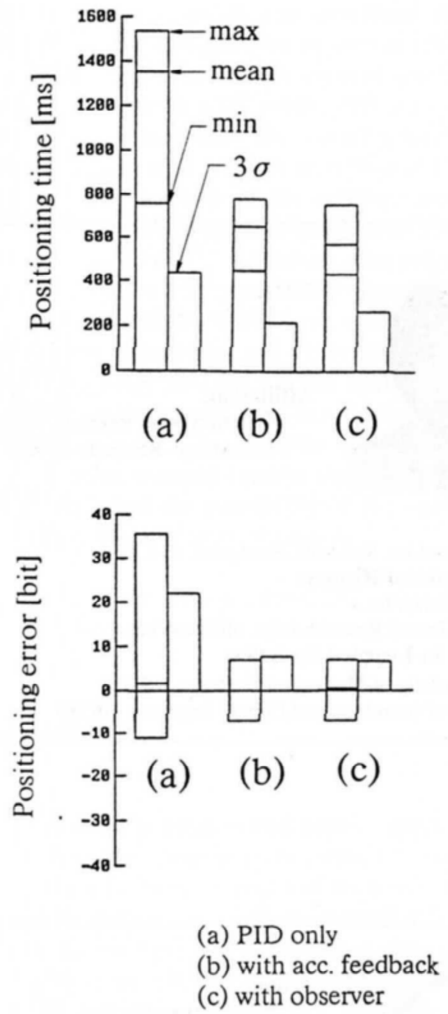


Fig. 9. Statistical positioning data in the case of Step & Repeat motion

long time to reach the desired position due to the vibration with the natural frequency of base plate. However, its resonance is effectively suppressed in case (B), and good convergence can be obtained in settling time.

3. Disturbance Observer

Because the horizontal acceleration feedback has the effect of disturbance rejection as shown in Fig.6, the settling time becomes shorter in comparison with the control loop without acceleration feedback. However, the previously published acceleration feedback method has three problems in the production field. These problems are summarized as follow:

(i) The servo type accelerometers with high sensitivity at low frequency are very expensive.

(ii) They are easily destroyed applying the mechanical shock.

(iii) Those frequency band are limited. Hence the effect of the disturbance rejection is also limited due to the finite time constants T_a.

In order to overcome the limitation mentioned above, we propose the disturbance observer[2] instated of acceleration feedback. **Figure 7** shows the block diagram of the closed loop containing the observer for the simulation study. The control system includes coarse and fine parts, and these are automatically selected according to the detected positioning error. When the positioning error becomes less than 1.0μm, the controller switches PI operation of velocity control to PID operation of positioning control. In this figure, the dashed line denotes the disturbance observer proposed by Prof. K.Ohnishi of Keio University et al., and the block diagram of yawing motion of X-Y stage is newly added to investigate the robustness and the stability for the spurious mode.

Figure 8 shows an example of calculated error responses for the step motion. The disturbance observer installed in the closed loop behaves well in comparison with the response of the system with acceleration feedback of base plate under the condition that we can set the frequency band of the observer as high as possible.

Figure 9 shows the statistical positioning data, i.e., positioning time and positioning error. It is evident from this figure that the disturbance observer installed in the closed loop has almost the same positioning performance as the closed loop with acceleration feedback. Although the experimental results do not satisfy our expectation for the disturbance observer, we can conclude in any case that the disturbance observer is equivalently exchangeable for acceleration feedback.

4. Conclusion

This paper has theoretically presented the effect of acceleration feedback in the reduction stepper system, and we have demonstrated that its feedback is equivalent to the zeroing. In order to further improve the closed loop performance, we propose the use of the disturbance observer. Within the limits of the numerical simulation study, the superior performance of the disturbance observer is illustrated. On the other hand, experimental data of the proposed control scheme only show the comparable positioning to the PID loop with acceleration feedback. From now on, we will investigate the realization of superior positioning using the disturbance observer.

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