Interaction Mode Control for the Machine Tool Table with Three Linear Servo Motors*

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Abstract

The linear motor driving mechanism has been adopted as a positioning mechanism of machine tools to realize high speed and precise positioning because of friction-free and no backlash. However, when two or more linear motors are used, minute differences between the thrust forces of each motor causes the generation of micro vibration and the change of velocity of each linear motor generated by yawing around the center of gravity. This study proposes interaction-mode-control method for the machine tool table driven by three linear motors to control the three modes defined as the position of the center of gravity, bending and yawing independently. The interaction-mode-control method is a new technique for applying interaction modes defined according to the workspace to the real control system through modal conversion. From the results of simulation and experiments, it is confirmed that the three modes are controlled independently and, therefore, bending and yawing can be suppressed using interaction-mode-control. Moreover, to combine the interaction-mode-control method with disturbance observer and feedforward control, this control system has high robustness and precise positioning can be realized.

Key words: Positioning, Machine Tools, Posture Control, Interaction-Mode-Control, Feedforward Control, Disturbance Observer

1. Introduction

In recent years, high driving speed and precise positioning is demanded in machine tools; especially, the demand of positioning accuracy is very high in order to produce products with a high accuracy and high efficiency [1-4]. The ball screw drive, an established mechanism of the feed drive, is usually applied to the machine tool table. With this drive, friction and backlash generated at the screw nut leads to a decrease in the machining accuracy [1,5]. Therefore, because it does not generate friction and backlash, the linear motor driving mechanism has been adopted as a positioning mechanism for machine tools instead the ball screw drive to realize high speed and precise positioning [6]. When the machine tool table is driven at high speed, two or more linear motors are installed in parallel for single-axis driving to obtain a high thrust force. When more linear motors are used, however, a posture change of the table occurs because the thrust force of each linear motor is not the same. In addition, this difference of the thrust force generates micro vibration and changes the velocity of each linear motor caused by yawing around the center of gravity (COG). In this study, an interaction-mode-control method is proposed for a table drive system with three linear motors that control the position of the center of gravity, the bending

and the yawing independently.

2. Driving mechanism of twin and triple linear motor drive table

In a conventional table driving system with linear motors, only one linear encoder is installed on a single axis, even if two or more linear motors are applied. Therefore, it is impossible to control the effect of bending and yawing on the posture of the table. If the position of each linear motor can be measured by an encoder installed in each motor, it is possible to control the posture of the table. For example, by using two linear motors and two linear encoders, yawing generated around the COG can be controlled, as illustrated in Fig. 1. Moreover, when the table is driven by three linear motors with three linear encoders, as shown in Fig. 2, it becomes possible to control not only the yawing but also the bending of table so that the micro vibration induced by bending could be suppressed if the sampling frequency of control system is higher than its oscillation frequency.

Fig.1 Twin motor drive

Fig.2 Triple motor drive

3. Structure of the Table System

In this study, a table system driven by three linear motors, each on a single axis, is developed. Figure 3 shows the structure and photograph of the table driven with three linear motors. The weight of the table is 9.8 kg, and it is supported by two linear guides of an antifriction bearing set at the front side and back side. Figure 4 shows a schematic diagram of the table system. A linear encoder is set for each linear motor so that each position and velocity of each linear motor can be measured through the encoder counter board. Then the measured position and velocity of the three motors are fed back to the controller and the command values are calculated. The command signal is changed from a digital signal to an analog signal through the D/A Board.

(a) Schematic diagram of driving table

(b) The appearance of driving table

Fig.3 Structure of driving table with three linear servo motors

Fig.4 Schematic diagram of table system

4. Principle of interaction-mode-control

The interaction-mode-control method is a new technique for applying interaction modes defined according to the workspace to the actual control system. Each position of a defined interaction mode is directly and independently controlled by this proposed method, which regulates three modes, that is, the center of gravity (COG), the bending and the yawing, using three linear motors. Figure 5 shows the position coordinates of each motor, and the position of the COG, bending and yawing, which are described as follows.

Fig.5 Position coordinates of each encoder

$$g = \frac{x_1 + x_2 + x_3}{3} \quad (1)$$

$$d = \frac{x_1 + x_3}{2} - x_2 \quad (2)$$

$$\theta = \frac{x_1 - x_3}{L} \quad (3)$$

Where g is the position of the COG, d is the amount of bending, θ is the amount of yawing, L is the total length between the three motors, and x_1, x_2, x_3 are the positions of the three motors. In this study, the position of COG mode x_g , bending modal position x_d and yawing modal position x_θ are defined as $x_1 + x_2 + x_3$, $x_1 + 2x_2 + x_3$ and $x_1 - x_3$ respectively. Therefore, relation between each mode and the position of the three motors is represented as follows.

$$\begin{pmatrix} x_g \\ x_d \\ x_\theta \end{pmatrix} = Q \begin{pmatrix} x_1 \\ x_2 \\ x_3 \end{pmatrix} \quad Q = \begin{pmatrix} 1 & 1 & 1 \\ 1 & -2 & 1 \\ 1 & 0 & -1 \end{pmatrix} \quad (4)$$

Where Q is the quarry matrix which transfers real positions into modal ones [8]. The command signal is sent to an actual linear motor by converting the position of each mode to each actual position of the linear motor, as given in Eq.(5).

$$\begin{pmatrix} x_1 \\ x_2 \\ x_3 \end{pmatrix} = Q^{-1} \begin{pmatrix} x_g \\ x_d \\ x_\theta \end{pmatrix} \quad Q^{-1} = \frac{1}{6} \begin{pmatrix} 2 & 1 & 3 \\ 2 & -2 & 0 \\ 2 & 1 & -3 \end{pmatrix} \quad (5)$$

Table 1 Parameters for interaction mode control simulation

Interaction mode control			
	Sampling time	1.0	ms
Command value of COG mode			
Amplitude		10.0	mm
Frequency		1.0	Hz
Command value of bending mode			
Amplitude		5.0	mm
Command value of yawing mode			
Amplitude		-2.5	mm
K_{p1}	Position control gain of COG mode	25.0	
K_{v1}	Velocity control gain of COG mode	100.0	
K_{p2}	Position control gain of bending mode	25.0	
K_{v2}	Velocity control gain of bending mode	100.0	
K_{p3}	Position control gain of yawing mode	5.0, 25.0, 50.0	
K_{v3}	Velocity control gain of yawing mode	20.0, 100.0, 200.0

Fig.6 Simulation result of interaction-mode-control

By direct control of the interaction modal positions using this method, it becomes possible to control each mode independently because the three modes are orthogonal each other. To evaluate this method, the interaction-mode-control is simulated as shown in Fig. 6. The parameters used in this simulation are listed in Table 1. From the result, it is confirmed that the position of each mode can be controlled. In addition, the response of yawing can be improved without interaction with the other two modes when the controller gain of yawing modal position only increases.

5. Application of feedforward control method and disturbance observer

In order to improve the responsivity of the position and compensate the disturbance error remarkably, feedforward control and disturbance observer [7-10] are applied in the proposed control system. Figure 7(a) shows a block diagram of the feedback controller. The transfer function G from the command position to the response position in the feedback system is described as

$$G(s) = \frac{x^{res}}{x^{cmd}} = \frac{K_p K_v}{s^2 + K_v s + K_p K_v} \quad (6)$$

where K_p is the position control gain, K_v is the velocity control gain, x^{res} is response position, x^{cmd} is command position, and the parameter s is in general complex.

This system generates a second-order time lag of response, as shown in the simulation result of Fig. 8. To improve the time lag of response, a feedforward control, as illustrated in

(a) Feedback controller

(b) Feedforward controller

Fig.7 Block diagram of feedback controller and feedforward controller

Fig.8 Feedback simulation

Fig.9 Feedforward simulation

(a) Structure of disturbance observer

(b) Equivalent block diagram of (a)

Fig.10 Block diagram of disturbance observer

Fig. 7(b), is applied. The feedforward system can be realized by giving the velocity and acceleration in the input command.

By applying the feedforward control, the transfer function between input and output becomes,

$$G(s) = \frac{x^{res}}{x^{cmd}} = \frac{s^2 + K_v s + K_p K_v}{s^2 + K_v s + K_p K_v} = 1 \quad (7)$$

Equation(7) indicates that the response value corresponds with the command value at all times without delay. As shown in Fig. 9, the responsivity improves remarkably.

In addition, a disturbance observer, represented in Fig. 10(a), was designed. This system is converted into equivalent block diagram as shown in Fig. 10(b). By using the disturbance observer, it becomes possible to estimate the disturbance error with high accuracy and to compensate for it. Figure 11 shows the effect of the disturbance observer by a simulation. When a constant disturbance is applied to this system after 1.0 s, the change of

position caused by the disturbance is immediately compensated by the disturbance observer. Moreover, by setting a higher gain of the disturbance observer, the effect of suppression against the position error is improved without interaction of the main control system. In order to obtain these effects, the proposed interaction-mode-control system is combined with the feedforward and the disturbance observer, as shown in Fig. 12.

Fig. 11: Effect of disturbance observer

Fig.12 Block diagram of the interaction-mode-control system

6. Performance evaluation of interaction-mode-control

6.1 Step response test

The proposed method regulates the positions of bending and yawing mode to attain bending and yawing suppression, since they correspond to the actual bending and yawing angle. In order to evaluate the effect of bending and yawing suppression, a step response test was carried out with both the interaction-mode-control method and the real motor control method to directly control the positions of the three linear motors. The control parameters are listed in Table 2. The results of the step response test are shown in Fig. 13-15. Bending and yawing are evaluated by the value of the each modal positions defined as $x_1 + 2x_2 + x_3$ [μm] and $x_1 - x_3$ [μm]. No difference in the response position of the COG between the interaction-mode-control and real motor control is observed. On the contrary, the amount of bending and yawing is effectively suppressed by using the interaction-mode-control because the bending mode and yawing mode can be controlled independently. However, steady state error of the bending modal position remains and, especially, increases gradually in real motor control, because of the severe structural and mechanical constraint. In this system, only quasi-static deformation of machine tool table can be controlled because sampling frequency is lower than natural frequency of bending mode. Actually, natural frequency of bending mode which is measured by impulse hammer test is 2.4 kHz.

Therefore, it is hard to control dynamic deformation based on natural frequency of bending mode. If the sampling time which is shorter than $50\mu\text{s}$ can be realized, apparently, it will be possible to suppress the dynamic deformation.

Table 2 Experimental parameter of real motor control and interaction mode control

Common parameter			Interaction mode control		
	Sampling time	1.0 ms		Command value of COG mode	10.0 mm
g_{dis}	Cut-off frequency of disturbance observer	700 Rad/s		Command value of bending mode	0.0 mm
				Command value of yawing mode	0.0 mm
Real motor control			K_{p1}	Position control gain of COG mode	20.0
	Command value	10.0 mm	K_{v1}	Velocity control gain of COG mode	80.0
K_p	Position control gain	20.0	K_{p2}	Position control gain of bending mode	80.0
K_v	Velocity control gain	80.0	K_{v2}	Velocity control gain of bending mode	320.0
			K_{p3}	Position control gain of yawing mode	80.0
			K_{v3}	Velocity control gain of yawing mode	320.0

Fig.13 Result of COG mode in step response

Fig.14 Result of bending mode in step response

Fig.15 Result of yawing mode in step response

Table 3 Experimental parameters of interaction-mode-control

Interaction mode control		
	Command value of gravity mode	0.01 m
	Command value of bending mode	0.0 μm
	Command value of yawing mode	2.0 μm
K_p	Position control gain of yawing mode	10.0, 20.0
K_v	Velocity control gain of yawing mode	40.0, 80.0

Fig.16 Result of yawing mode

In addition, by giving the command value of the yawing mode, as shown in Table 3, it becomes possible to deliberately generate yawing. Figure 16 shows the comparison of the yawing response between the yawing mode gain Kv_3 of 40 and 80. As a result, yawing is generated according to the command of the yawing mode. Moreover, the responsivity of the yawing mode to the command is improved when the yawing mode gain is set higher.

6.2 Evaluation of independent control

In order to investigate the independency of each mode, a step response test was carried out under various experimental conditions, shown in Table 4. Figure 17 shows a comparison of each mode response for case A and case B. There is no difference of the response position in both the COG mode and bending mode because of the same gains set in the COG mode and bending mode as the experimental conditions of case A and case B. On the other hand, the suppression effect of yawing is considerably different. Similarly, Fig. 18 shows a comparison of the responses for case A and case C. It is possible to control the bending mode independently. From these results, it is clearly possible to design the responsiveness of each mode independently in the interaction-mode-control system because each mode does not interact with each other. This advantage makes the gain tuning of the controller considerably easier.

Table 4 Experimental parameters used in evaluation of independent control

Command value			
Command value of COG mode	0.01	m	
Command value of bending mode	0.0	μm	
Command value of yawing mode	0.0	μm	

Controller gain			
case	Kp_1, Kv_1	Kp_2, Kv_2	Kp_3, Kv_3
A	20, 80	80, 320	80, 320
B	20, 80	80, 320	20, 80
C	20, 80	20, 80	80, 320

Fig.17 Comparison of each response between case A and case B

Fig.18 Comparison of each response between case A and case C

6.3 Evaluation of the feedforward effect using the frequency response test

A frequency response test was carried out to compare the responsivity of the feedback control and feedforward control. Figure 19 shows the position response of the COG with respect to the sinusoidal command using feedback control and feedforward control. The frequency response characteristics of feedback control and feedforward control are shown in Fig. 20. It is apparent that responsivity is improved by applying feedforward control.

Fig.19 Comparison of responsivity between feedback control and feedforward control

Figure 20: Frequency characteristics of feedback control and feedforward control

6.4 Influence of thrust force saturation on interaction-mode-control

The thrust force is limited by a servo drive controller not to exceed 125 N. However, when the mechanical assembly of the table system is not suitable, the thrust force tends to increase more and more due to the effect of the disturbance observer, and then the thrust force becomes saturated. This causes the increase of micro vibration and yawing because of the lack of control during saturation of thrust force. Therefore, the influence of thrust force on the responses of the bending mode and yawing mode is investigated. Figure 21 shows the response with thrust force saturation, and Fig. 22 shows the response without thrust force saturation. In both figures, (a) shows the response of bending and yawing, and (b) shows the response of thrust force. These results indicate that the bending and yawing mode cannot be controlled when the thrust force is saturated. Therefore, it is necessary to prevent thrust force saturation for control of the bending and yawing modes. In order to escape this saturation from the viewpoint of mechanical solution, mechanical accuracy of the table system should be improved such as straightness for the total motion length of each motor and parallelism of three linear motors and encoders. As the other solution from the viewpoint of controller design, the stiffness of bending mode should be reduced to improve the performance of yawing suppression. For high performance realization of interaction-mode-control, it is important to assure the high assembly accuracy of machine tool table.

Figure 21: Result with thrust force saturation

Figure 22: Result without thrust force saturation

7 Conclusions

In order to control the position of the center of gravity, the bending and the yawing independently, the interaction-mode-control method was proposed using three linear motors for the driving table. From the results of simulation and experiments, it is confirmed that the three modes are controlled independently and, therefore, bending and yawing can be suppressed by using interaction-mode-control. Moreover, by combining the interaction-mode-control method with the disturbance observer and feedforward control, this control system has high robustness, and high responsivity can be realized. Finally, to realize the suppression of bending and yawing, it is necessary to consider the influence of thrust force on the responses.

References

- (1)Neou, J. et al., Investigation of the Microscopic Performance of an Aerostatic Precision Positioning Stage Driven by the Air-lubricated Friction Drive, *The Third International Conference on Leading Edge Manufacturing in 21st Century*, Vol.3, No.05-204(2005), pp1003-1008.
- (2)Mizumoto, H. et al., A Picometer Positioning System using Active Aerostatic Guideway, *The Third International Conference on Leading Edge Manufacturing in 21st Century*, Vol.3, No.05-204(2005), pp1009-1014.
- (3)Hashimoto, S. et al., Ultra-Precision Stage Control Based on Friction Model of Non-Resonant Ultrasonic Motor, *The 8th IEEE International Workshop on Advanced Motion Control*, (2004), pp. 559-564.

- (4)Maeda, J.G. et al., Control of an XY Nano-Positioning Table for a Compact Nano-Machine Tool, *The Third International Conference on Leading Edge Manufacturing in 21st Century*, Vol.3, No.05-204(2005), pp. 987-992.
- (5)Fujita, T. et al., Dynamic Characteristics and Positioning Performance of Piezoactuator-integrated Ball Screw Drive, *Proceedings of the 4th International Conference on Leading Edge Manufacturing in 21st Century*, No.07-205(2007), pp507-510.
- (6)Kaneko, S., Sato, R., Tsutsumi, M., Mathematical Model of Motor Stage with Non-linear Friction Characteristics, *Proceedings of the 4th International Conference on Leading Edge Manufacturing in 21st Century*, No.07-205(2007), pp501-506.
- (7)Ohnishi, K., Robust Motion Control by Disturbance Observer, *Journal of the Robotics Society of Japan*, Vol.11, No.4(1993), pp. 486-493.
- (8)Katsura, S. et al., Quarry of Environmental Information Based on Twin Robot System ~Environment Quarrier~, *The 8th IEEE International Workshop on Advanced Motion Control*, (2004), pp. 243-248.
- (9)Katsura, S., Ohnishi, K., Vibration Control of Multi-Mass Resonant System Based on Phase-Lead Compensator, *IEEJ Transactions on Industry Applications*, Vol.126, No.12(2006), pp1601-1607.
- (10)Shinno, H., Hashizume, H., Yoshioka, H., Sensor-Less Monitoring of Cutting Force During Ultraprecision Machining, *Annals of the CIRP*, Vol.52/1/2003(2003), pp303-306.