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Control system design of a linear motor feed drive system using virtual friction

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ABSTRACT

This paper proposes a friction compensator and a design method for control systems to improve the response characteristics of linear motor feed drive systems. The proposed friction compensator cancels the real nonlinear friction of feed drive systems by using the nonlinear friction model proposed in this study and introduces virtual linear friction to facilitate the control system design. The proposed design method enables the determination of servo gains and friction compensator parameters that lead to desirable tracking performance and disturbance rejection without many trial-and-error tuning processes. In addition, the proposed method facilitates the design of the velocity feedforward compensator by using the inverse transfer function of the velocity control loop to correct the position tracking errors for various position commands. The effectiveness of the proposed method with the friction compensator and the velocity feedforward compensator was verified in simulations and experiments using a one-axis feed drive system consisting of a rod-type linear motor and linear roller guides. The results confirmed that the proposed method enables desirable overshoot-free responses and corrects motion trajectory errors due to nonlinear friction characteristics, and the proposed velocity feedforward compensator can correct tracking errors in both constant velocity motion and circular motion.

1. Introduction

Linear motor feed drive tables are one of the machine tool direct drive applications suitable for high-speed and high-precision motion. Since the servo gain has no limit due to mechanical resonance, linear motor feed drive systems can achieve higher servo gain settings as compared to ball screw feed drive systems [1,2]. However, disturbances such as friction forces and cutting forces directly act on the motor and can deteriorate the positioning accuracy [3].

Friction forces have nonlinear characteristics, especially at velocity reversal, and cause protruding tracking errors called quadrant glitches during the circular motions in *X–Y* stages [3,4]. However, a moderate amount of friction force can stabilize servo systems and allow higher servo gain settings that lead to high-precision motion control [5]. Especially in the case of linear motor feed drive systems, the friction force provides stiffness and damping in the feed direction. Therefore, for improvement of the response characteristics of linear motor feed drive systems, it is desirable that the friction characteristics are clarified and taken into account in the control system design.

The cascade P–PI controller is widely used in machine tool servo systems. The controller consists of a proportional (P) component for position feedback and a proportional-integral (PI) component for velocity feedback. In general, the servo gains for each component are tuned by trial and error while checking the actual behavior. Although higher servo gain settings improve the tracking performance and disturbance rejection, the nonlinear friction can only be partially compensated by cascade P–PI controllers based on linear feedback control strategies [3].

As a solution, this paper proposes a friction compensator and a parameter design method for cascade P–PI position control systems using the friction compensator.

The friction compensator is characterized in that it has a nonlinear friction model and a linear friction model. The nonlinear friction model is the nonlinear function of position and velocity that cancels the real nonlinear friction forces of feed drive systems and the linear friction model is the linear function of position and velocity. In this study, the linear friction model is called virtual friction because it is introduced into feed drive systems on behalf of the real nonlinear friction forces that provide stiffness and damping in the feed direction.

The design method using the virtual friction enables the determination of the servo gains for the P–PI position controller and the friction compensator parameters that lead to desirable overshoot-free responses for machine tools to avoid uncorrectable machining errors. Moreover, the proposed method can facilitate the design of

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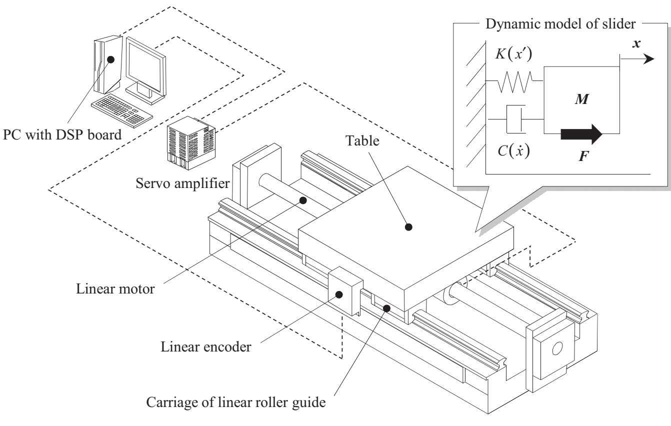

Fig. 1. Experimental setup.

the velocity feedforward compensator by using the inverse transfer function of the velocity control loop to eliminate the position tracking errors for various position commands. The feedforward compensator parameters can be uniquely determined by applying the control system design method without trial-and-error processes.

The validity of the proposed friction compensator and the control system design method is verified by simulations and experiments using a one-axis feed drive table consisting of a rod-type linear motor and linear roller guides.

This paper is organized as follows. Section 2 describes the experimental setup and nonlinear friction model used in the friction compensator. Section 3 proposes the friction compensator and the control system design method. Section 4 presents the validity of the proposed method by a comparison with the conventional cascade P–PI position control system tuned through trial and error. Section 5 discusses the velocity feedforward compensator design and its effectiveness.

2. Modeling of linear motor feed drive system

2.1. Experimental setup and mathematical model

The experimental setup is a one-axis feed drive system consisting of a rod-type linear motor, a table, a set of linear roller guides, and a linear encoder, as shown in Fig. 1. The table is supported by four carriages of linear roller guides with retainers. The feed drive system is controlled by a personal computer (PC) with a digital signal processor (DSP) board. The command values are input to the controller from the PC, and the position of the table is detected by a linear encoder having a resolution of 0.1 µm. This setup is mounted on a stone surface plate supported by a set of air servo-dampers.

Because the mass of the surface plate is much larger than that of the table, the vibration of the surface plate can be ignored. Therefore, the dynamic model is a single-degree-of-freedom system consisting of a mass, a spring, and a damper, as also shown in Fig. 1. The motion equation is

$$M\ddot{x} + C(\dot{x})\dot{x} + K(x')x' = F \tag{1}$$

where x [m] is the table position, x' [m] is the table displacement from the motion direction reversal position, M [kg] is the mass of the table with four carriages, $C(\dot{x})$ [Ns/m] is the coefficient of damping that varies with the table velocity $\dot{x}$ [m/s], $K(x')$ [N/m] is the stiffness that varies with x', and F [N] is the thrust command from the amplifier. Both $K(x')$ and $C(\dot{x})$ are mathematically modeled as the friction characteristics of the linear roller guides, which are described in detail in the following section.

Table 1
Parameters for simulation model of linear motor feed drive system.

Parameters	Unit	Values
DA converter constant, DA	V	66.6
Thrust command gain, T_{rg}	N/V	10
Time constant of the thrust command filter, T_{fil}	ms	0.10
Time constant of the servo amplifier, T_i	ms	0.35
Control period, T_c	ms	0.20
Mass of slider, M	kg	43

The block diagram of the simulation model for the linear motor feed drive system with the proposed nonlinear friction model is shown in Fig. 2. The controller is a cascade P–PI controller. In this figure, r [m] is the reference position, d [N] is the thrust disturbance other than the nonlinear friction, K_{pp} [1/s] is the proportional gain of the position loop, K_{vp} [s/m] is the proportional gain of the velocity loop, K_{vi} [1/m] is the integral gain of the velocity loop, T_c [s] is the control period, DA [V] is the DA converter constant, T_{rg} [N/V] is the thrust command gain, T_{fil} [s] is the time constant of the thrust command filter, and T_i [s] is the time constant of the servo amplifier. The controller is modeled as a discrete-time system with consideration of the quantization error of the linear encoder. The values of the parameters for the simulation model are shown in Table 1. T_i is identified by the simulation and the experimental results of the velocity step response, and the other parameters are determined from the design values.

In the experiments, the friction force is equivalent to the thrust command for the amplifier because the inertia force is negligible [6,7]. Therefore, in this paper, thrust commands recorded in the analog monitor of the servo amplifier are displayed as friction forces.

2.2. Modeling of nonlinear friction characteristics

It is well known that the relationship between the friction force and the displacement of linear roller guides exhibits a hysteresis loop under very small displacements of a few hundred micrometers. This represents the nonlinear spring characteristic and influences the dynamic behavior of the feed drive system, especially in the microscopic displacement region [4,6–12]. The friction force of the linear roller guides is also dependent on the velocity of the table. In this study, the nonlinear friction characteristics of the linear roller guides are mathematically modeled with consideration of both the nonlinear spring characteristic and the velocity dependency of the friction force [12].

The nonlinear spring characteristic is modeled by Eq. (2), which was previously proposed for linear ball guides [7].

$$f(x') = K(x')x' = \left(\frac{K_1}{\left|x'/x_1\right|^a + 1} + \frac{K_2}{\left|x'/x_2\right|^b + 1} + \frac{K_3}{\left|x'/x_3\right|^c + 1} \right) x' \tag{2}$$

where $K(x')$ is equivalent to the stiffness that varies with the table displacement from the motion direction reversal position in Eq. (1), and x' is calculated by using the integrator that is reset depending on the sign of the velocity, as shown in Fig. 2.

To identify the nine parameters (x_1, x_2, x_3, K_1, K_2, K_3, a, b, and c) in Eq. (2), the hysteresis loops described by the displacement and the friction force were measured as simple harmonic motion with very small amplitude and very low frequency so that the velocity dependence of the friction force could be ignored. The hysteresis loops described by the displacement and friction force are shown in Fig. 3(a). The loops were measured at an amplitude of 50 µm and a frequency of 0.05 Hz. The friction characteristic was changed by using two kinds of grease (A and B) filled into the linear roller guide. The identification results of the $K(x')$ parameters are shown