Disturbance Observer and Error Model-based Control of Ball Screw Drives

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⟨Abstract⟩

Ball screw drives are widely used in industry, and many studies have been devoted on precise, fast and robust control of ball screw drives. In this study, a novel position control algorithm for ball screw drives is proposed, which consist of a PD controller, a friction feedforward and a disturbance observer. The dynamics and the position error of such controller are analyzed to establish an error model, which can be used to predict the resulting position error of the given desired trajectory. Using the proposed error model, the desired trajectory can be modified so that the predicted position error can be compensated in a feedforward manner. The proposed algorithm does not require the model of the system for the error prediction, and thus can be easily applied to conventional control systems. The performance of the system is verified through simulations and experiments.

Keywords : Ball screw drivers, Disturbance observer, Error model, Position control, Proportional-differential control

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1. Introduction

Ball screw drives are widely used in industry automation, such as Cartesian robots or computerized numerical control (CNC) machines. In order to improve the performance of these automation equipment, a control algorithm which allows fast, precise and robust control of ball screw drives is highly desired. Proportional-integral-differential (PID) controllers have been widely used to control ball screw drives. However, the position tracking performance of a PID controller is limited as often, ball screw drives are subject to external disturbances or nonlinear effects, such as friction.

Friction is a highly nonlinear phenomenon which is related to the velocity of the system. Often, friction is compensated by a model-based feedforward controller, and many friction models have been proposed, such as Dahl [1], LuGre [2] or generalized Maxwell-slip (GMS) [3]. It has been shown that using these models and a feedforward controller, friction can be effectively compensated [4-7]. Based on friction models, many research efforts have been made to achieve fast, precise and robust control of ball screw drives.

Several studies aimed to completely replace PID controller. In order to achieve high control bandwidth, adaptive sliding mode controller is employed in [4]. A notch filter and active vibration cancellation are further employed to suppress the torsional vibration. Active disturbance rejection control (ADRC) is another controller aimed to replace PID controller [8]. It address several drawbacks of PID controller, such as oversimplification and noise degradation, and has been successfully used to control a ball screw drive [6].

On the other hand, many studies aimed to improve the performance PID controller by combining it with additional controllers. Many studies point out that external disturbances and model uncertainties are the major factors that degrade position tracking performance of ball screw drives [7, 9-12]. One solution would be feedforward control. In [9], iterative learning control is employed to update the parameters for feedforward controller, which is used to compensate model uncertainties and external disturbances. Another solution to model uncertainties and external disturbances is a disturbance observer (DOB) [5, 7, 10-12]. A DOB outputs the difference between the nominal model and the actual plant, and it can be used to improve the robustness of a controller [7]. A cascade P-PI controller is used with an inverse model based disturbance observer, a friction feedforward controller, an inverse model feedforward controller and a repetitive controller, and it is shown that the proposed controller can effectively suppress the effect of the injected synthesized cutting force [5]. Similar control structure has been introduced in [7], which consists of a PI controller, a disturbance observer, and a friction feedforward controller. Furthermore, the disturbance observer is further used to achieve automatic tuning of model parameters, such as effective inertia and viscous coefficient.

In this study, a novel control algorithm for fast, accurate and robust control is proposed. The proposed algorithm is based on a PD controller with a DOB and a friction feedforward controller. To improve the position tracking performance of the controller, the dynamics of the system has been analyzed, and an error model is constructed to predict the position error of the given desired trajectory. The developed error model can be used to modify the desired trajectory to take account of the predicted error. Usually, error prediction is model-based and thus it requires an accurate model of the system. However, through the use of the disturbance observer and friction feedforward controller, it is shown that accurate prediction of position error can be achieved without the need of the system model. Previous studies have also shown that error model is an effective way to improve the position tracking performance [6]. The performance of the proposed error model is verified through simulations. Furthermore, the proposed algorithm is implemented on a 1 DOF linear stage to show the position tracking performance of the proposed algorithm compare to a PD controller.

The remainder of the paper is organized as follows: Section 2 provides a brief introduction on an inverse model disturbance observer. The proposed error model is introduced in section 3, and its performance is verified through simulations in section 4 and experiments in section 5. Section 6 concludes the paper.

2. Disturbance Observer

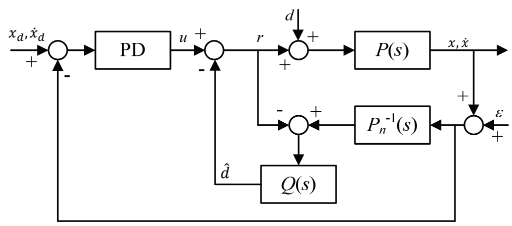

Fig. 1 Structure of inverse model based disturbance observer with PD control

Figure 1 shows a PD controller with an inverse model-based disturbance observer [10], where x, $\dot{x}$, x_d, $\dot{x}_d$, u, r, d, ε, $\hat{d}$, s, $Q(s)$, $P(s)$ and $P_n(s)$ are system position, system velocity, desired position, desired velocity, PD controller output, system input, unknown external disturbance, sensor noise, DOB output, Laplace variable, Q-filter, plant and nominal plant, respectively. Usually, a low pass filter is used as a Q-filter, and its roles are to 1) ensure that the system is proper and 2) suppress the sensor noise. In this study, a second order Butterworth filter is used as a Q-filter and its transfer function is as follows:

$$Q(s) = \frac{\omega_n^2}{s^2+\sqrt{2}\omega_n s+\omega_n^2} \tag{1}$$

where ω_n is the cutoff frequency of the filter. From Fig. 1, it can be shown that the transfer functions between the input u and the output x, and the disturbance d and the output $\dot{x}$ can be derived as [10]:

$$G_{uy}(s) = \frac{P(s)P_n(s)}{P_n(s)+(P(s)-P_n(s))Q(s)} \tag{2}$$

and

$$G_{dy}(s) = \frac{P(s)P_n(s)(1-Q(s))}{P_n(s)+(P(s)-P_n(s))Q(s)} \tag{3}$$

From Eq. (2) and (3), it is important to note that if $Q(s) \approx 1$, $G_{uy}(s) \approx P_n(s)$ and $G_{dy}(s) \approx 0$. This implies that if the frequency of the disturbance is lower than the cutoff frequency of the DOB, the plant $P(s)$ would behave like the nominal plant $P_n(s)$ and the effect of the disturbance is compensated.

3. Error Model-based Feedforward Controller

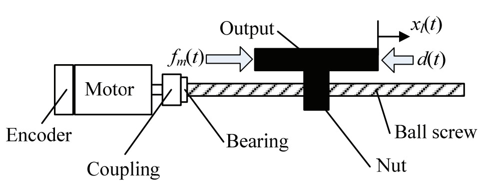

Fig. 2 Structure of ball screw drive

Typical structure of a linear stage using a ball screw is shown in Fig. 2, which consists of a motor, a coupling, a bearing, a nut, a ball screw and an output. The goal of the controller is to control the position of the output $x(t)$, and the control input is the equivalent force $f_m(t)$ applied by the motor through a coupling, a bearing and a ball screw. External disturbance $d(t)$ is also acting on the system. In overall, the system can be modeled as a two inputs (motor force $f_m(t)$ and disturbance $d(t)$) and two outputs (motor position $x_m(t)$ and output position $x(t)$) system. The model of the system can be expressed as [6]:

$$\begin{bmatrix} X_M(s) \\ X_L(s) \end{bmatrix} = \begin{bmatrix} G_1(s) & G_{DM}(s) \\ G_2(s) & G_{DL}(s) \end{bmatrix} \begin{bmatrix} F_M(s) \\ D(s) \end{bmatrix} \tag{4}$$

where $G_{DM}(s)$ and $G_{DL}(s)$ are the transfer functions between motor and load position to disturbance, and $G_1(s)$ and $G_2(s)$ are defined as:

$$G_i(s) = \frac{1}{ms^2+bs} + \sum_{k=1}^{n} \frac{\alpha_{ik}s+\beta_{ik}}{s^2+2\zeta\omega_{nk}s+\omega_{nk}^2}, i = 1, 2 \tag{5}$$

$m = m_m + m_l$ is the total equivalent mass (m_m : motor mass and m_l : output mass), b is the viscous coefficient, ω_{nk} is the natural frequency, ζ is the damping ratio, α_{ik} and β_{ik} are residual modes and n is the order of modes. A ball screw drive has high stiffness and transmission accuracy [13], and thus, it can be simplified as a rigid body model (n = 0) [6, 14]. Thus, the transfer functions can be simplified as:

$$G_i(s) = \frac{1}{ms^2+bs}, i = 1, 2 \tag{6}$$

and the dynamics of a 1 DOF linear stage can be expressed as:

$$m\ddot{x} + b\dot{x} + f_{frc}(\dot{x}) = f_m + d \tag{7}$$

where $f_{frc}()$ is friction force.

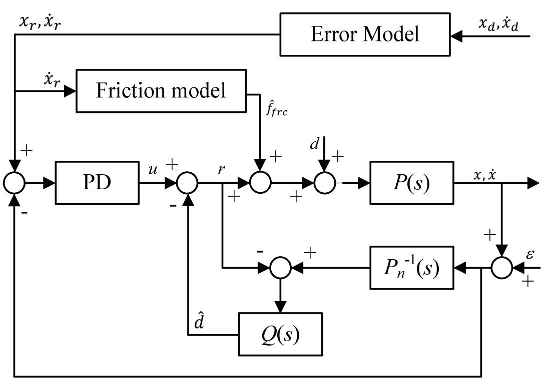

Fig. 3 Proposed position controller

The controller proposed in this study is depicted in Fig. 3, where x_r is the modified desired trajectory and $\hat{f}_{frc}$ is the friction feedforward term. It can be seen that the proposed controller consists of a PD controller, a model based friction feedforward, and a disturbance observer. It can be noted that for the friction feedforward, the desired velocity is used instead of the actual velocity; this is done for the stability of the controller and this configuration is valid as long as the system can reasonably track the given desired trajectory [5].

Omitting the error model feedforward and only considering the PD controller, the friction feedforward and the disturbance observer, Eq. (7) becomes:

$$m\ddot{x} + b\dot{x} = k_p e + k_d \dot{e} + \tilde{f}_{frc}(\dot{x}_d) - \tilde{d} \tag{8}$$

where $\tilde{f}_{frc}(\dot{x}_d) = \hat{f}_{frc}(\dot{x}) - f_{frc}(\dot{x})$, $e = x_r - x$, $\dot{e} = \dot{x}_r - \dot{x}$ and $\tilde{d} = \hat{d} - d$. Assuming that the friction model is reasonably accurate and the cutoff frequency of the DOB is chosen such that the maximum frequency of the disturbance is within the cutoff frequency of the low pass filter, Eq. (8) becomes:

$$k_p e + k_d \dot{e} \approx m_n \ddot{x} + b_n \dot{x} \tag{9}$$

where m_n and b_n are nominal mass and nominal viscous coefficient such that the nominal model is:

$$m_n \ddot{x} + b_n \dot{x} = f \tag{10}$$

From Eq. (9), the transfer function between the modified desired position x_r and the actual resulting position x can be derived as:

$$\frac{X(s)}{X_d(s)} = \frac{k_d s + k_p}{m_n s^2 + b_n s + k_d s + k_p} \tag{11}$$

From Eq. (11), error model $G_e(s)$, which is the transfer function between the modified desired trajectory xr and the resulting error, can be defined as follows:

$$G_e(s) = 1 - \frac{X(s)}{X_d(s)} = \frac{m_n s^2 + b_n s}{m_n s^2 + b_n s + k_d s + k_p} \tag{12}$$

As can be seen from Eq. (12), the error model is a function of m_n, b_n, k_d and k_p, which are known to the controller. Thus, given the reference trajectory, the error model can predict the resulting position error without knowing the model of the system.

Using Eq. (12), the reference trajectory can be modified using Eq. (13), where k_f is the feedforward gain.

$$x_r(s) = \left(1 + k_f G_e(s)\right) X_d(s) \tag{13}$$

4. Simulation

The accuracy of the error model proposed in Eq. (12) is validated through simulations. A 1 DOF linear stage is simulated, with mass of 0.001 kg and viscous coefficient of 0.01 Ns/m. The system is controlled by a PD controller and a DOB at 2 kHz, and it is assumed that the friction of the system is reasonably compensated such that its effect is negligible. The nominal model in Eq. (10) is chosen so that the nominal mass and the nominal viscous coefficients differ from the actual values by 10~20% to simulate possible errors in system model identification. The cutoff frequency of the DOB is set to be 15 Hz.

An acceleration-limited trapezoidal velocity profile is used as the desired trajectory [15], and the maximum velocity and acceleration are chosen as 200 mm/s and 350 mm/s2, respectively. The desired displacement is set to 175 mm. The desired position, velocity and acceleration used for the simulation are depicted in Fig. 4 (a), (b) and (c), respectively.

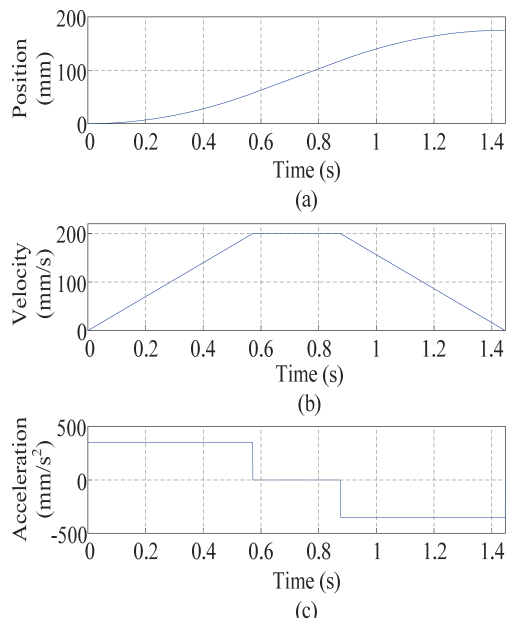

Fig. 4 Desired trajectory for simulation: (a) position, (b) velocity and (c) acceleration

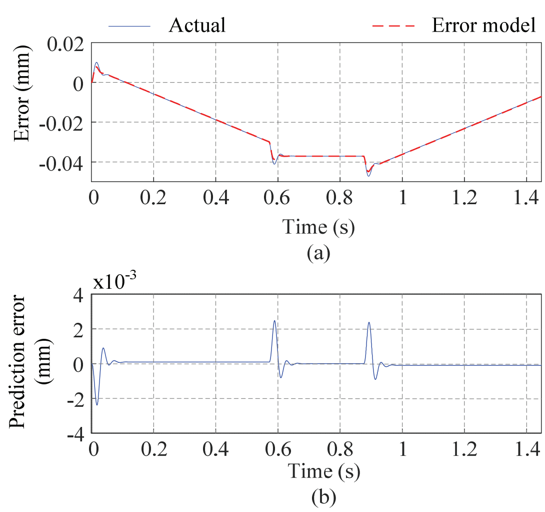

Fig. 5 Error prediction using error model: (a) actual and predicted errors and (b) prediction error

Figure 5 (a) shows the actual tracking error and the tracking error predicted by the error model whereas Fig. 5 (b) illustrates the prediction error of the error model. As can be seen from the results, the error model can accurately predict the position error of the system without using the system model. This implies that the proposed error model can be used to modify the desired trajectory to improve the tracking performance of the controller.

5. Experiments

5.1 Experimental Setup

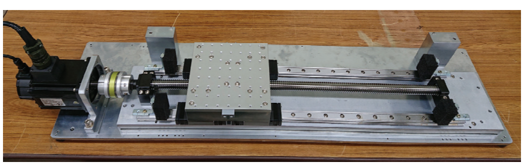

Fig. 6 Experimental setup

Figure 6 shows the 1 DOF linear stage used for the experiments. The ball screw is driven by a 1500 kW servo motor (MSME152G1G, Panasonic, Japan), which has the maximum velocity of 3000 rpm and the continuous torque of 4.77 Nm. A ball screw with pitch of 5 mm is used, and the output of the linear stage is supported by two linear guides.

A PC with real-time patched Linux is used to control the servo motor at 2 kHz through EtherCAT communication with the servo drive (MDDHT5540BA1, Panasonic, Japan). The motor is set to torque control mode (cyclic synchronous torque mode). The servo motor is equipped with a 20-bit incremental encoder, and the available feedbacks from the servo drive are servo motor position, velocity and torque.

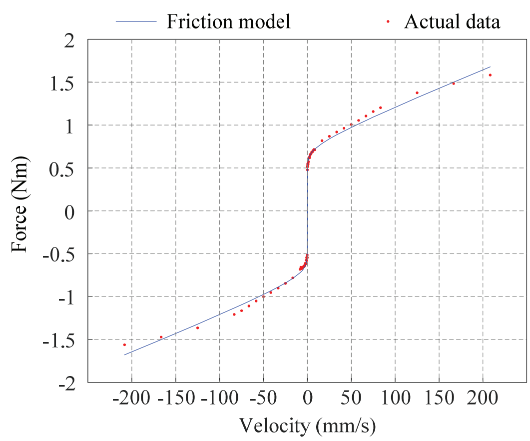

Fig. 7 Friction model

Prior to apply the proposed algorithm, the model of the system is identified. From the motor torque, velocity and acceleration, the mass and viscous coefficient in Eq. (7) can be estimated. As for the friction, LuGre model is used [2]. Average torques at 70 different constant velocities are collected, and the obtained force-velocity data are fitted to estimate the friction coefficients. The obtained friction model is shown in Fig. 7, and it can be seen that the model can accurately estimate the fiction torque.

For smooth motion and vibration suppression, a jerk limited trajectory is used for the experiments [15]. To validate the performance of the proposed algorithm under various circumstances, three trajectories, each with different maximum velocity and acceleration, are prepared. The maximum velocity and acceleration of 50 mm/s and 50 mm/s2 are used for trajectory 1 whereas 125 mm/s and 300 mm/s2 are used for trajectory 2 and 215 mm/s and 700 mm/s2 are used for trajectory 3. The maximum velocity and acceleration of each trajectory is chosen considering the capacity of the motor used for the experiments. The position, velocity, acceleration and jerk of trajectory 2 are depicted in Fig. 8 (a), (b), (c) and (d), respectively. The displacement of 200 mm and jerk of 1000 mm/s3 are used for all three trajectories.

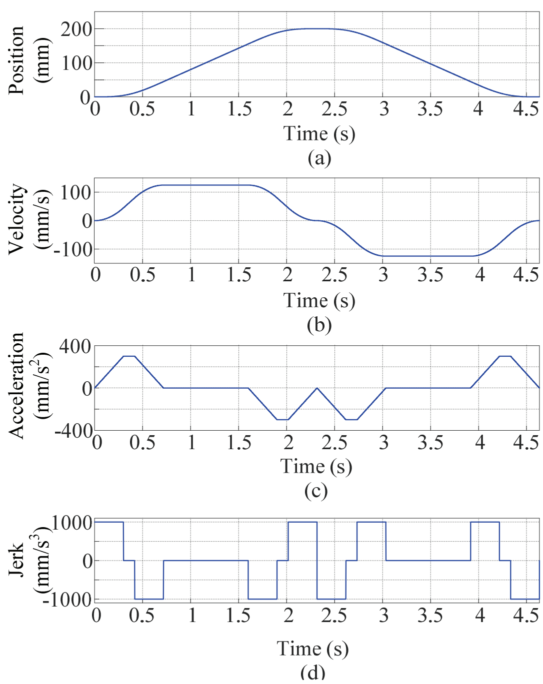

Fig. 8 Desired trajectory: (a) position, (b) velocity, (c) acceleration and (d) jerk

To control the 1 DOF system, a PD controller with a DOB is implemented on the PC. The nominal mass mn in Eq. (10) is chosen as the same as the actual system mass obtained from the model identification step. As for the nominal viscous coefficient bn (Eq. (10)), a value larger than the actual value is used to reduce the vibration. The cutoff frequency of the DOB is experimentally chosen to be 15 Hz, to maximize the disturbance suppression performance of the DOB while ensuring the stability of the system. The gains of the PD controller is also chosen experimentally. For the modification of the desired trajectory, 1.0 and 0.7 are used as the feedforward gain kf in Eq. (13) for trajectory 1 and 2, and 3, respectively.

5.2 Experimental Results

To show the performance of the proposed algorithm, comparative experiments have been conducted using the 1 DOF linear stage on three prepared trajectories using two control algorithms: 1) proposed controller and 2) the PD controller with friction feedforward. The resulting errors are presented in Fig. 9, and the root mean square (RMS) and the maximum values of errors for each case are given in Table 1.

As can be seen from the results presented in Fig. 9 and Table 1, the proposed algorithm can greatly improve the position tracking performance of the PD controller. Clear reductions in both the maximum errors and

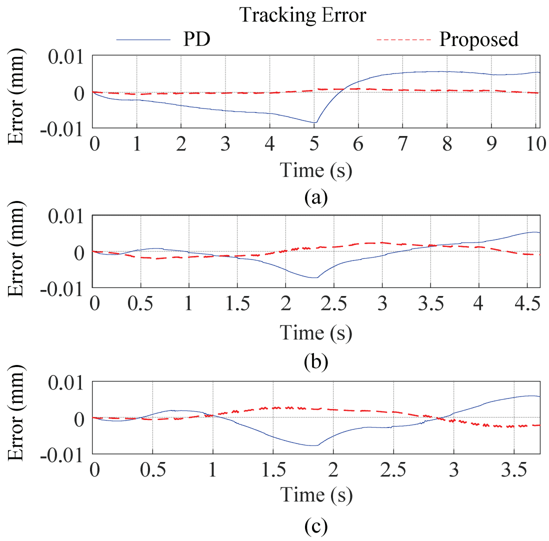

Fig. 9 Resulting position error: (a) trajectory 1, (b) trajectory 2 and (c) trajectory 3

Table 1. RMS and maximum values of position errors

	PD		Proposed	
	RMS (mm)	Maximum (mm)	RMS (mm)	Maximum (mm)
Trajectory 1	0.0048	0.0085	4.45e-4	8.5e-4
Trajectory 2	0.003	0.0073	0.0014	0.0025
Trajectory 3	0.0037	0.0077	0.0017	0.0029

the RMS values of the error can be noted.

The error model proposed in this paper is based on the assumptions that friction force can be reasonably compensated and the DOB can compensate the disturbances so that the system would behave like the nominal model defined by the DOB. However, as shown in section 3, the second assumption would no longer valid if the frequency of the disturbance is higher than the cutoff frequency of the DOB. It is important to set the cutoff frequency of the DOB as high as possible; however, high cutoff frequency may leads to instability. During the experiments, it was noted that a higher cutoff frequency can be used at a faster control frequency. In this study, the proposed algorithm is implemented on the PC and thus, its control period is limited by the communication speed between the PC and the driver. It is expected that much higher cutoff frequency can be used if the algorithm is implemented on the motor drive and thus much faster control loop can be established.

The proposed control algorithm is simple in structure and the required computational load is light. Furthermore, if needed, the algorithm can be implemented in offline to minimize the online computational load. In this study, a control structure with a single PD controller is considered; however, many studies uses a cascade PID controller [6, 12] or a feedforward controller [12, 14]. The proposed algorithm can be easily expanded for these control structures by updating Eq. (8) to find the error model corresponding to the given controller. Another advantage of the algorithm is that the accurate model of the system is not required. The nominal model is required for the DOB; however, as can be seen from the simulations and experiments, the nominal model does not have to be the same as the actual plant model.

6. Conclusion

In this study, a novel position control algorithm based on the PD controller and the DOB is proposed. The dynamics of a ball screw drive under PD control and DOB is analyzed, and the error model of the system is derived. Based on the error model, the desired trajectory can be modified to compensate for the predicted position error in a feedforward fashion to improve the position tracking accuracy of the system. The performance of the proposed algorithm is verified through simulation and experiments. For the future study, the algorithm will be implemented on a servo drive, to achieve higher control frequency, which would improve the accuracy of the error model and disturbance rejection performance of the DOB.

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References

[1] Dahl, P. A.: Measurement of solid friction parameters of ball bearings. Proceedings of the 6th annual symposium on Incremental Motion, Control System and Devices, 25, (1977).
[2] Johanastrom, K. and Canudas-de-Wit, C. IEEE Contr. Syst. Mag., 28, 6, 101, (2008).
[3] Al-Bender, F., Lampaert, V. and Swevers, J. IEEE T. Automat. Contr., 50, 11, 1883, (2005).
[4] Erkorkmaz, K. and Kamalzadeh, A. CIRP Annals, 55, 1, 393, (2006).
[5] Jamaludin, Z., Brussel, H. V., Pipeleers, G. and Swevers, J. CIRP Annals, 57, 1, 403, (2008).
[6] Zhang, C. and Chen, Y. IEEE T. End. Electron., 63, 12, 7682, (2016).
[7] Huang W. S., Liu, C. W., Hsu, P. L. and Yeh, S. S. IEEE T. End. Electron., 57, 1, 420, (2010).
[8] Han, J. IEEE T. End. Electron., 56, 3, 900, (2009).
[9] Wu, J., Han, Y., Xiong, Z. and Ding, H. Int. J. Mach. Tool. Manu., 117, 1, (2017).
[10] Onishi, K. Trans. Japanese Society of Electrical Engineering, 107, D, 83, (1987).
[11] Eom, K. S., Suh, I. H. and Chung, W. K.: Disturbance observer based path tracking control of robot manipulator considering torque saturation. Proceedings of 8th international Conference on Advanced Robotics, Monterey, pp.7-9, (1997).
[12] Hsia, T. C., Lasky, T. A. and Guo, Z.: Robust Independent Robot Joint Control: Design and Experimentation. Proceedings of IEEE International Conference on Robotics and Automation, Philadelphia, pp.1329-1334, (1988).
[13] Hanifzadegan, M. and Nagamune, R. J. Dyn. Sys. Meas. Control., 136, 1, 10.1115/1.4025154, (2013).
[14] Liu, C., Liu, J., Wu, J. and Xiong, Z: High precision embedded control of a high acceleration positioning system. Proceedings of International Conference on Intelligent Robotics

and Applications, Montreal, pp.551-560, (2012).
[15] Altintas, Y.: Manufacturing Automation: Metal Cutting Mechanics, Machine Tool Vibrations, and CNC Design. 2nd Ed., Cambridge University Press, New York, pp.211-239, (2012).

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